IARJSET
ISSN (Online) 2393-8021
ISSN (Print) 2394-1588
International Advanced Research Journal in Science, Engineering and Technology
ISO 3297:2007 Certified
Vol. 3, Issue 11, November 2016

Study of Thermal Performance between Plate-fin, Pin-fin and Elliptical Fin Heat Sinks in Closed Enclosure under Natural Convection

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Abstract: Thermal performances of plate-fin, pin-fin and elliptical heat sinks with vertical base plate were compared in natural convection. Comparison is performed with same base plate dimensions and height of fin condition. In the work herein, steady-state natural convection heat transfer and thermal performance comparison between rectangular finned heat sinks and pin fin heat sink from vertically-oriented base plate is investigated. After generating and validating the existing analytical results for continuous fins, a systematic numerical study is conducted on the effect of the elliptical fin. ANSYS and SOLIDWORKS software are used in order to develop a three-dimensional numerical model for investigation of different fin geometries effects. Results show that alterations in fin geometry to vertical oriented base plate fins enhances the thermal performance of fins and reduces the weight of the fin arrays, which leads to lower manufacturing costs. The optimum spacing for maximum fin array thermal performance is found. This study suggests that the most important geometric parameter influencing the heat transfer from pin fin arrays is the ratio of the fin diameter to the center to center spacing.

Keywords: Fins, natural convection, numerical model, optimization.

I. INTRODUCTION

Sparrow and Vemuri [1] studied the optimal number of pin-fin with fixed base plate dimensions and fin diameter. With the optimized result, they compared pin-fin and plate-fin heat sinks. However, their comparison was performed based on the same surface area. The design of efficient and economical cooling methods is crucial for reliable performance of high power density electronics. A number of failure mechanisms in electronic devices, such as inter-metallic growth and void formation, are connected to thermal effects. In fact, the rate of such failures nearly doubles with every 10°C increase higher than the operating temperature ~80°C of high power electronics. Besides the damage that excess heat will cause it to increases the movement of free electrons within conductors and semiconductors, causing associated degree increase in signal noise. Consequently, electronics thermal management is of crucial importance as is reflected back in the market. Natural convection heat sinks have various configurations to enhance the thermal performance but typically plate-fin and pin-fin array are widely used because they are cost-effective. Plate-fin array usually has larger surface area than pin-fin array because of the area loss caused by cross cut in pin-fin array. Pin-fin array usually has higher heat transfer coefficient because of the depression of growth of the thermal boundary layers.

Heat sinks function expeditiously by efficiently transferring thermal energy that is heat from an object at high temperature to a second object at a lower temperature with a much larger heat capacity. This rapid transfer of thermal energy quickly brings the initial object into thermal equilibrium with the second, lowering the temperature of the first object, fulfilling the heat sink's role as a cooling device. Efficient operation of a heat sink depends on speedy transfer of thermal energy from the former object to the heat sink and the heat sink to the latter object. The most common design of a heat sink is a metal arrangement with many fins. The high thermal conductivity of the metal combined with its large surface area result in the rapid transfer of thermal energy to the encircling cooler, air. This cools the heat sink and whatever it’s in direct thermal contact with. Use of fluids such as coolants used in refrigeration and thermal interface material in heat dissipation from electronic devices ensures good transfer of thermal energy to the heat sink. Similarly, a fan may improve the transfer of heat energy from the heat sink to the air. Currently, the thermal losses of power electronic devices are increasing. At the same time, their sizes are decreasing. Consequently, heat sinks have to dissipate higher heat fluxes in every new design. Therefore, devising efficient and economic cooling solutions to meet these challenges is of dominant importance and has direct impacts on the performance and dependability of electronic and power electronic devices.

Also, extracting the generated heat from the element will increases component’s potency, efficiency and also its performance .Eventually profiting the user. The techniques used in the cooling of high power density electronic devices may vary largely, and depending on the application and the required cooling capacity. The heat generated by the electronic components needs to pass through a very complex network of thermal resistances to the environment. Passive cooling methods are most likely

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preferred for electronic and power electronic devices since they provide low-price, noiseless, and trouble free solutions. Some passive cooling techniques include: heat pipes, natural convection air cooling, and thermal storage using phase change materials (PCM). Heat pipes can very efficiently transfer heat from heat sources in high power density converter components to a heat sink based on phase change of a working fluid.

Air-cooling also is recognized as an important technique in the thermal design of electronic packages, because besides its availability, it is safe, does not contaminate the air and does not add vibrations, noise and moisture to the system in which it is used. Such features of natural convection stimulated considerable research on the development of optimized finned heat sinks and enclosures. Using fins is one of the most inexpensive and common ways to dissipate unwanted heat and it has been successfully used for many engineering applications.

There have been many researches about plate-fin heat sinks [2, 3, 4, 5, and 6], so plate-fin heat sinks would be optimized using correlations suggested by previous researches. The objective of the present study is to determine which type of heat sink performs better under fixed volume conditions between plate-fin heat sinks and pin-fin heat sinks and elliptical heat sink. The fixed volume condition means that, physically, the space specified by the length, width, and fin height of a heat sink is fixed in the present study. Objective function used for the comparison is total heat dissipation for a given base-to-ambient temperature difference.

Thus, the higher value of the first objective function means a greater heat dissipation capability under a given volume of a heat sink. For plate-fin heat sink, pin fin heat sink the correlation of the heat transfer coefficient suggested by Younghwan Joo and Sung Jin Kim [7] is used. In the present study, thermal performances of plate-fin and pin-fin and elliptical heat sinks with vertical base plate would be compared based on the same base plate dimensions and fin height condition and fixed volume condition To achieve maximum thermal performance for the constraints, each array type would be optimized using correlations of convective heat transfer coefficient.

II. DESCRIPTION AND PRINCIPLES OF FINS

As fins are introduced to enhance heat transfer from a base which is at high temperature rectangular fins, cylindrical fins and elliptical fins are considered for analysis. According to Incropera F. P., DeWitt D. P [8] the term extended surface is commonly used to depict an important special case involving heat transfer by conduction within a solid heat transfer by convection from the boundaries of a solid. Although there are many different situations that involve such combine conduction- convection effect, the most frequent application is one in which an extended surface is used to increase the heat transfer rate between the solid and adjoining fluid.

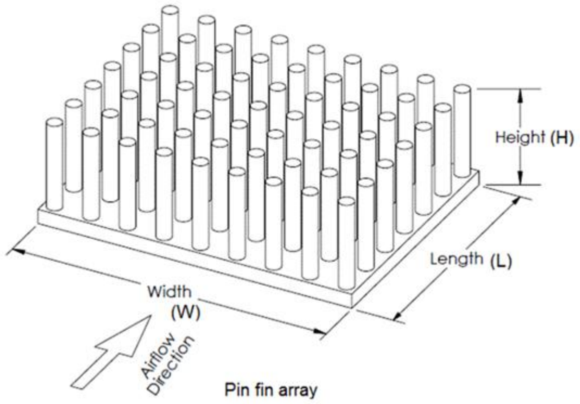

Fig1.Heat sink with inline array of pin fin

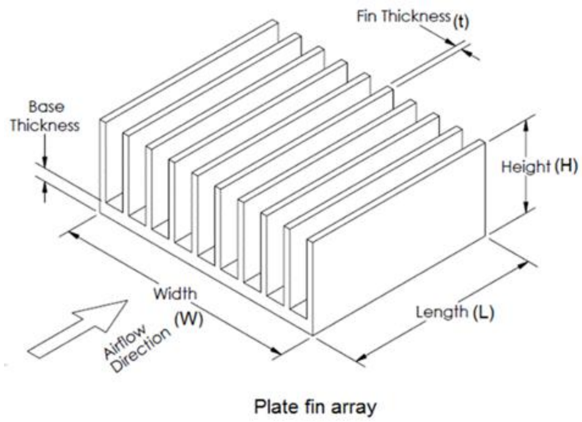

Fig2. Heat sink with array of plate fin arrangement.

Such an extended surface is called is called as fin. While heat sinks are routinely used in most electronics applications, the rationale for selecting a particular design of heat sink or more specifically a particular fin cross sectional profile, remains somewhat uncertain. Most often these types of selection procedures are based exclusively on performance evaluations consisting of formulations for extended surface heat transfer found in most fundamental heat transfer text book. Geometry of fins plays a very important role in performance characteristics. Therefore fin geometry is optimized for maximum heat dissipation through extended surfaces As mentioned above to enhance the ability of heat transfer through fins main concentration lies on fin available surface area for convection. Different types of fins used in heat sink are rectangular fin, cylindrical fins, splayed fins, triangular fins etc. our main concentration in this study is of optimization pin fins. And to carry out further investigation on thermal characteristic of elliptical fin heat sink.

Design considerations for heat sinks Material: Copper has excellent heat transfer characteristics which makes it one of the best materials to be used for heat sinks. Copper is also resistant to corrosion and bio fouling. Copper finds its application in industrial thermal facilities, solar power systems, power plants, HVAC systems, etc. However, most of the heat sinks used commercially for electronic chip cooling are made of aluminum alloys. Aluminum

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alloys have heat transfer coefficients. Aluminum alloys are preferred over copper for commercial applications as they are cheaper than copper.

Thermal Resistance: Thermal is a measure of the temperature difference by which a material resists heat flow. Thermal conduction is a material property. Thermal resistance is the temperature difference across a system when heat flows through it.Thermal resistance gains its importance in systems and not individual components. Resistances occur at the interface between two connected parts. When individual parts are connected, micro gaps are formed at the interface due to grooves and crests that are inherently present on the surface. The gaps thus formed get filled with air, which has a low thermal conductivity. This means that the energy has to get transferred through a medium of low conductivity.

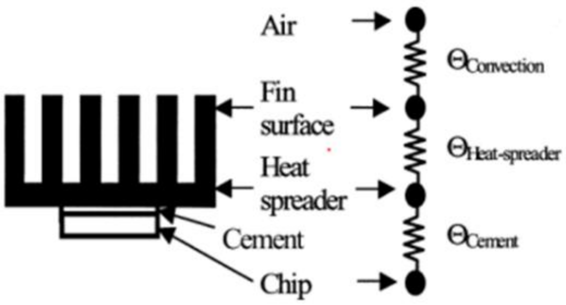

Fig3 Thermal resistances generally encountered in chip cooling

This reduces the overall heat transfer coefficient of the system. Although thermal resistance between surfaces cannot be totally overcome, it can be mitigated. There are various ways of managing these resistances.

- Use of thermal tape. Twin sided tapes made of thermally conductive materials are one of the widely used, cheapest form of attachment that is used. The tape, being compressive, reduces the number of air pockets when pressure is applied.
- Epoxy. Epoxy paste can be applied between the two contacting surfaces to reduce the thermal resistance. It is not as common as thermal tape as it is more expensive than the tape.
- Pressure. Contacting surfaces can be held tightly in place by use of pressure. Pressure is exerted on the surfaces by use of clips, push pins with compression springs.

III. MATHEMATICAL FORMULATIONS

3.4.1 Governing equations

For the numerical analysis, the following assumptions are imposed.

(1) The flow is steady and three dimensional.

(2)The Boussinesq approximation is used for natural convection flows.

(3)The direction of the gravitational acceleration is the negative x-direction.

Equations governing natural convection

Continuity equation

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \qquad \text{(i)}$$

Momentum equation

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} = -g - \frac{1}{\rho}\frac{\partial P}{\partial x} + v\left(\frac{\partial(\partial u)}{\partial(\partial x)}\right) + \left(\frac{\partial(\partial u)}{\partial(\partial y)}\right)\left(\frac{\partial(\partial T)}{\partial(\partial z)}\right) \qquad \text{(ii)}$$

Energy equation

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z} = \alpha\left(\frac{\partial(\partial T)}{\partial(\partial x)} + \frac{\partial(\partial T)}{\partial(\partial y)} + \frac{\partial(\partial T)}{\partial\,\partial z}\right) \qquad \text{(iii)}$$

Surface area of plate-fin heat sink can be expressed with design parameters shown in Figure 1

$$A_{array} = n_{fin} \times (2HL + 2w_c H + w_w L) \qquad \text{(iv)}$$

$$A_{base} = WL - n_{fin} w_w L \qquad \text{(v)}$$

$$n_{fin} = (W - w_c) / (w_w + w_c) \qquad \text{(vi)}$$

Surface area of pin-fin array can be calculated using equations as follows.

$$A_{array} = n_{fin} \times \pi d\,(H + d/4) \qquad \text{(vii)}$$

$$A_{base} = WL - n_{fin}\frac{\pi d^2}{4} \qquad \text{(viii)}$$

Surface area for elliptical fin array can be calculated using

$$A_{array} = n_{fin}\, 4\pi\left(\frac{(ab)^{1.6} + (ac)^{1.6} + (bc)^{1.6}}{3}\right)^{.625} \qquad \text{(ix)}$$

$$A_{base} = WL - (n_{fin} * \pi ab) \qquad \text{(x)}$$

For calculating optimized number of fin fins in array

$$n_{fin} = n_v n_h + (n_v - 1)(n_h - 1) \qquad \text{(xi)}$$

$$n_v = (L-d)/s_v + 1 \qquad \text{(xii)}$$

$$n_h = (W/2 - d)/s_h + 1 \qquad \text{(xiii)}$$

The terms H,L, W, n_{fin}, n_h, n_v, A_{array}, A_{base}, w_c, w_w height of fin, length of fin, width of fin, number of fins, number of fins in horizontal direction, number of fins in vertical direction, area of fins on base plate area of base plate, channel spacing in plate fin, width of plate fin respectively.

With above equations, we can express Q_{total} which means total heat dissipation from heat sink.

IARJSET

ISSN (Online) 2393-8021
ISSN (Print) 2394-1588

International Advanced Research Journal in Science, Engineering and Technology

ISO 3297:2007 Certified

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$$Q_{total} = Q_{base} + (xv) \tag{xiv}$$

$$Q_{total} = (h_{base}\, A_{base} + h_{fin}\, A_{array}\, \eta_{fin})\,(T_b - T_\infty) \tag{xv}$$

With above equations, we can express Qtotal which means total heat dissipation from heat sink.

Then, thermal resistance (Rth) which is target dependent variable for thermal performance comparison is expressed as follow. Thermal resistance in fins is given by difference in temperature to total heat dissipated by fins

$$R_{th} = \frac{Tb - T\infty}{Q} = \frac{Tb - T\infty}{(hbase\ \ Abase + hfin\ Aarray\ \ \eta fin\ )} \tag{xvi}$$

Efficiency of fins is expressed by following equation

$$\eta_{fin} = \frac{\tanh\ (mH\,)}{mH} \tag{xvii}$$

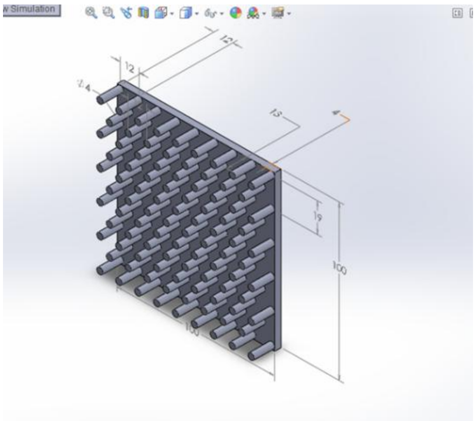
IV. NUMERICAL SIMULATION

Thermal steady state, commercial software based on the finite volume method and provided by ANSYS, Inc., is used as the simulation tool. The geometrical configurations for the numerical model are shown in Fig.5. The model consists of a heat sink base and fins. For the numerical analysis, the following assumptions are imposed.

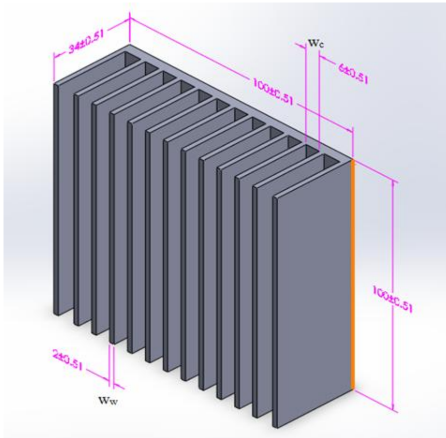
(1) The flow is steady and three dimensional.
(2) The Boussinesq approximation is used for natural convection flows.
(3) The direction of the gravitational acceleration is the negative x-direction.
The governing equations are as follows: For the fluid phase,
Continuity equation, momentum equation, energy equation given in (i),(ii) and (iii)

The boundary conditions are indicated in Fig.5. The boundary condition for the min.-y side wall is a symmetric condition. The boundary conditions for the other walls are opening conditions with ambient temperature and ambient pressure.

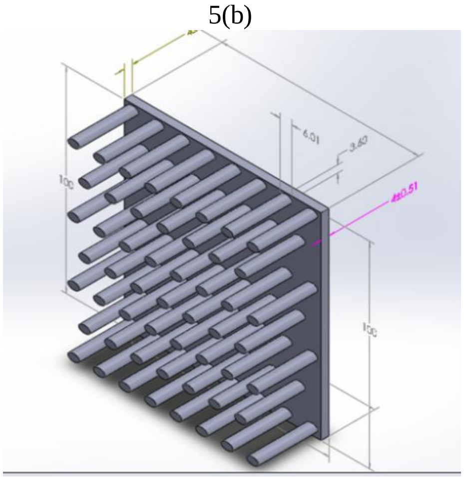
In the present study, ambient temperature is assumed to be 20 ^{O}C. The boundary condition for the bottom of a heat sink base is a constant heat flux condition.



5(a)



5(b)



5(c)

Fig.5 (a) (b) (c) Geometry modelling for pin fin, plate fin and elliptical fin heat sink on SOLIDWORKS Software.

Table1.Dimensions of pin-fin heat sinks for numerical simulation.

Type of heat sink	L(mm	W(mm)	H(mm)	d(mm)/ w_w	s_v (mm)	s_h (mm)/ w_c
Pin fin1	100	100	30	4	19	12
Pin fin2	100	100	20	1.8	19	8
Pin fin3	100	100	20	4.9	19	12
Pin fin4	100	100	30	2.6	19	10
Plate fin1	100	100	30	1.20	-	8
Plate fin2	100	100	30	2	-	5.28

IARJSET
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International Advanced Research Journal in Science, Engineering and Technology
ISO 3297:2007 Certified
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Table 2 Geometric modelling parameters for elliptical heat sink under fixed volume condition.

Heat sink	Minor diameter (d_1) mm	Major diameter (d_2) mm	Height of fin (H) mm
Elliptical fin1	2.54	4.46	30
Elliptical fin2	2.3	3.84	30
Elliptical fin3	3.6	6.012	30

Domain selection for fin array: When a natural convection problem is considered, the computational domain size should be determined carefully because the accuracy of the results is affected by the domain size. To determine the domain size, the effects of the height, the width, and the length of the domain on the numerical results are investigated.

Increasing the height (z-direction), the width (x-direction), and the length (y-direction) beyond 10H, 5L, and 7L, respectively, changes the average heat sink base temperature by less than 0.05 oC. Therefore, the domain size is determined to be 10H for the height, 5L for the width, and 7L [10] for the length for simulation.

Meshing: Meshing of geometries imported form solid works software is done on CFD package ANSYS under meshing setup. Medium meshing is selected as

• Meshing size is selected as 2.50mm
• Medium smoothing is assigned
• Transition is by default set as fast.
• Span angle is set at course.
• Minimum edge length is 2mm
• Mesh morphing is disable for numerical simulation
• Number of nodes in meshing 106737
• Number of elements in meshing 63222

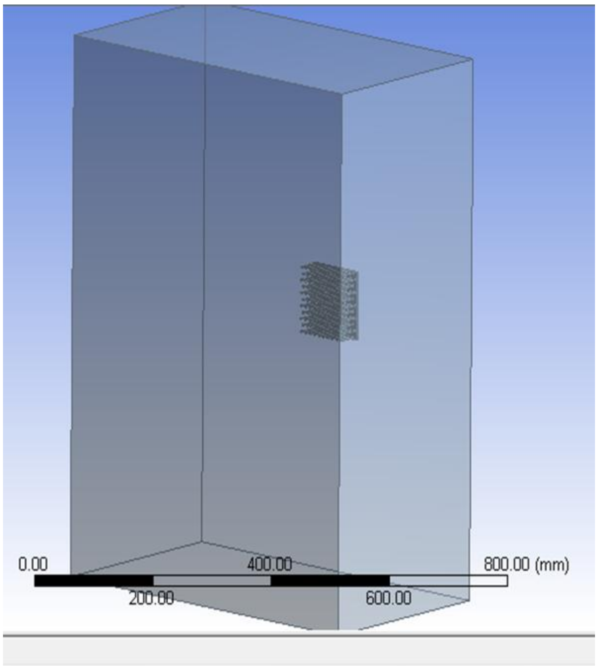

Fig.6 Computational domain creation for numerical simulation on Ansys Workbench.

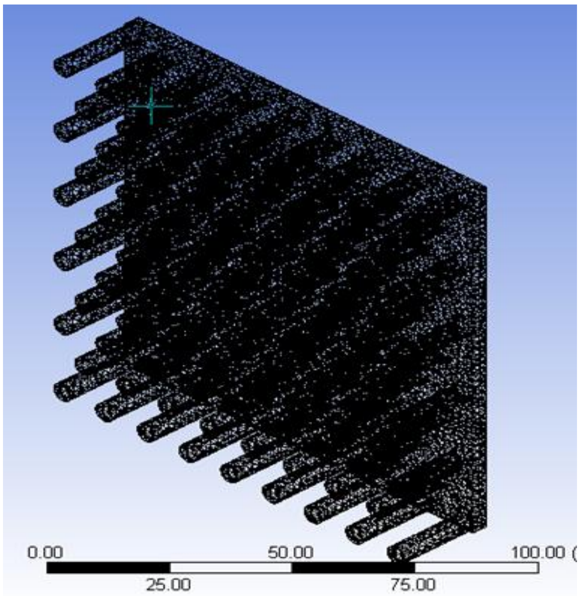

Fig7(a) (a) Meshing size 1mm

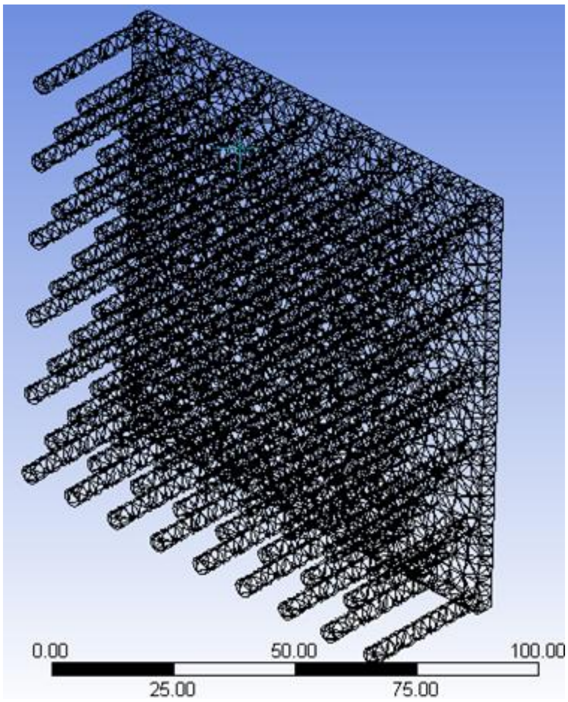

Fig7 (b) (a) Meshing size 1mm

Above Fig 7(a) and 7(b) depicts meshing visualization of pin fins as element size 1mm in (a) and element size as 2.50mm in (b) on making the meshing size finer no considerable change in temperatures were obtained so meshing size is set at 2.50 mm saving computational time and cost.

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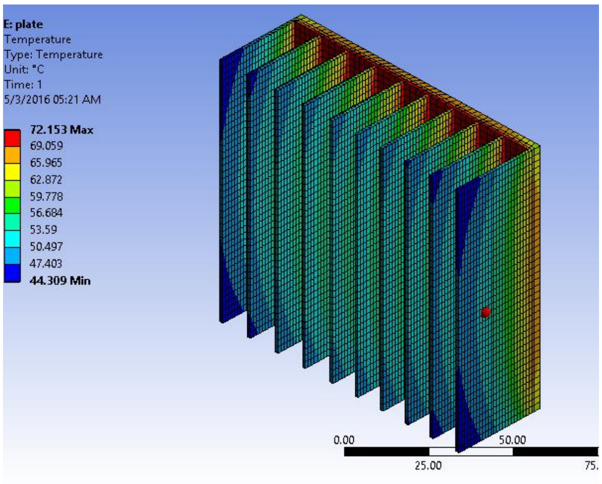

Fig 8(a). Steady state contours showing temperature variation in plate fin heat sink

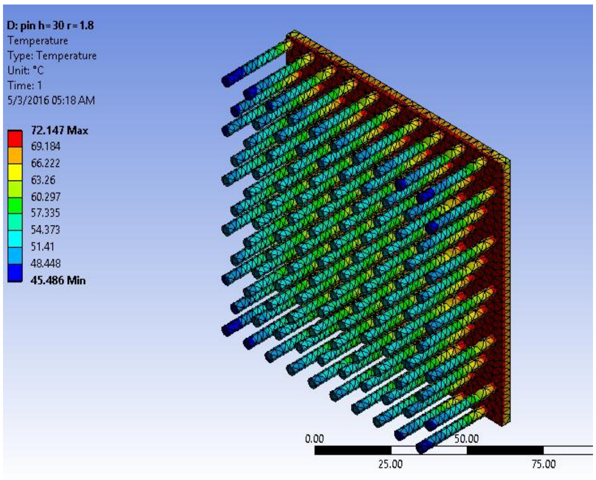

Fig8(b) Steady state contours showing temperature variation in pin fin heat sink

Optimization of heat sink

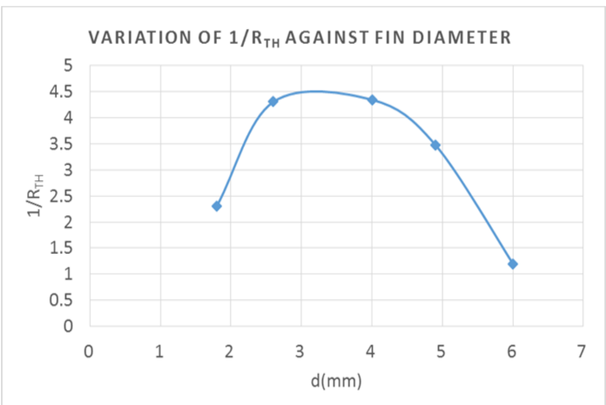

Fig.9 Depiction of optimization of pin fin versus diameter.

Thermal performances of plate-fin and pin-fin heat sinks are compared analytically. The constraints for the comparison are same base plate dimensions, 2) same fin height. The definition of thermal resistance ratio is as follow

$$R_{th,\ ratio1} = \frac{R_{th,pin}}{R_{th,plate}}\ ,\quad R_{th,\ ratio2} = \frac{R_{th,pin}}{R_{th,elliptical}}\ ,$$

$$R_{th,\ ratio3} = \frac{R_{th,elliptical}}{R_{th,plate}}$$

$R_{th,\ ratio} > 1$ shows that plate fin heat sink has better performance than pin fins according to [4] Younghwan Joo, Sung Jin Kim. And also better than elliptical heat sink.

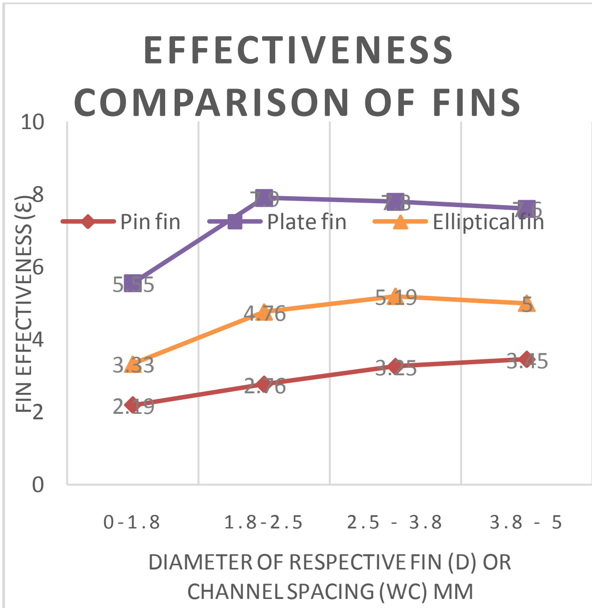

Fig.10 Effectiveness comparison of fins on line graph

Graph obtained shows the superiority of vertically oriented plate fins in natural convection. From above graph it is clear that elliptical fins can replace pin fins or serve as an alternative to pin fins. As surface area available for elliptical fin geometry for natural convection is much higher than those of pin fins. But when comparison is made with plate fin, results show that plate fin is still better than pin fin and elliptical fins for total heat dissipation. To compare the optimized heat sinks, each heat sink type was optimized based on the correlations for convective heat transfer coefficient of fin array. So we come to a conclusion that heat dissipation rate in plate in is highest due to large surface area available for convection. Then elliptical fin emerge as more efficient and reliable than pin fin for heat dissipation under natural convection.

V. CONCLUSION

In this study result are evaluated on basis of TOTAL HEAT DISSIPATION under fixed volume condition.
In the present study, thermal performances of plate-fin and pin-fin and elliptical fin heat sinks were compared for the fixed base plate dimensions and fin height under fixed volume condition. When objective function is considered plate fin performs better than pin fins. Thermal performances of plate-fin and pin-fin heat sinks were compared for the fixed base plate dimensions and fin

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ISSN (Print) 2394-1588
International Advanced Research Journal in Science, Engineering and Technology
ISO 3297:2007 Certified
Vol. 3, Issue 11, November 2016

diameter. To compare the optimized heat sinks, each heat sink type was optimized based on the correlations for convective heat transfer coefficient of fin array. Using the correlations for both types of heat sinks, thermal performances were compared and a region map was suggested. According to the region map, we can conclude that plate-fin heat sinks show better thermal performance in most practical regions. Therefore, it is recommended to use plate-fin heat sinks when the total heat dissipation for a given volume of a heat sink is to be maximized under fixed volume condition.

Plate-fin array usually has larger surface area than pin-fin array because of the area loss caused by cross cut in pin-fin array. Pin-fin array usually has higher heat transfer coefficient because of the depression of growth of the thermal boundary layers. When elliptical fins comes in comparison then it is better to use elliptical fins rather than pin fins. But elliptical fins lack the large surface area available in plate fins for natural convection. They dissipate lower heat energy than plate fin ejects heats from the equipment.

So therefore plate fin dissipate the most heat from the equipment in case of total heat dissipation in present study. As surface area after optimization of each fins, plate fins control the largest area available for convection under fixed volume condition. So, least material will be used to manufacture plate fin other than pin fin and elliptical fin. Therefore at last we can conclude that elliptical fins can be a replacement for pin fins. But elliptical fin were not able to compete plate fins as larger area is exposed to convection in plate fins. The elliptical pin heat sink analysis represents only one set of design parameters relating pin spacing and shape based upon minor and major axes. There may exist other designs which produce better results in overall thermal performance. A study looking at reduced spacing, pin alignment, pin staggering, and an array of ellipse axis ratios would be advantageous to the heat sink industry.

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BIOGRAPHY

Akshendra Soni is a post graduate in Thermal Engineering from Gautam Buddha University, Greater Noida. He is working as a lecturer in Bundelkhand Institute of Engineering and technology. His areas of interest include modelling and simulation in heat transfer and fluids flow.

Journal of Refrigeration, Air Conditioning, Heating and Ventilation
ISSN: 2394-1952 (Online)
Volume 4, Issue 2
www.stmjournals.com

Investigation of Free Convection Heat Transfers in an Attic Shaped Enclosure

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Absrtact

Numerical investigation of free convection heat transfers in an attic shaped enclosure with differentially heated two inclined walls and filled with air is performed in this study. The left inclined surface is uniformly heated whereas the right inclined surface is uniformly cooled. There is a heat source placed on the right side of the bottom surface. Rest of the bottom surface is kept as adiabatic. Finite volume based commercial software ANSYS 15 (Fluent) is used to solve the governing equations. Dependency of various flow parameters of fluid flow and heat 0.2 to 0.6, heater position from 0.3 to 0.7 and aspect ratio from 0.2 to 1.0 with a fixed transfer is analysed including Rayleigh number, Ra ranging from 10^3 to 10^6, heater size from Prandtl number of 0.72.

Keywords: *Natural convection, attic shape, numerical model*

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INTRODUCTION

Buoyancy induced flows form an important integral part of the subjects of fluid dynamics and heat transfer because these flows display a rich variety of dynamical behavior. Buoyancy driven flows in fluid motions and transports that affect human lives are found in the air circulation around human bodies in the enclosures. In recent years, considerable interest is shown in understanding the process induced by buoyancy forces, due to the growing demand for detailed subjective data concerning such prompted movements in the climate, water bodies and semi - strong bodies, for example, the earth, the enclosures and process equipment. Buoyancy induced flows caused by thermal transport or combination of thermal and diffusion effects form a topic of interest because of its applications in engineering and technology. For example, amid cleaning operations lingering liquid diffuses into the encompassing liquid in the curing of plastics, in the make of mash protected cables and in many chemical processes where concentration differences of dissimilar species exist [1].

Buoyancy driven flows square measure thought of advanced as a result of essential coupling between transport phenomena of flow and thermal fields. Specifically, internal flow issues square measure significantly additional advanced than external ones. This is often as a result of massive Grashof variety, classical physical phenomenon theory assumes simplifications for external flow issues, namely, the region outside the physical phenomenon is unaffected by the physical phenomenon. For confined natural convection, in distinction, boundary layers type close to the walls; however, the region external to them is self-enclosed by the boundary layers and forms the core region. Since the core is partly or totally encircled by the boundary layers, the core flow isn't without delay outlined from the physical phenomenon conditions; however, rely on the physical phenomenon that successively, is influenced by the core. The interactions between the boundary layer and the core constitute major complexity in the problem. This coupling between core and the boundary layer constitutes major difficulty in obtaining analytical solutions. Hence, internal natural convection phenomena are mostly analyzed by numerical or experimental techniques [2,3].

The study of laminar natural convection in rectangular shaped enclosures with two walls at obligatory temperatures provides a helpful description of the behavior of the confined fluids in several sensible things. However, departures from the essential state of affairs are typically encountered. In fields like alternative energy assortment and cooling of electronic parts, the active walls are also subjected to abrupt temperature non-uniformities as a result of shading or different effects. Also, the relative positions and size of the new and cold wall regions have vital effects on the flow patterns and warmth transfer. Previous studies of natural convection in cavities have targeted primarily on steady and transient natural convectional flow in rectangular cavities with each vertical walls at uniform; however, totally different temperatures [4]. Steady-state natural convection in rectangular enclosure with part heated walls has been extensively reviewed. But, terribly less attention has been given to the unsteady natural convection with synchronal partial heating and cooling of vertical walls.

In the gift study, thus, AN unsteady natural convection downside during a sq. enclosure with part heated and cooled vertical walls are being analyzed numerically. Its assumed that the heated and cooled surface components face one another in an opposed manner. Very cheap wall is assumed to be linearly heated/thermally insulated and therefore, the high wall is assumed to be adiabatic. This pure mathematics will be accustomed model flow conditions in furnaces or within electronic parts consisting of huge arrays of vertical heat generating components [5,6].

Rafee et al. [7] investigated unsteady natural convection in an exceedingly part heated rectangular cavity and conferred results for variety of heated surface component sizes, cavity facet ratios and Lord Rayleigh numbers. The results showed that the temporal changes in heat transfer were very little laid low with the position of the heated component unless the component was near the horizontal wall.

PHYSICAL MODEL

Physical model is characterized in Figure 1. Measurements and limit state of each side wall is shown. In this model, heater is situated at the base cross section with length of L and separation of its middle to the triangle midline is S. Length of triangle height and half base is H and W. Right and left slanted walls have constant cold and hot temperatures while the base heater is warmed to a same temperature from left wall. Different parts of triangle base are adiabatic. No slip limit condition is relegated for the whole-body enclosure.

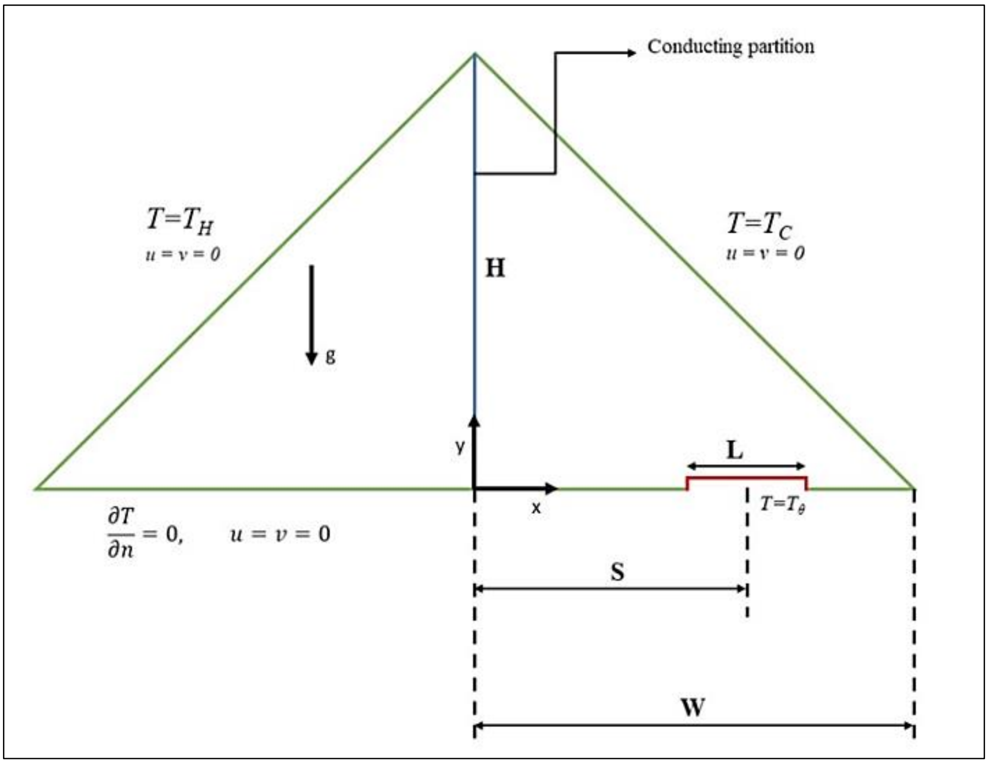

Fig. 1: *Physical Model.*

Geometrical Modelling and Meshing

The geometrical modelling of physical model is created on design modeler of ansys workbench. A triangular enclosure is created with vertical partition and also the heater cavity is created at right side of the bottom surface (Figure 2).

Meshing is one of the most important part of which helps us to give the more accurate result analysis. Following figure shows the computational distribution of mesh inside the triangular enclosure. The following meshing is performed in ansys workbench in which 15789 nodes are created (Figure 3).

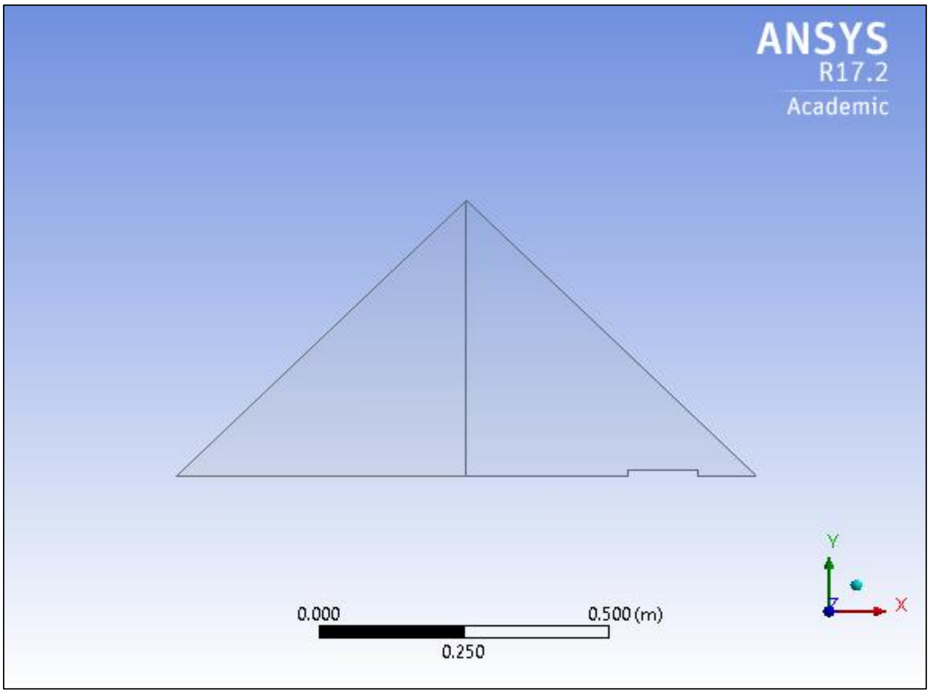

Fig. 2: *Geometrical Model.*

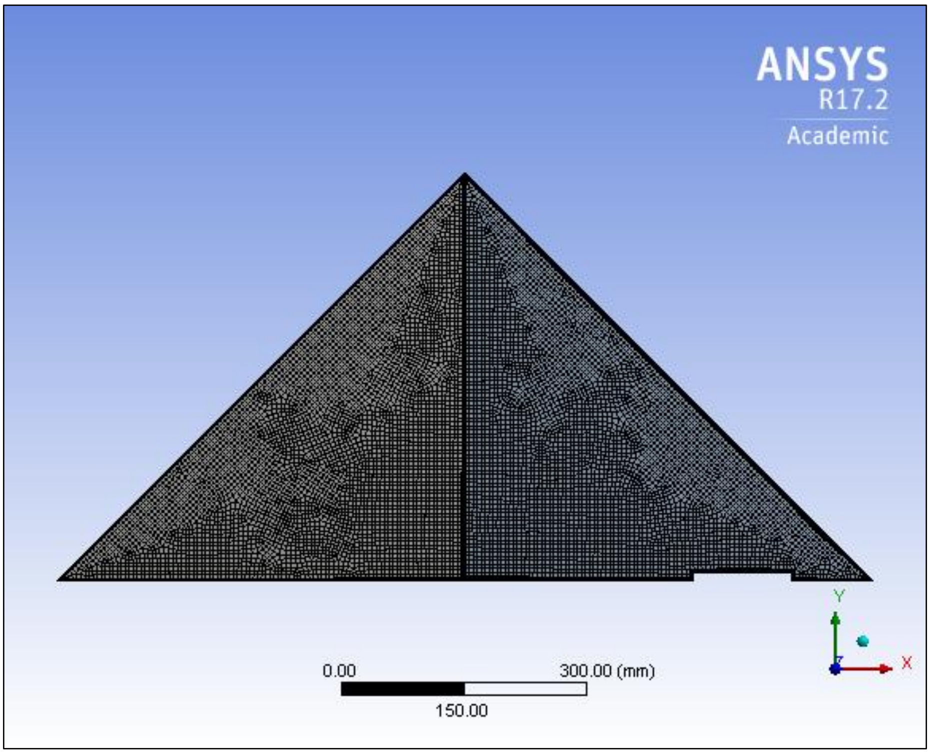

Fig. 3: *Meshing of Enclosure.*

MATHEMATICAL FOUNDATION

The numerical model for fluid flow and heat transfer within the square channel was developed below the subsequent assumptions:

1. Steady two-dimensional fluid flow and heat transfer.
2. Negligible radiation heat transfer.

The whole domain is discretized using fine mesh generation on the geometry. Compressed air and superheated steam is used as convecting medium with ambient conditions. Pressure inlet and temperature inlet boundary condition is applied on the inlet of the test section and mass flow rate is applied on the outlet of the test section. Convection takes place on the solid liquid interface. Analysis is done in ANSYS 16 and thermal analysis is carried out.

GOVERNING EQUATION WITH SOLUTION METHOD

Natural convection of air within triangular enclosures is considered. Governing equation for laminar regime in 2D assuming unsteady state, incompressible fluid with Boussinesq approximation for coupling the temperature fields in the flow field are:

$$\frac{du}{dt}+u\frac{du}{dx}+v\frac{du}{dy}= -\frac{1}{p}\frac{dp}{dx}+v(\frac{d^2u}{dx^2}+\frac{d^2u}{dy^2})$$

$$\frac{dv}{dt}+u\frac{dv}{dx}+v\frac{dv}{dx}=-\frac{1}{p}\frac{dp}{dy}+v\left(\frac{d^2v}{dx^2}+\frac{d^2v}{dy^2}\right)+ g\beta(T-T0)$$

Conservation Law of Energy

Derived from the First Law of Thermodynamics, the energy equation states that the rate of change of energy within a control volume with respect to time must equal the net rate of heat addition to the fluid within the control volume plus the net rate of work done by surface forces on the fluid. Applying this law to a 3D control volume and using Fourier's Law of Heat Conduction the 3D energy conservation equation is:

$$\frac{dT}{dt}+u\frac{dT}{dx}+v\frac{dT}{dy}=\alpha(\frac{d^2T}{dx^2}+\frac{d^2T}{dy^2})$$

Natural convection - Boussinesq model

$$(\rho-\rho_0)g=-\rho_0\beta(T-T_0)g$$

where, T and T_0 are temperature of fluid and reference temperature, respectively, x and y are main Cartesian coordinate variables, u, v, t, p, g, β, α, ρ, and υ are velocity component in x direction, velocity component in y direction, time, pressure of fluid, earth acceleration, density of fluid, thermal expansion coefficient and thermal diffusivity, respectively. The employed non-dimensional variables are given as:

$$X=\frac{x}{H} \quad Y=\frac{y}{H} \quad U=\frac{uH}{\alpha} \quad V=\frac{vH}{\alpha} \quad \vartheta=\frac{T-Tc}{Th-Tc}$$

The governing equations area unit determined exploitation the business software system FLUENT 16.2 exploitation its double exactitude pressure based mostly separate convergent thinker. The second order upwind theme is applied for convective term discretization and straightforward algorithmic rule is finished for pressure and speed coupling. Orthogonal structured cells area unit is done to mesh the domain permanently accuracy. Mesh refinement is formed near the heater, walls and also the vents to resolve the expected high gradients. Comparison of most non-dimensional temperature given during this work with those reported in [3]. The most deviation of the results was below a hundred and twenty fifth and therefore, additional studies were applied with the given boundary conditions (Table 1).

Boundary Conditions

Grid sensitivity and validation of results, numerical experiments performed mistreatment completely, different mesh sizes starting from 12000 up to 60000 triangular components. Its deduced that the best process mesh that contributed grid freelance solutions composed of fifteen, 676 triangular components. Symmetrical meshes with relation to the middle plane of the triangular enclosure were distributed. Grid density was multiplied in sensitive regions like close to the walls and bottom heater. Grid independence for a few quantities is obtained inside one. 4% at the best Rayleigh variety, Ra = 10^6.

Table 1: *Boundary Conditions.*

Temperature of hot wall	400 K
Temperature of cold wall	300 K
Temperature of bottom wall	400 K
Heat Flux from the heater	400 W/m^2
Initial Temperature of Fluid (air)	300 K

Validating numerical results, we've assumed an entire adiabatic thermal stipulation for bottom wall, whereas the proper and left walls square measure differentially heated. Considering a triangular enclosure of AR = 1 with no middle partition with Gr = 2.1577 × 10^7 and Pr = 0.71, we tend to compare numerical findings of native letter for cold inclined wall that shows a satisfying agreement between this results and also the previous experimental study. Next components of this paper contend with transient flow development, steady state results and warmth transfer phenomena of the natural convection among triangular enclosure with submissively heated inclined walls and bottom heat supply with extremely semi conductive partition put in at the centre of the region.

RESULTS AND DISCUSSION

Transient flow development: In this half transient flow development together with dimensionless temperature statistic at numerous points and development of temperature and stream perform contours area unit conferred. Three completely different stages area unit were seen throughout the temperature variation. First, suddenly rises when applying differential heating of aspect walls and bottom heat supply wherever formation of thermal physical phenomenon is dominated by physical phenomenon heat transfer mechanism. Its seen that less time is required for higher Ra to succeed in the steady stage and price of dimensionless temperature is lower for higher Ra. This issue is attributed to the thickness of thermal physical phenomenon shaped on top of all-time low heat supply that is a smaller amount thicker (creating beguiler temperature gradient) for Ra = 10^6 with relation to Ra = 10^4. Moreover, unforeseen respectful heating of inclined walls ends up in formation of a chilly thermal physical phenomenon adjacent to right wall.

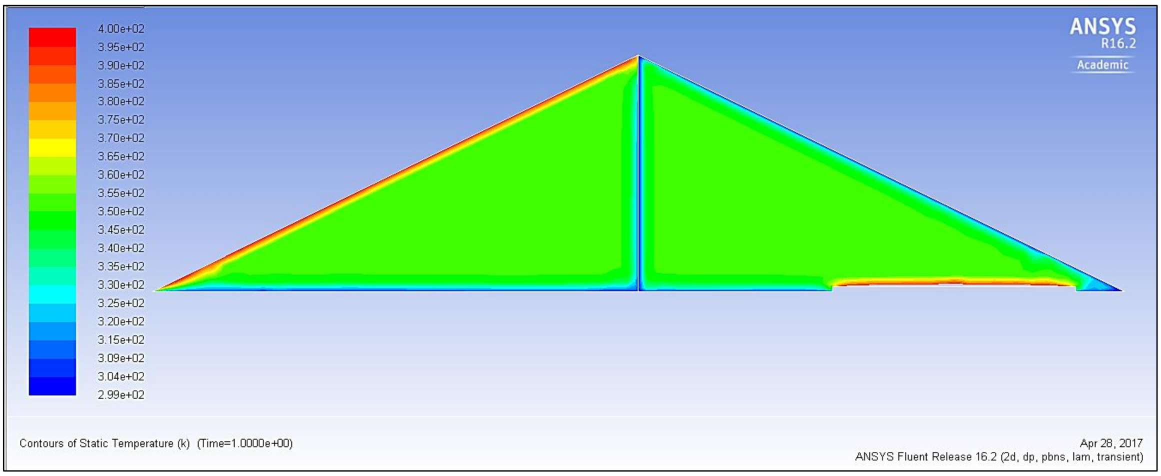

Fig. 4: *Countours of Static Temperature at Time = (1.0000e+00).*

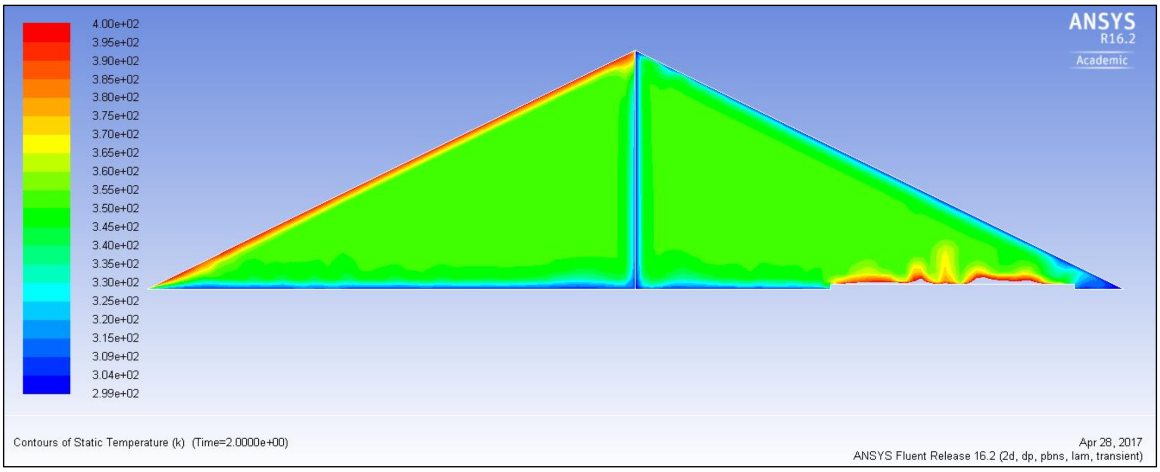

Fig. 5: *Countours of Static Temperature at Time = (2.0000e+00).*

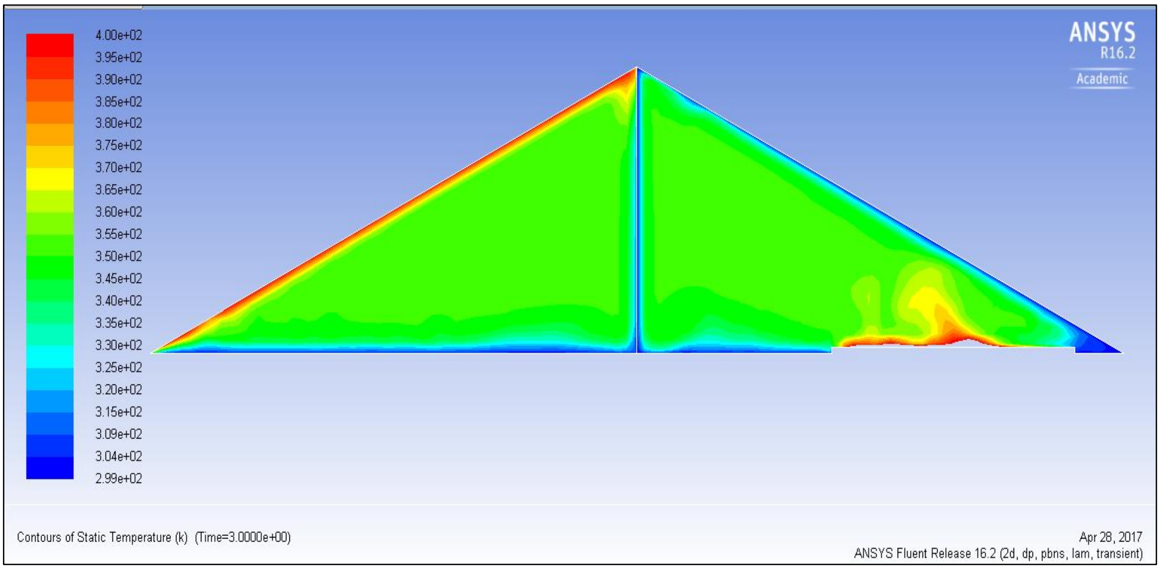

Fig. 6: *Countours of Static Temperature at Time = (3.0000e+00).*

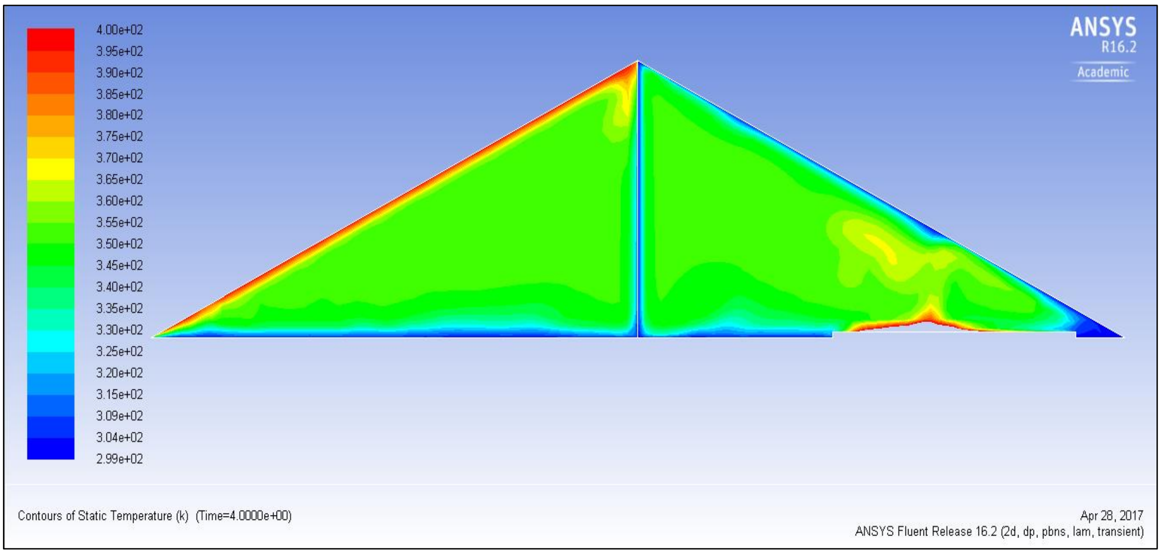

Fig. 7: *Countours of Static Temperature at Time = (4.0000e+00).*

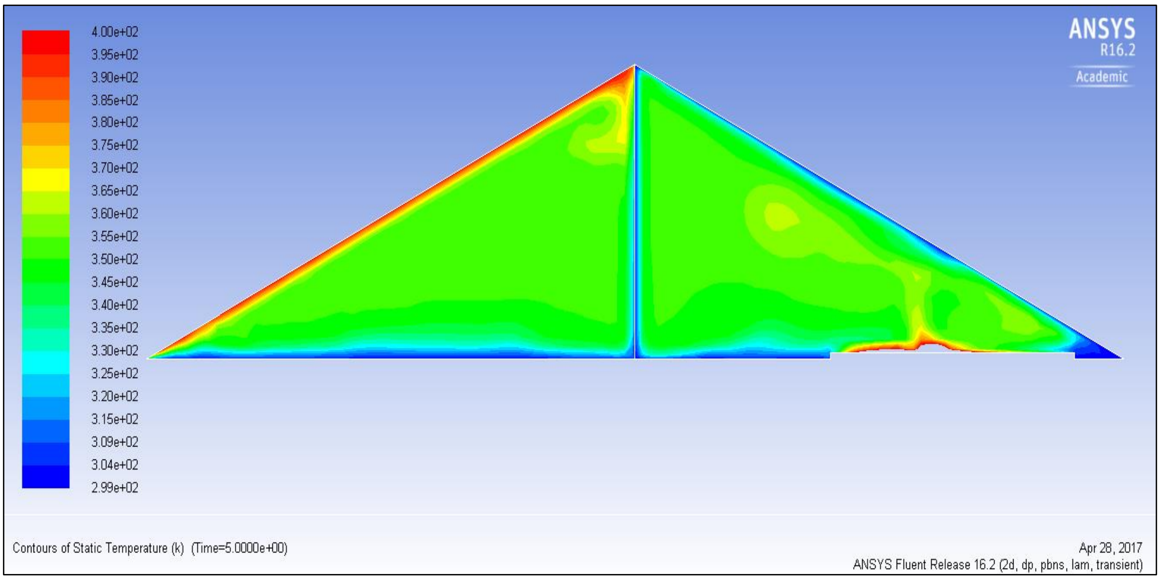

Fig. 8: *Countours of Static Temperature at Time = (5.0000e+00).*

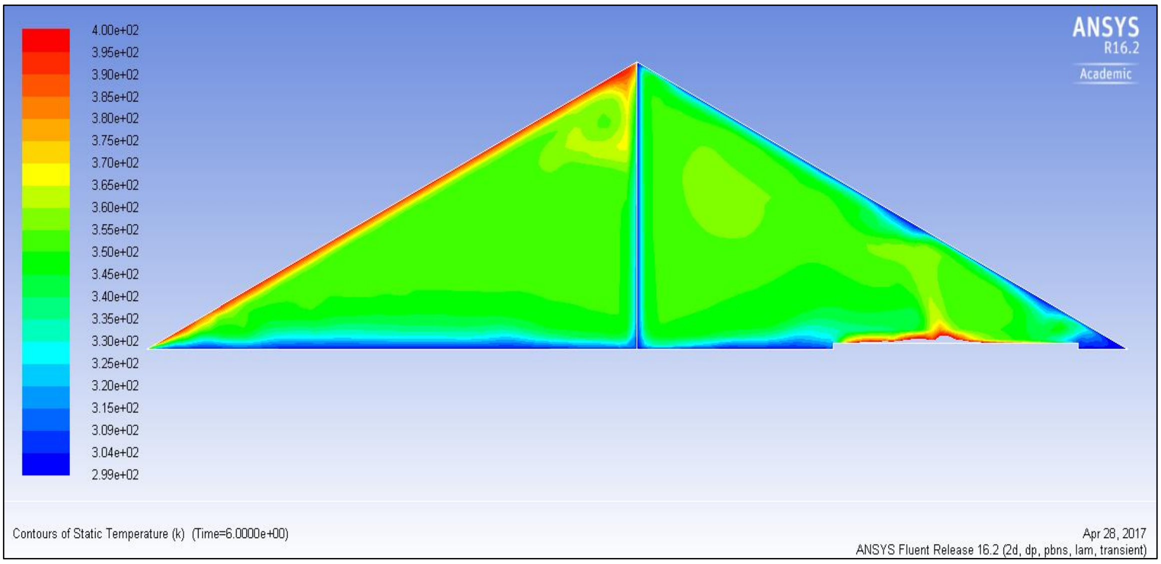

Fig. 9: *Countours of Static Temperature at Time = (6.0000e+00).*

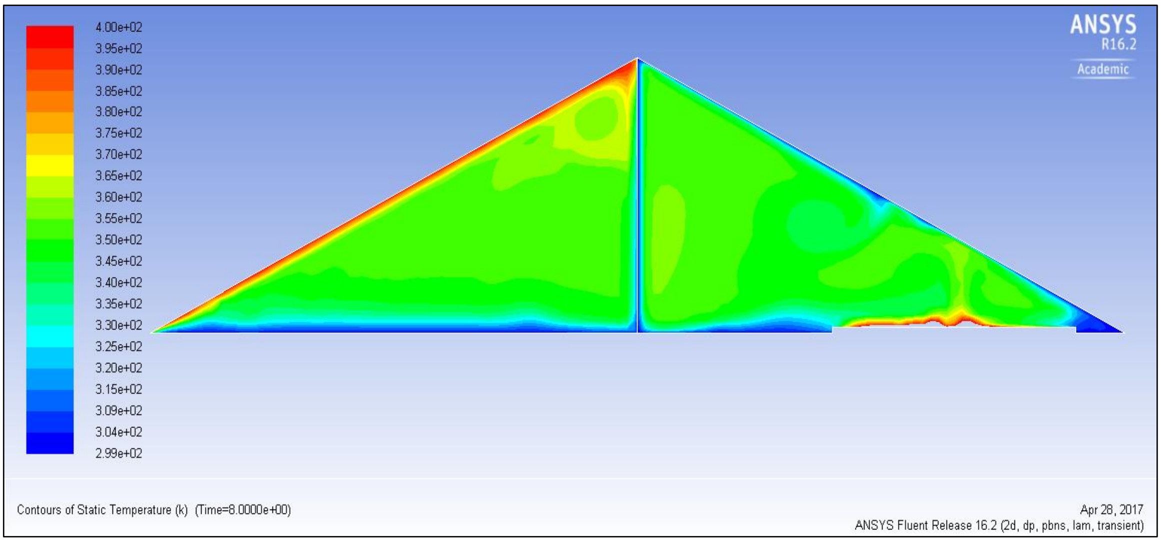

Fig. 10: *Countours of Static Temperature at Time = (8.0000e+00).*

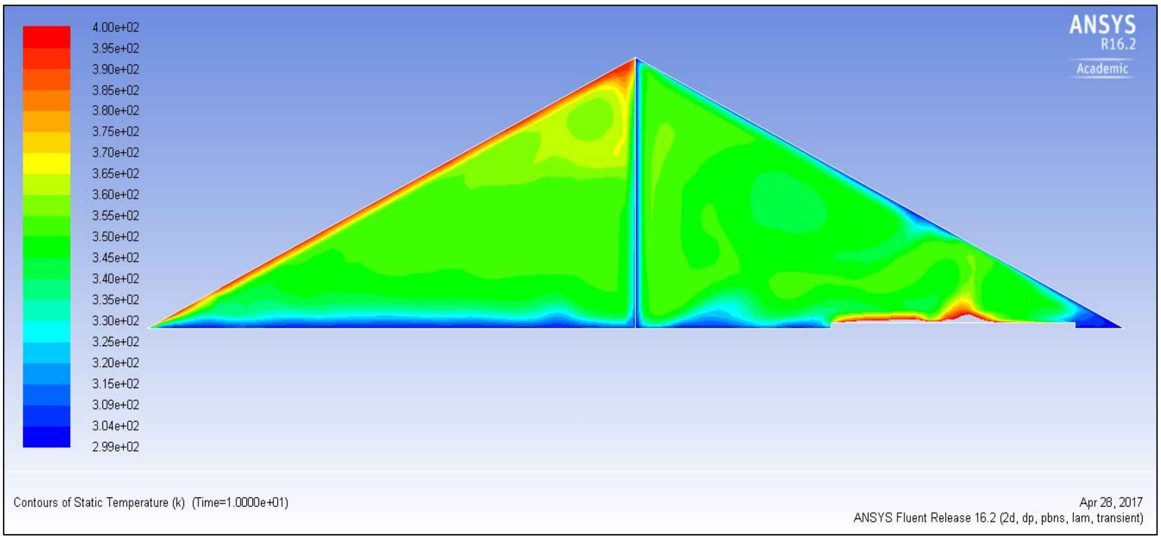

Fig. 11: *Countours of Static Temperature at Time = (1.0000e+01).*

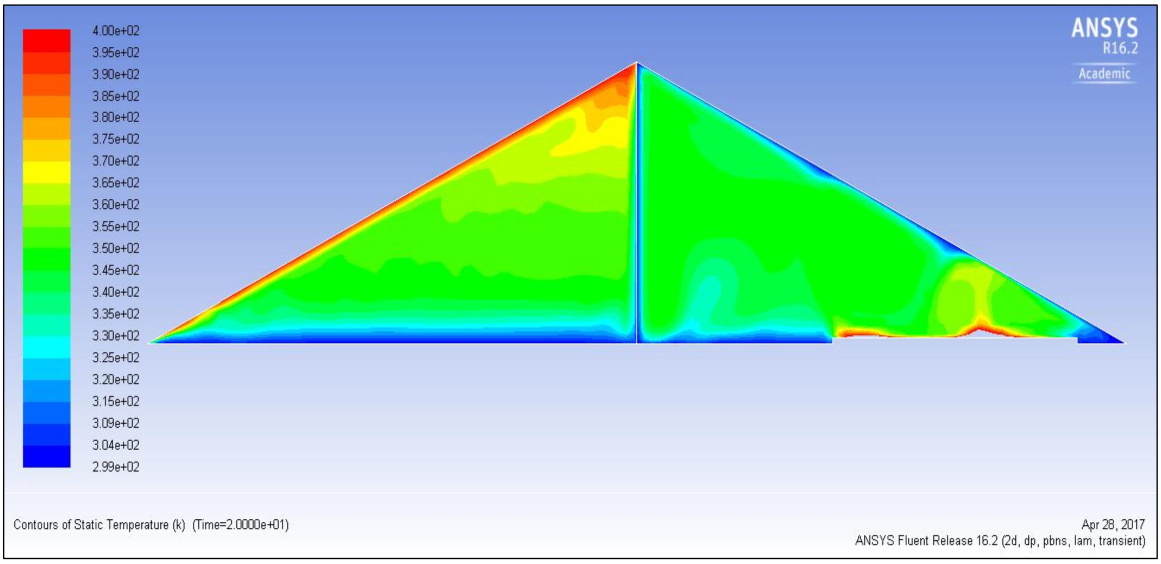

Fig. 12: *Countours of Static Temperature at Time = (2.0000e+01).*

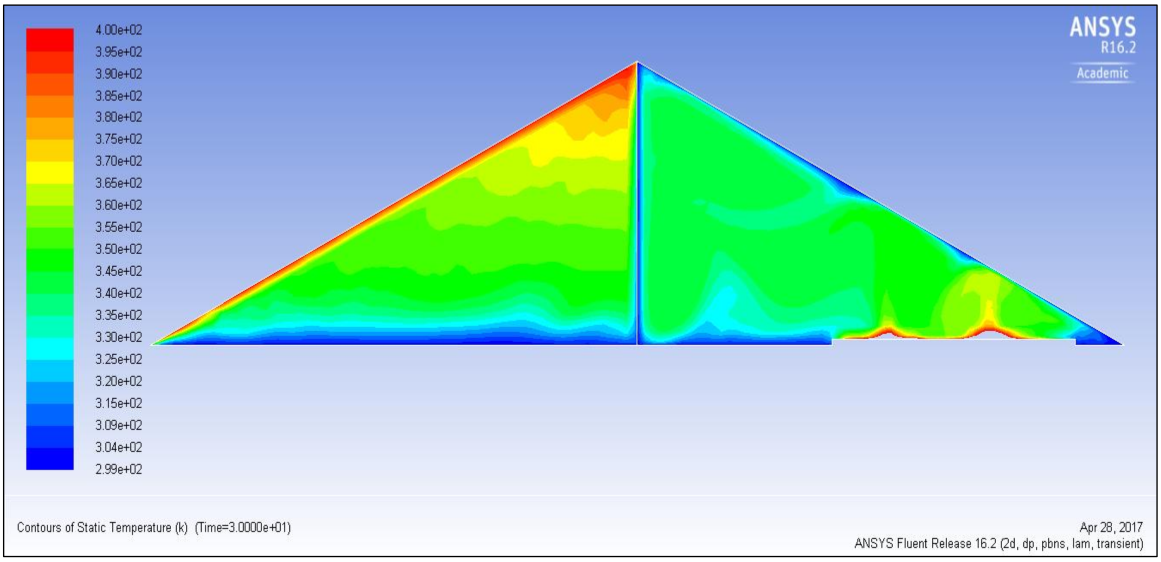

Fig. 13: *Countours of Static Temperature at Time = (3.0000e+01).*

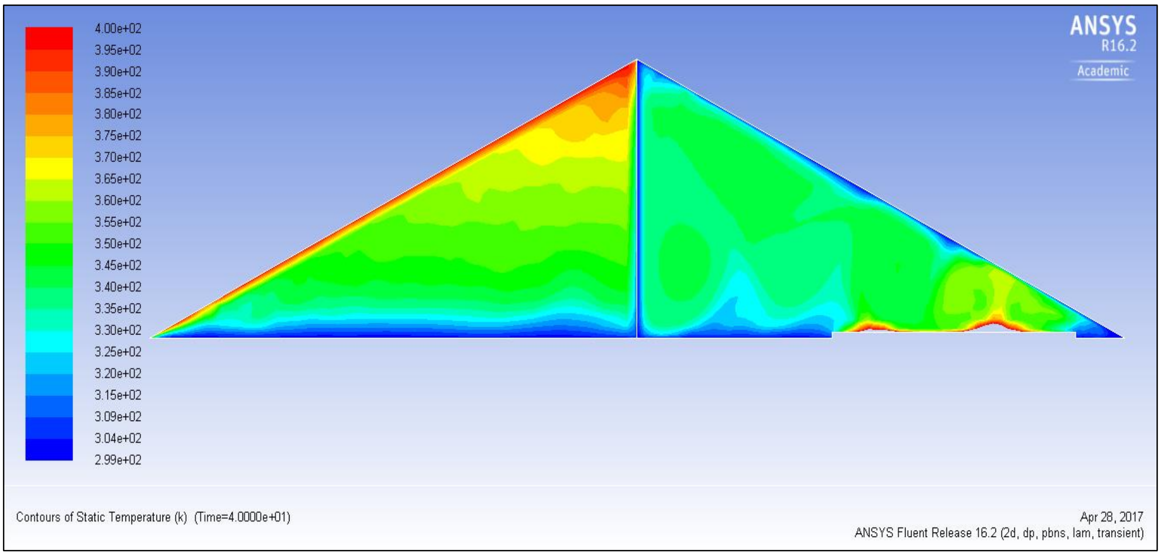

Fig. 14: *Countours of Static Temperature at Time = (4.0000e+01).*

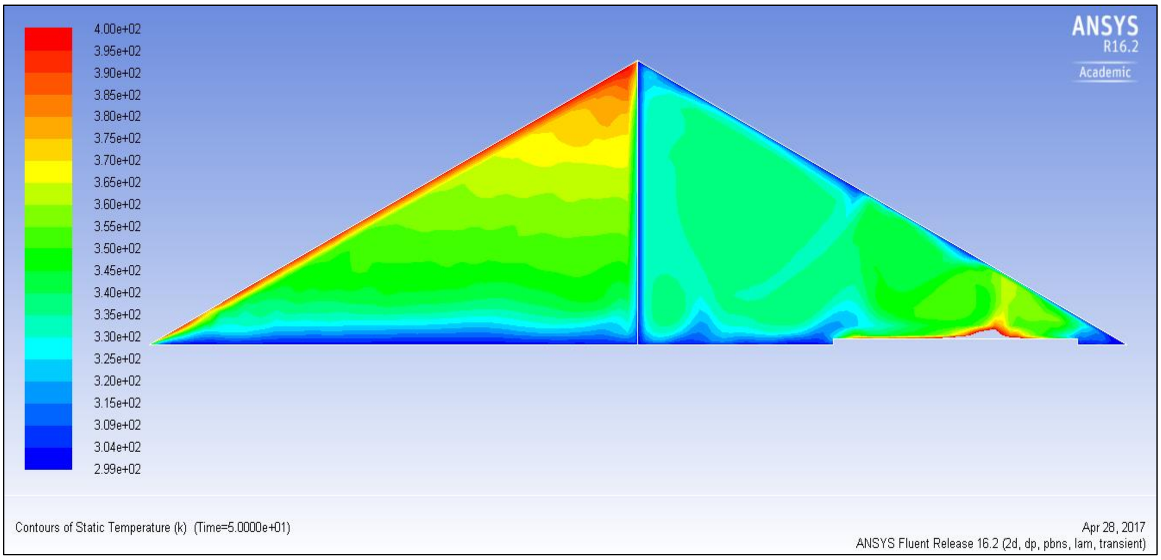

Fig. 15: *Countours of Static Temperature at Time = (5.0000e+01).*

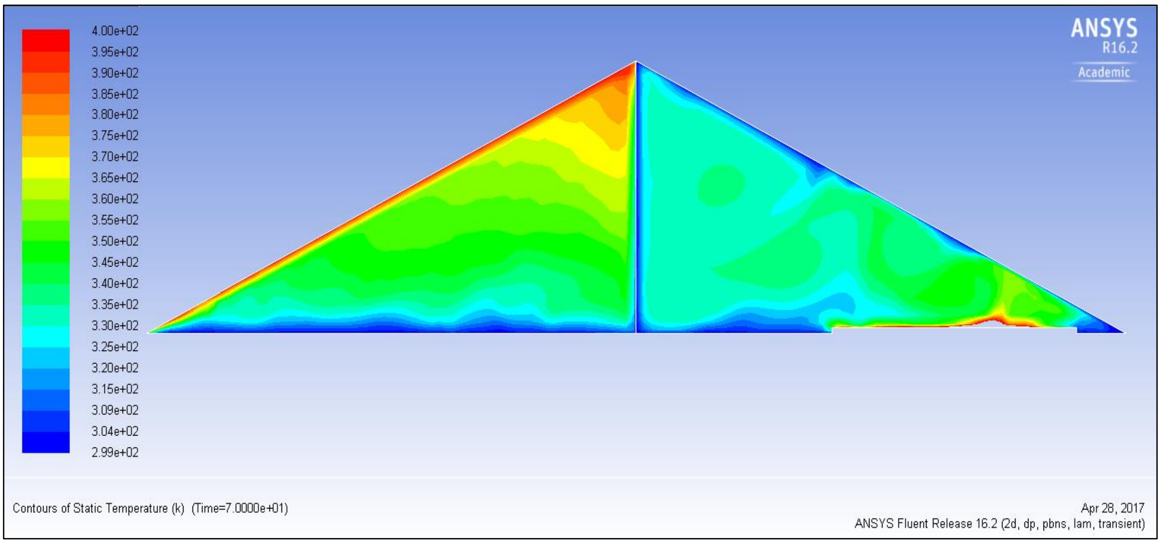

Fig. 16: *Countours of Static Temperature at Time = (7.0000e+01).*

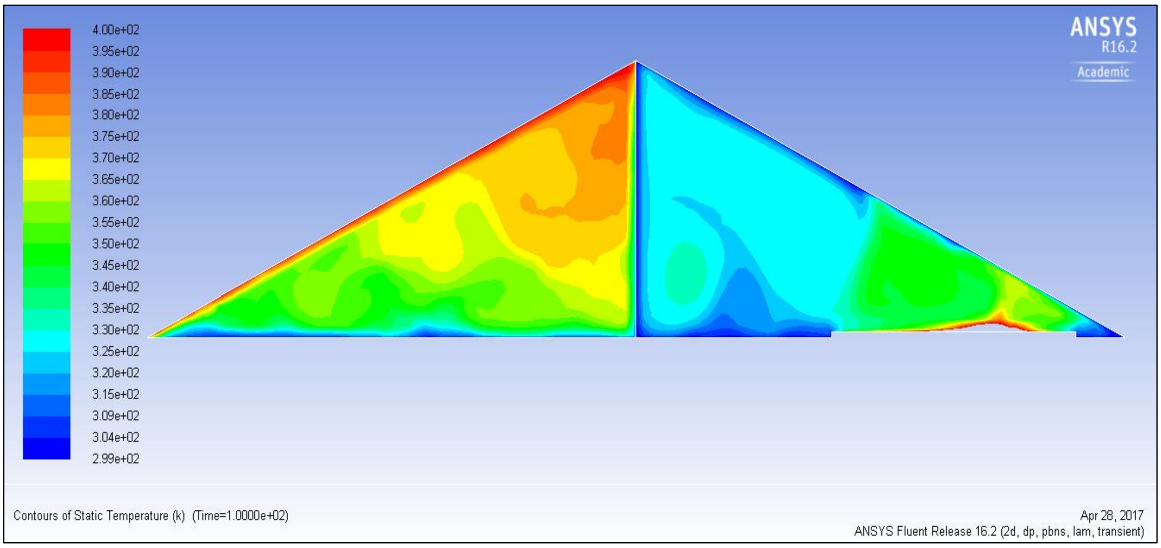

Fig. 17: *Countours of Static Temperature at Time = (1.0000e+02).*

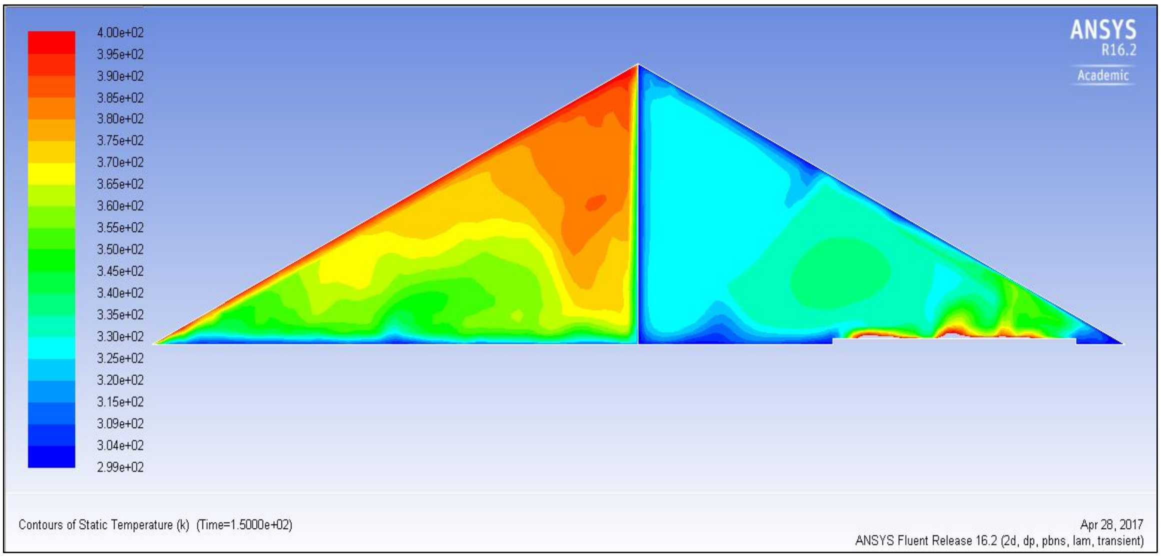

Fig. 18: *Countours of Static Temperature at Time = (1.5000e+02).*

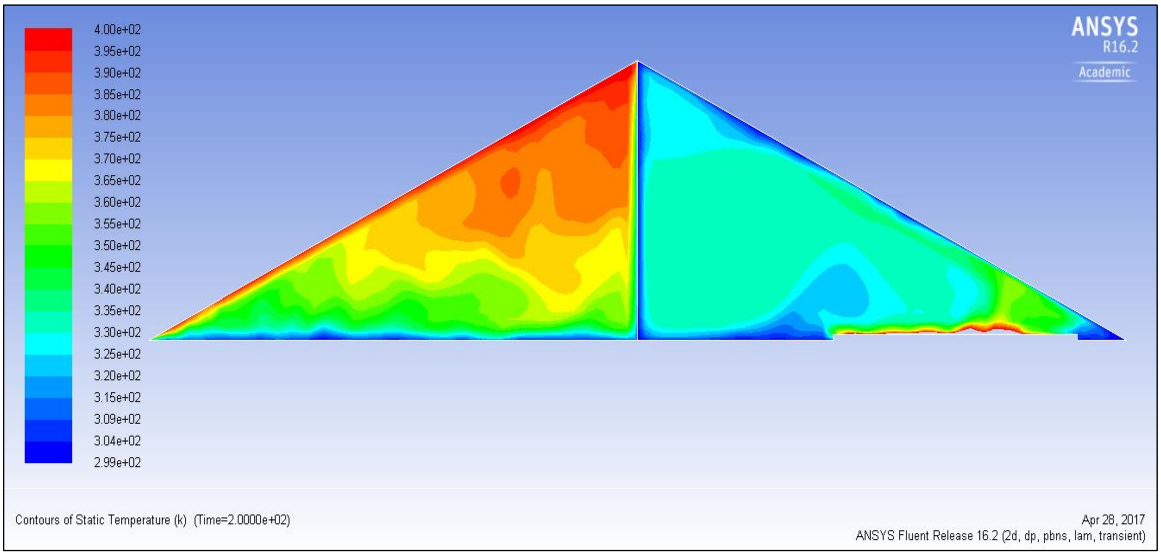

Fig. 19: *Countours of Static Temperature at Time = (2.0000e+02).*

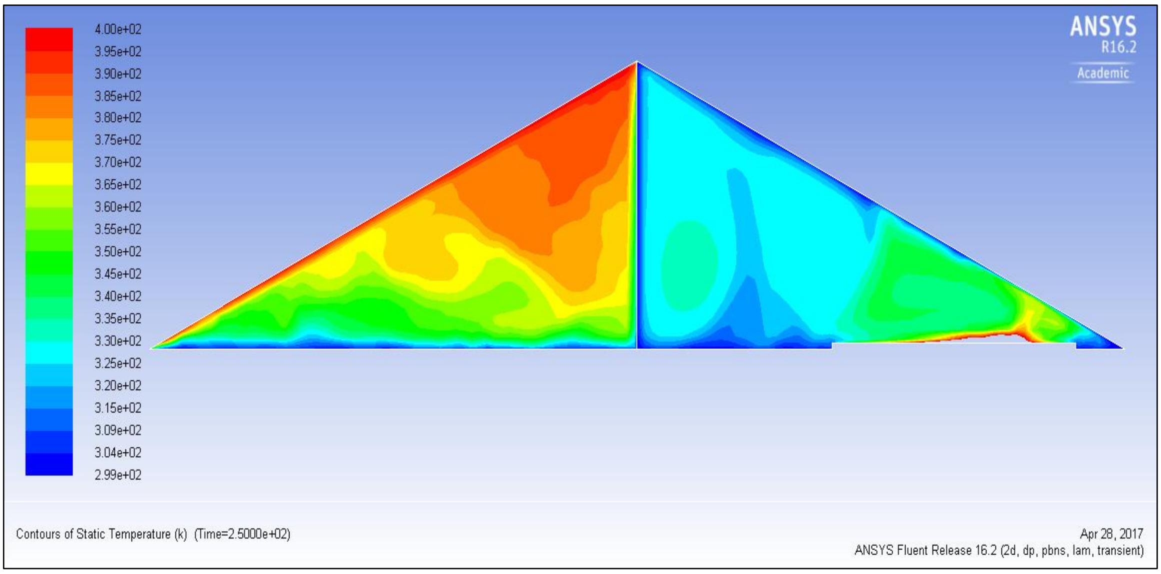

Fig. 20: *Countours of Static Temperature at Time = (2.5000e+02).*

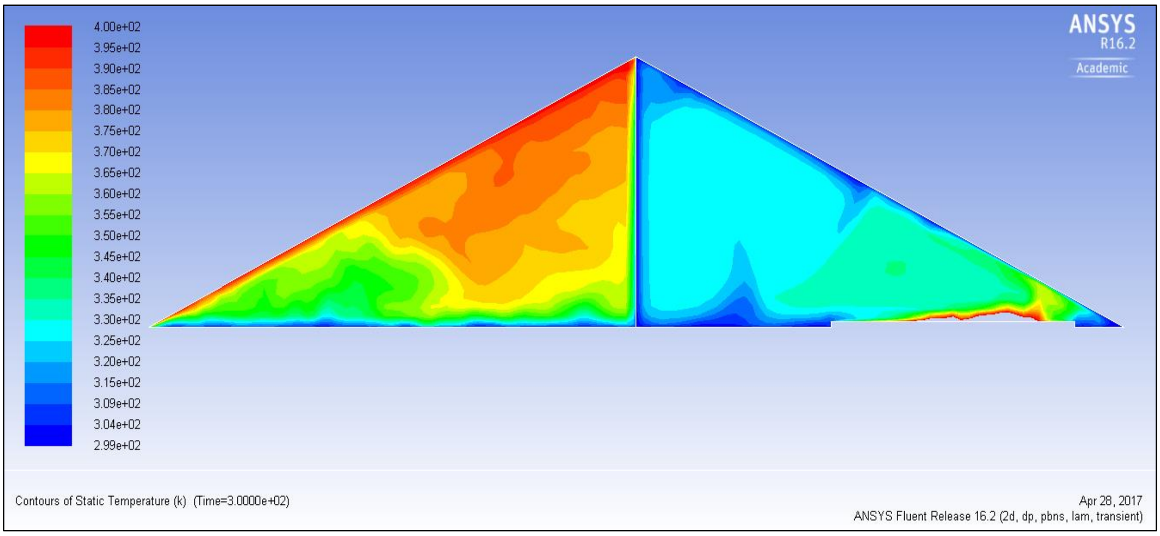

Fig. 21: *Countours of Static Temperature at Time = (3.0000e+02).*

For higher Ra, left zone of the enclosure can miss higher quantity of thermal energy. This truth will be understood by subtracting the values of steady dimensionless temperatures of left and right sides of the partition for varied Ra. Its conjointly seen that less time takes for higher Ra to achieve steady stage and this could be associated with quicker heat transfer of two coupled thermal systems (left and right zones of the enclosure).

Above contour shows temperature and stream operate contours for various dimensionless time once Ra = one hundred and five, AR = 0.5, L/W= 0.5 and S/W= 0.4 (Figures 4-21). Its ascertained that for early stages, two elongated cells *AR* shaped on the aspect walls. After t= 0.01, the cell occupies the whole left zone and also the single massive cell is remained for the steady state condition. Except for the proper zone, an extra tiny cell is created to a lower place the massive elongated one. Once t = 0.02, this tiny cell rises up on the conductive partition. For t = 0.04, two cells stay together forming one massive cell at the proper zone occupying the whole right domain. Except for the steady state condition, it's seen that the massive cell is split into three smaller vortices making cellular flow pattern. This is often attributed to the dominancy of natural convection heat transfer mechanism created by thermal energy transferred to the proper aspect, bottom heat supply and right cold incline wall. It’s evident that once t = zero.005, thermal physical phenomenon’s adjacent to the aspect heated walls get thicker wherever the cold raging boundary layer is traveling on the proper wall toward rock bottom edge. At t = 0.02, hot fluid intrusion into core of the proper cavity is seen from rock bottom heater. Once the cold raging layer diffuses to besides bottom heater (t = zero.04), the mentioned hot fluid intrusion is separated from the heater arising to the core of the cavity. The separated hot fluid is spread inside the proper triangle. Finally, it's seen that a hot fluid intrusion from bottom heat supply to the proper cold wall is created for steady state and at the same time a chilly fluid intrusion to core of the proper triangle is established.

Figure 22; exhibits comparison of mean Nu of bottom heater at steady state for partitioned enclosure. Above figure 22 relates mean Nu for different AR at Ra = 10^5, S/W = 0.5 and L/W = 0.4 and it is clear that for both cases, similar pattern is reportable and almost equal values for mean Nu is seen.

From Figure 23 we conclude that, with increase of Ra, mean Nu of bottom heat supply is augmented and there is no considerable difference of mean Nu of bottom heat source is seen between partitioned enclosures.

It is evident from above Figure 24 that higher values of mean Nu are seen for divided enclosure with the rise of bottom heat supply size. This is often as a result of the actual fact that for divided cavity, bottom heat supply just has thermal communication with cold incline wall and existence of a conductive partition reduces the thermal affiliation of right and left enclosures.

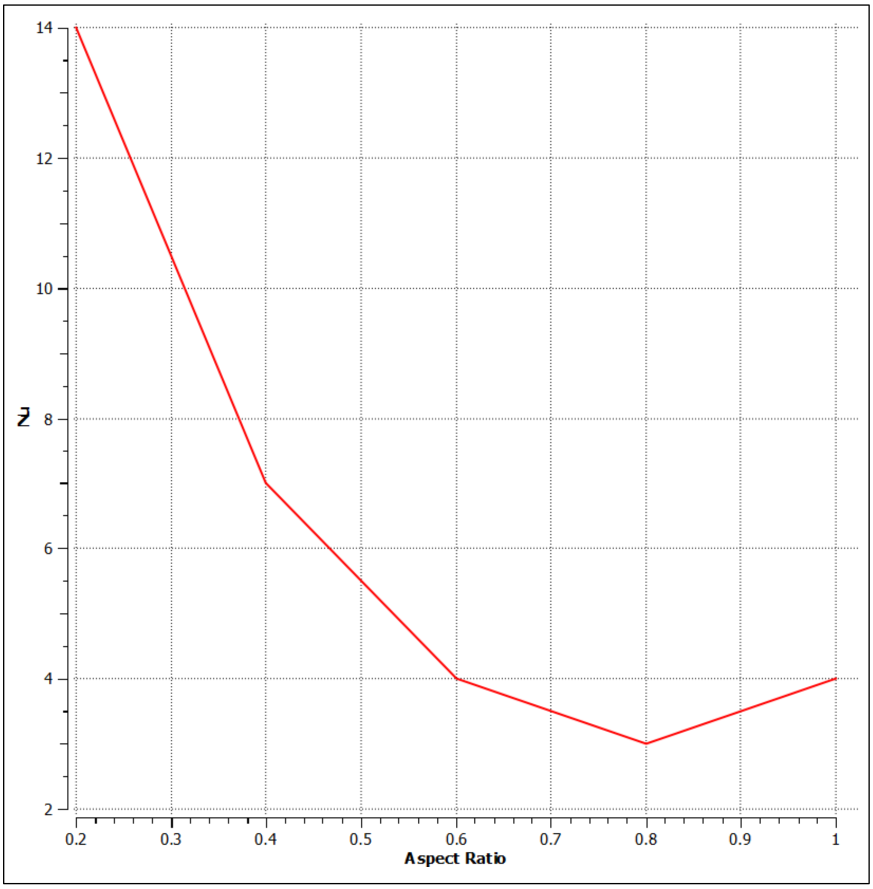

Fig. 22: *Comparison of Mean Nu of Bottom Heater at Steady State for Partitioned Enclosure.*

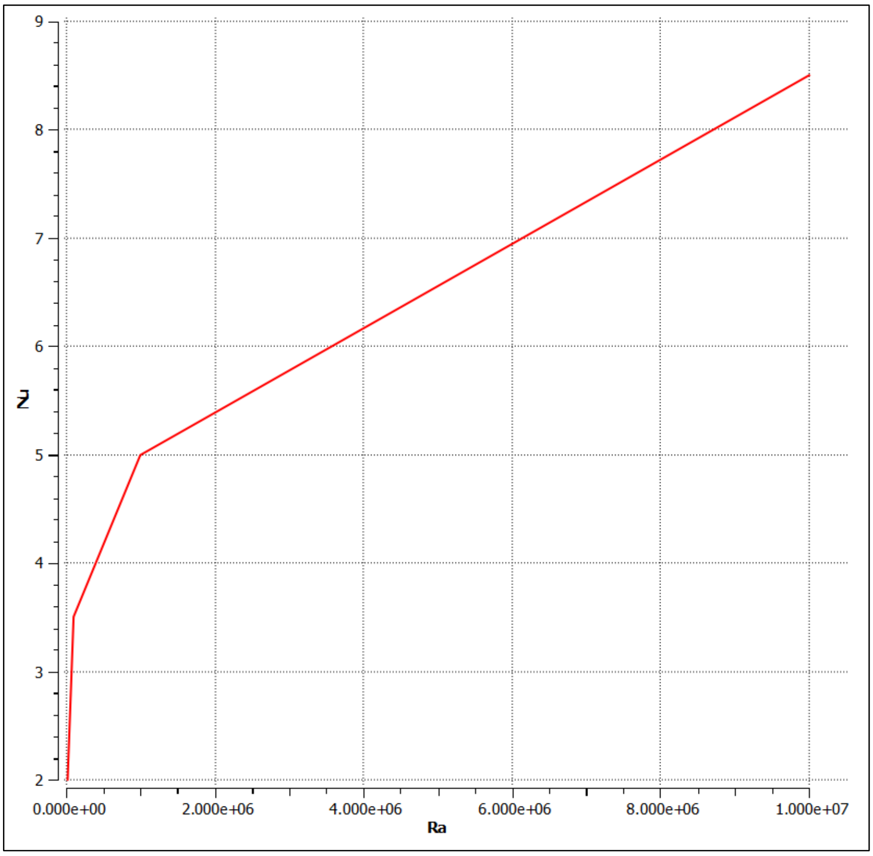

Fig. 23: *Variation of Nu with respect to Ra for partitioned enclosure.*

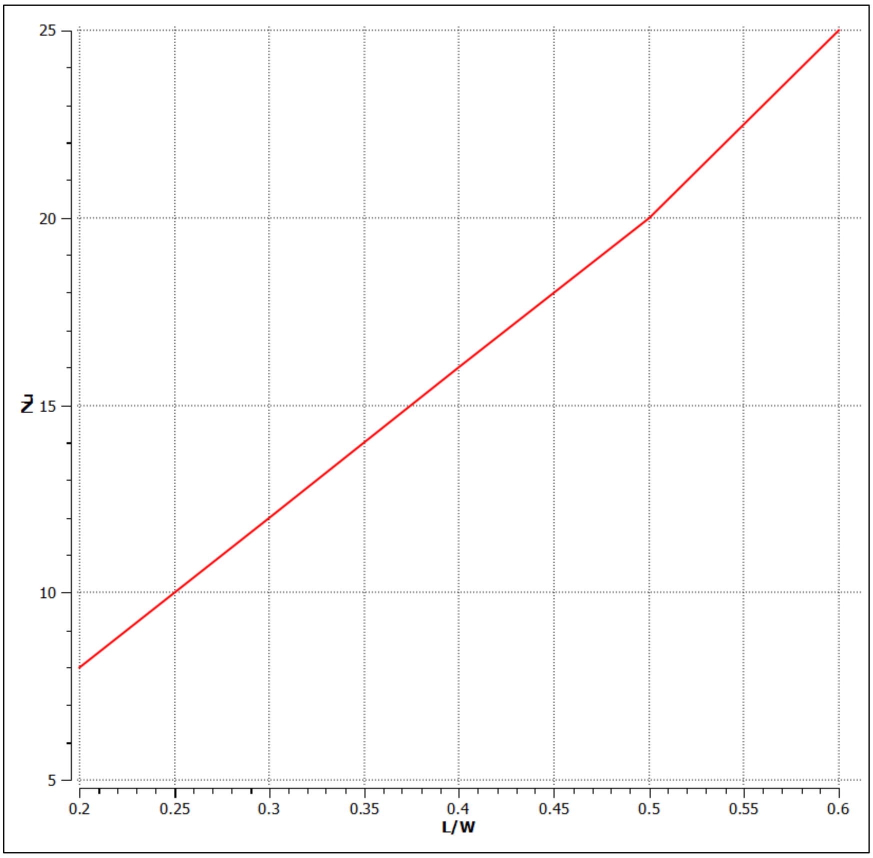

Fig. 24: *Variation of Mean Nu of Bottom Heater at Steady State.*

CONCLUSIONS

Numerical investigations were conceded out to study the thermal behaviour and flow field of naturally air convection in the interior a partitioned tri-angular cavity of differentially heated inclined walls with a heat source at bottom wall. Significant dependency of several flow parameters on fluid flow and heat transfer is studied including Ra from 10^3 to 10^6, heater size, L/W from 0.2 to 0.6, heater position, S/W from 0.3 to 0.7 and aspect ratio, AR from 0.2 to 1.0 with a fixed Prandtl number of 0.71. A conductive panel place at the middle of cavity and links the right and left part of the cavity.

It can be seen that the variation of Ra, AR, S/W and L/W has marginal effect on left wall and after a certain period of time is passed mean nu of left wall decreases.

The number of vortices start decreases with increase of Ra and only a large vortices is formed towards right cavity with AR=1.

The thickness of hot thermal boundary layer increases with increase in the value of Ra.

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Cite this Article
Akshendra Soni, Ajay Kumar, Mukesh Prakash Mishra,,Investigation of free convection heat transfers in an attic shaped enclosure. *Journal of Refrigeration, Air Conditioning, Heating and Ventilation*. 2017; 4(2): 13–26p.

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Reference:BIET-ME/GC/2017-1780

DATE: 18/12/17

To Whom It May Concern

This is to certify that **Mr. Akshendra Soni** S/O Mr. Sachchida Nand Soni has worked in this institution as Guest Lecturer in Department of Mechanical Engineering for Undergraduate and Postgraduate programme from July 2016 to June 2017.

I found him a devoted and sincere faculty who performed his duties efficiently. He bears a good moral character. I wish him best of luck for his bright future.

(Dr.N. P. Yadav)

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NO: REC/DIR/2018-91

Dated: 19/01/2018

To Whom It May Concern

It is to certify that Er. Akshendra Soni, S/o Shri Sachchida Nand Soni worked as Guest Faculty in the Department of Mechanical Engineering of this institute w.e.f. 01 August, 2017 to 8 December 2017.

He offered Heat and Mass Transfer, Element of Mechanical Engineering to B. Tech. students as a part of curriculum.

Er. Akshendra Soni is a hard working and sincere candidate committed to the academics.

I wish him all the best for future endeavors.

D Yadav

(Prof. D. S .Yadav)
Director