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Effect of Different Geometry and Inclination Angle on Heat Transfer in Natural Convection

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ABSTRACT

In this we will comparing the geometries (cylindrical & rectangular) for different inclination angle with same hydraulic diameter. Natural convection is a, heat transport mechanism, in which the fluid motion is by naturally means no external source is required. temperature gradient is occurring due to density difference. Forced convection is a heat transport mechanism in which fluid motion is by an externally .It is one of the most important heat transfer mechanism in which maximum amounts of heat energy can be transferd very efficiently. We compare by calculating average heat transfer coefficient and Nusselt number for both geometries, for inclination angle (i.e. 0, 15^0, 30^0, 45^0, 60^0, 75^0, 90^0). It is concluded that average Nusselt number, for angles (15^0, 30^0, 60^0, 75^0, 90^0) shows maximum heat rate for cylindrical test section and for angle (45^0) shows maximum heat rate for square test section and also we found that heat transfer coefficient rate increases with inclination angle in both cases.

Keywords: Convection, Geometry, Inclination angle

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I. INTRODUCTION

Many studies have been conducted on natural convection in enclosures, as represented by the Rayleigh–Bénard convection. It has a wide variety of applications including solar collectors, heat storage, and cooling of electrical devices. Among the studies on natural convection in an annulus, which is one of many enclosure configurations, there have been a numerical investigation of natural convection in eccentric horizontal cylindrical annuli by Hirose and colleagues . Although it belongs in the same category, however, for natural convection heat transfer between concentric rectangular pipes, there seems to be no previous study except a preliminary one. This type of natural convection heat transfer appears, for example, in cooling of transformers. An understanding of its characteristic features will help improve the cooling performance.[1]

Natural convection is a heat transport process, in which the fluid motion is generated by naturally, only due to density differences in the fluid and density differences are occur due to temperature gradients[4]

In order to enhance natural convection heat transfer, it is required to improve the performance of heat transfer paying full attention to compactness and the light weight of heat transfer devices and inevitably to increase the heat transfer coefficient itself[8]

Forced convection is heat transfer process in which fluid motion occur due to external devices. It is one of the important methods of heat transfer as maximum energy can be transported very easily. Consider a hot object kept in cold air. The temperature of the outside surfacewill drop as heat transfer with cold air, and temperature of adjacent air the object will increase. And, the object is surrounded with warm layer of airand then heat transfer from thislayer to the outer layer The temperature layer adjacent to the hot object is greater , and it lowers the density. So heated air rises upward. That process of movement of air is called the natural convection current. Anddue to absence of this process, heat would be transfer by only conduction and its rate will be much lower.

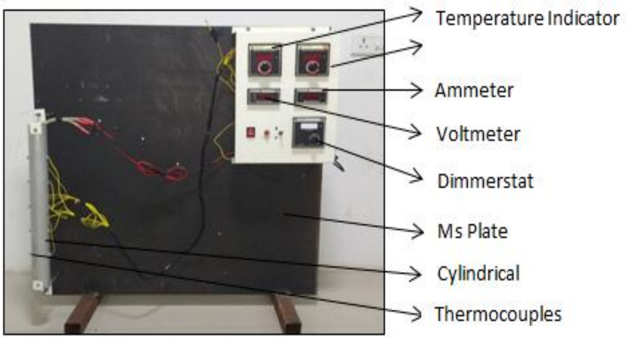
In gravitational field, a net force which pushes a light fluidwhich placed heavier fluid upward. This is called the buoyancy force. The magnitude of buoyancy force is the weight of the fluid displaced by the body Cool air.The control of the heat inflow from a wall is an important topic for energy saving. Heat transfer between a fluid and a solid wall is mainly carried out by convection and radiation. From the natural heat convection perspective, general use is that a thick and good heat insulating material plate is used to cover a heating surface to inhibit the inflow or emission of heat from the heating surface. It is impossible to reduce the heat transfer resistance, though the thermal conductivity resistance increases. Furthermore, in the case of using a heat insulating material including high porosity materials such as glass wool, heat transfer becomes active because natural convection occurs under the condition of a large temperature difference between the wall and ambient.[7]

2. Literature Review

Yuichi Funawatashiet. Al [1] studied a natural convection in which heat transfer between concentric parallel pipes for low Rayleigh number Ra (L 3500) with aspect to ratio of inner parallel pipe of 2.0, 4.0,
6.0, and 8.0. In this flow pattern for high Rayleigh number in a space over inner parallel pipe are ring or rectangular role. And as a aspect ratio increases number of role. The flow pattern which is oblong in circulation in side space , which extends towards the bottom space. The local Nusselt number distribution at the top of the surface of the inner parallele pipe has peak at the stagnation points. The theNusselt and Rayleigh numbers relation at the top surface is similar to that of the Rayleigh–Bernard convection obtained, and on the side and bottom of surface and the Nusselt number increase with Rayleigh number.
Yuichi Funawatashiet. Al [2] conducted study on natural convection inan enclosures, as represented by the Rayleigh–Bénard convection. It has so many applications including solar collectors, heat storage, and cooling of electrical devices.
M. Al-Arab et al [3] investigations available on heat transfer on vertical plate by natural convection. Only a limited number of investigations are, however, available on inclined plates and in only some of them were the investigation extended into the turbulent region. The plates used in most cases were of ‘finite width’ and the results suffered from the presence of side-edge effects. The present work is the result of experiments carried out to investigate local and average natural convection heat transfer from isothermal, vertical and inclined plate which facing upwards air in both laminar and the turbulent region.
Kimura [4] reported on a differentially heated partial sector-shaped enclosure. As a series of our studies to discuss the effects on the enclosure shape on natural convection, a previous paper focused on natural convection in a vertical and inclined semicircular enclosure heated differentially, where flat surface was heated and radial surface was cooled.
Yue-Tzu Yang et al [5] The present investigation shows that study of convective heat transfer from a horizontal circular cylinder under the effect of a solid plane wall. The full Navier-Stokes and energy equations for two-dimensional steady flow are solved by finite element method. It present the Variations in surface shear stress, local pressure and Nusselt number along the surface of cylinder as well as predictedaverage Nusselt number values, also shows the location of separation and some flow and temperature fields are presented.
Wei Qiet. al[6]Experimental investigation shows natural convection heat transfer of air which is in layer in vertical annuli are presented. In these experiment inner cylinder is maintained at a constant heat flux condition and the outer is cooled in the atmosphere. so to obtain the convective contribution, the overall heat transfer data are corrected for thermal radiation and axial conduction losses from the end plates in the annuli

3. Experimental Setup

Experimental setup consist of two test section, one is hollow cylindrical (with dimension, diameter 50mm, thickness 4mm and length 500mm.) and another hollow square with dimension (50×50×500mm) with thickness 4 mm. Material used for both test section are aluminium 19000 with thermal conductivity 210 W/mK, nichrome heater of 1 kW capacity is used for heating the test section, which is placed inside the test section. For cylindrical test section round shaped heater used and for square test section square heater is used. For temperature measuremet



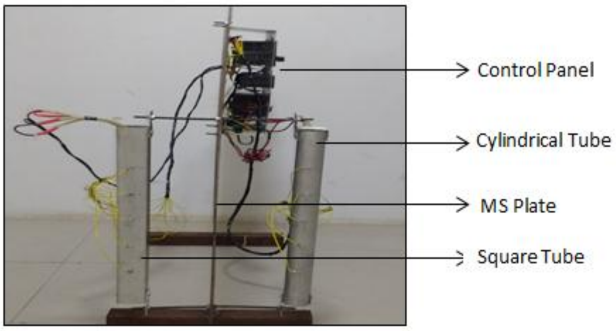


Fig.1 Experimental setup

k type thermocouple are used. Thermocouples are connected by tapping it on outer surface of both geometries. Both test section attached to MS plate shown in fig.1. It’s one end is fixed and another is hinged. They are moving 0 to 90^0.The control panel placed on MS plate and it consists of Ammeter, voltmeter, dimmer stat and two temperature indicators.The heat supplied to heater is measured by Ammeter and voltmeter and it is varied by dimmer stat. Temperature at different sections on test section is indicateed on temperature indicator.

Experimental Data :

Table 1 : Square Rod θ= (60^0)

Time (Min.)	TI	TII	TIII	TIV	TV	YVI	TVII	TVIII	TIX	TX	T∞
30	125	126	126	123	119	119	123	126	128	126	30
40	133	134	133	130	125	125	129	133	135	133	30
50	135	136	135	132	127	127	131	135	137	134	30
60	135	136	134	132	126	126	131	136	138	135	30
70	136	137	135	132	127	128	132	136	138	135	30

Table 2: Cylinder Rod θ= (60^0)

Time (Min.)	TI	TII	TIII	TIV	TV	YVI	TVII	TVIII	TIX	TX	T∞
30	125	126	126	123	119	119	123	126	128	126	30
40	133	134	133	130	125	125	129	133	135	133	30
50	135	136	135	132	127	127	131	135	137	134	30
60	135	136	134	132	126	126	131	136	138	135	30
70	136	137	135	132	127	128	132	136	138	135	30

4. Result and Discussion

4.1 Validation of Experimental Setup

4.1.1 For Cylindrical Test Section

In the beginning, results of the present cylindrical rod are validated with those obtained from the standard empirical correlation of Adam, Churchill and other two relations for heat transfer coefficient given below,

Nusselt number correlation

i. Empirical correlation of Churchill and Chu.

$$Nu = 0.68 + \frac{0.67 R_a^{1/4}}{[1+(\frac{0.492}{P_r})^{9/16}]^{4/9}}$$

$$Nu = \{0.825 + \frac{0.387 R_a^{1/6}}{[1+(0.492/P_r)^{9/16}]^{8/27}}\}^2$$

ii. Empirical correlation of Adam

$$Nu = 0.59 \times R_a^{1/4}$$

iii. Empirical correlation

$$Nu = 0.655 \times R_a^{1/4}$$

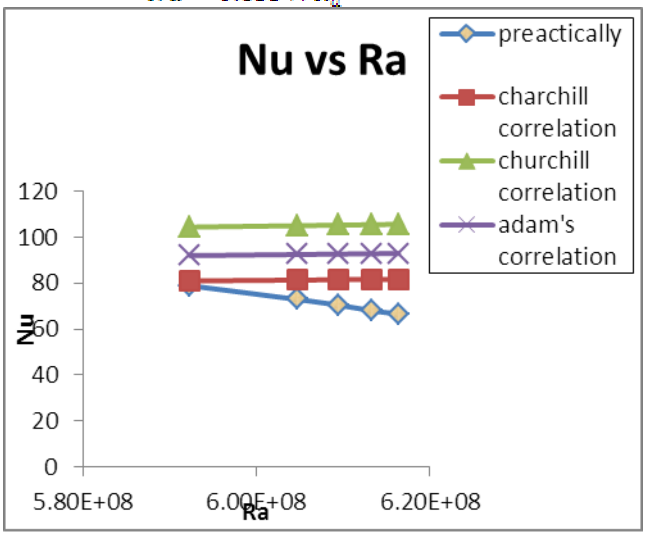


Fig.2 Validation of Cylindrical Test Section

The comparision of Nusselt number and coefficient of heat transfer for present square rod with existing correlation shown in fig.4. In this figure shows that validation experiment for heat transfers in terms of Nusselt number and heat transfer coefficient for cylindrical rod are in good arrangement with result obtained from all empirical correlation. It is found that Nusselt number in the present cylindrical tube agree with those from all empirical correlation within ± 10.470 shown in fig 4.

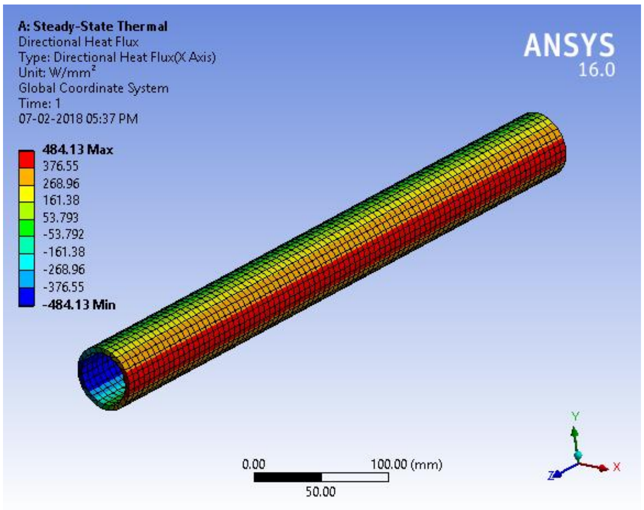


Fig.3 Details of steady state heat flux in cylindrical rod

Fig.3 shows that steady state heat flux throughout the cylindrical rod .maximum heat flux occur in the test section is 484.13 max.

4.1.2 For Square Test Section

In the beginning, results of the present square rod are validated with those obtained from the standard empirical correlation of Adam, Churchill and other two relations for heat transfer coefficient given below

Nusselt number correlation

i. Empirical correlation of Churchill and Chu.

$$Nu = 0.68 + \frac{0.67 R_a^{1/4}}{[1+(\frac{0.492}{P_r})^{9/16}]^{4/9}}$$

$$Nu = \{0.825 + \frac{0.387 R_a^{1/6}}{[1+(0.492/P_r)^{9/16}]^{8/27}}\}^2$$

ii. Empirical correlation of Adam

$$Nu = 0.59 \times R_a^{1/4}$$

iii. Empirical correlation

$$Nu = 0.655 \times R_a^{1/4}$$

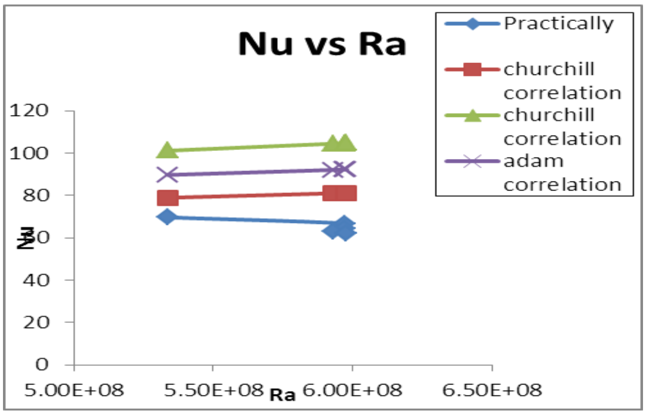


Fig.4 Validation of Square Test Section

The comparision of Nusselt number and coefficient of heat transfer for present square rod with existing correlation shown in fig.4. In this figure shows that validation experiment for heat transfers in terms of Nusselt number and heat transfer coefficient for cylindrical rod are in good arrangement with result obtained from all empirical correlation. It is found that Nusselt number in the present cylindrical tube agree with those from all empirical correlation within ± 10.470 shown in fig 4.

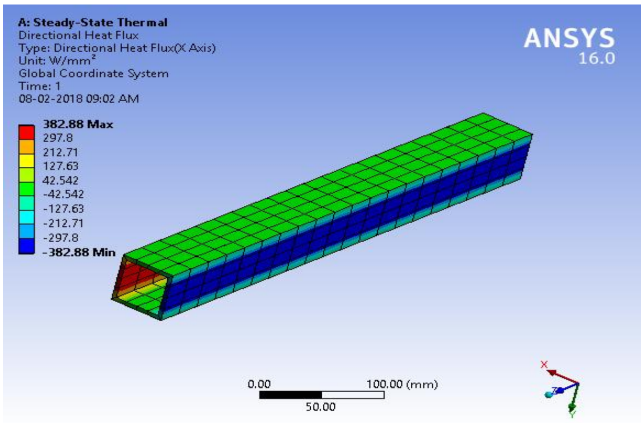
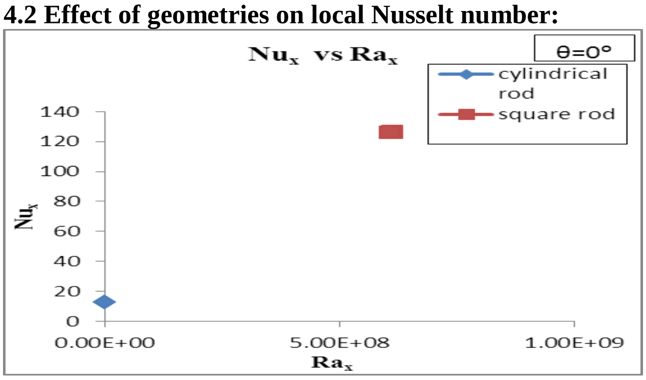


Fig .5 Details of stedy state heat flux in square rod

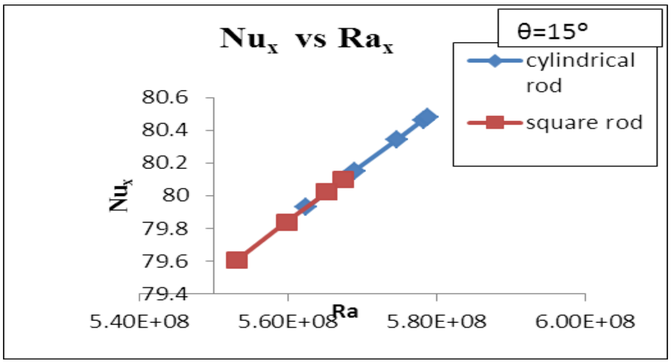
Fig.5 shows that steady state heat flux throughout the square rod .maximum heat flux occur in the test section is 382.88 max.

4.2 Effect of geometries on local Nusselt number:



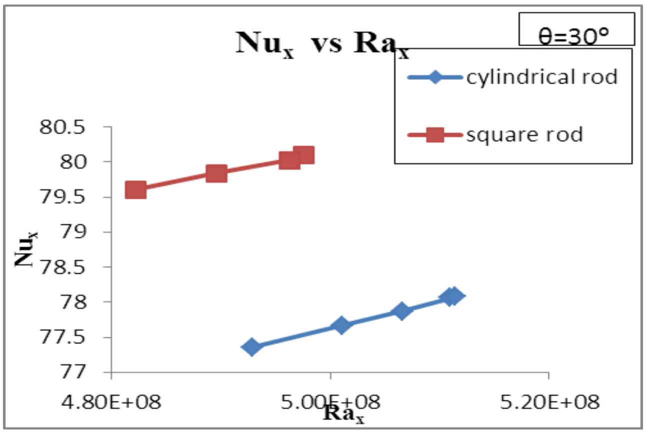
Graph. 4.2.1 *Effect of geometries on local Nusselt number at 0^0*

The above Graph. 4.2.1shows that, local Nu Vs Ra ,at $\theta = 0^0$, heat transfer rate in cylindrical test section is less than square test section



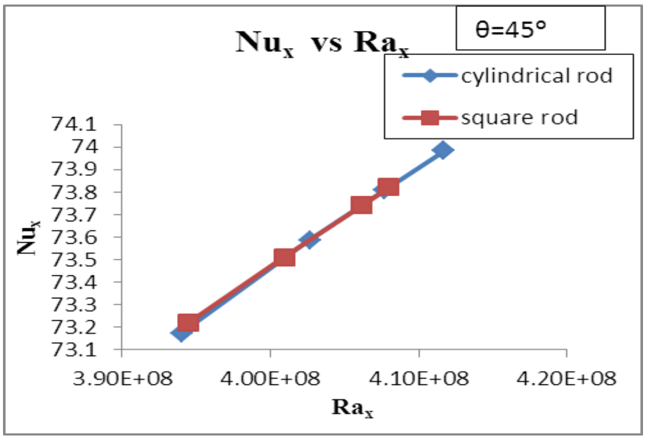
Graph. 4.2.2 *Effect of geometries on local Nusseltnumber at 15^0*

The above Graph. 4.2.2 shows that, local Nu Vs Ra at $\theta = 15^0$, heat transfer rate in cylindrical test section is more than square test section.



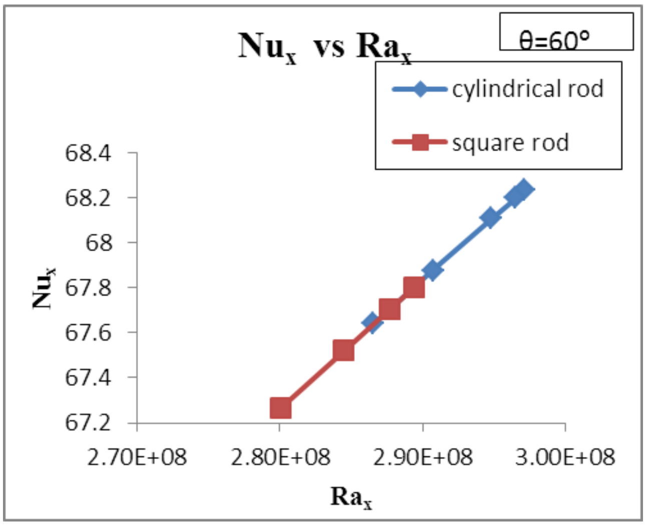
Graph. ***4.2.3*** *Effect of geometries on local Nusselt number at 30^0*

The above Graph.4.2.3 shows that, local Nu Vs Ra at $\theta = 30^0$,heat transfer rate in square test section is more than cylindrical test section



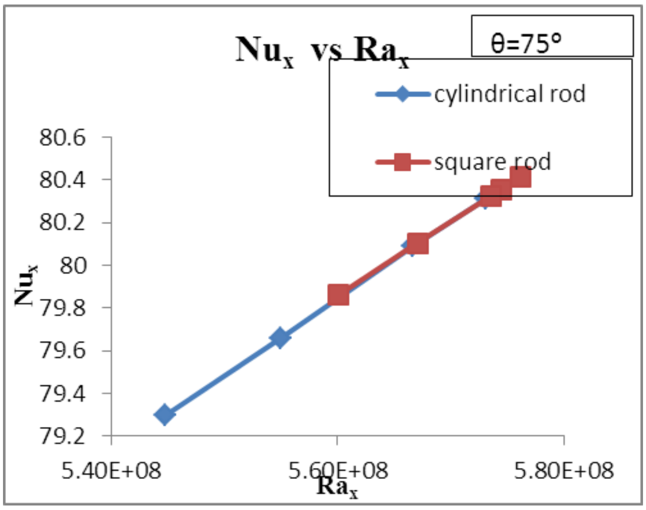
Graph. ***4.2.4*** *Effect of geometries on local Nusselt number at 45^0*

The above Graph. 4.2.4 shows that , local Nu Vs Ra at θ=45^{0,} heat transfer rate in cylindrical test section is more than square test section



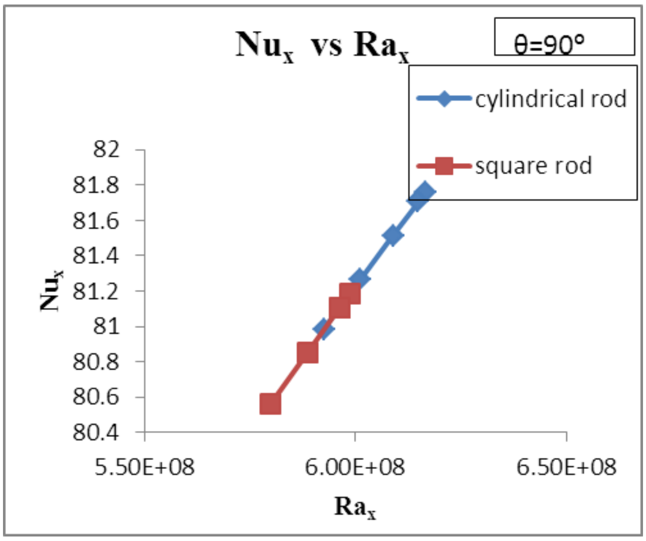
Graph. ***4.2.5*** *Effect of geometries on local Nusselt number at 60^{0}*

The aboveGraph. 4.2.5shows that , local Nu Vs. Ra at θ=60^{0} ' heat transfer rate in cylindrical test section is more than square test section



Graph .*4.2.6* *Effect of geometries on local Nusselt number at 75^{0}*

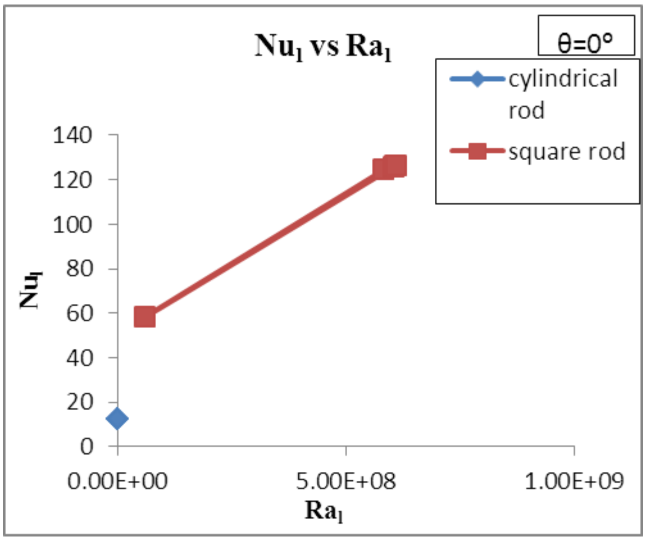
The above Graph .4.2.6shows that, local Nu Vs Ra at θ=75^{0} ' heat transfer rate in square test section is more than cylindrical test section



Graph. ***4.2.7*** *Effect of geometries on local Nusselt number at 90^{0}*

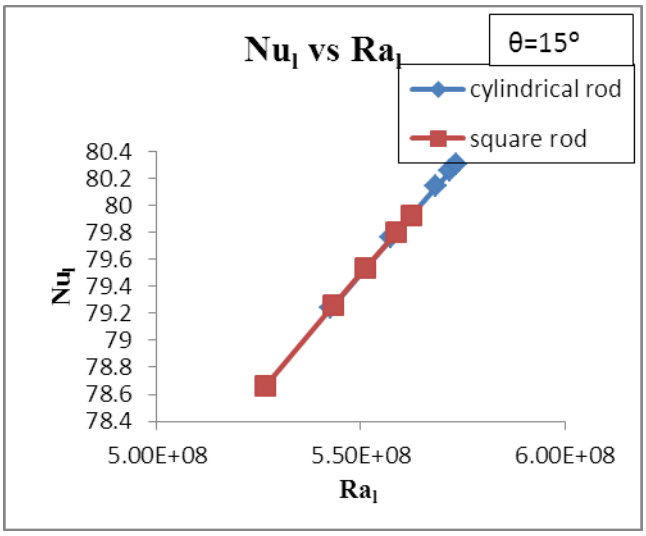
The above Graph .4.2.7 shows that, local Nu Vs Ra at θ=90^{0} 'heat transfer rate in cylindrical test section is more than square test section.

4.3 Effect of geometries on Average Nusselt number



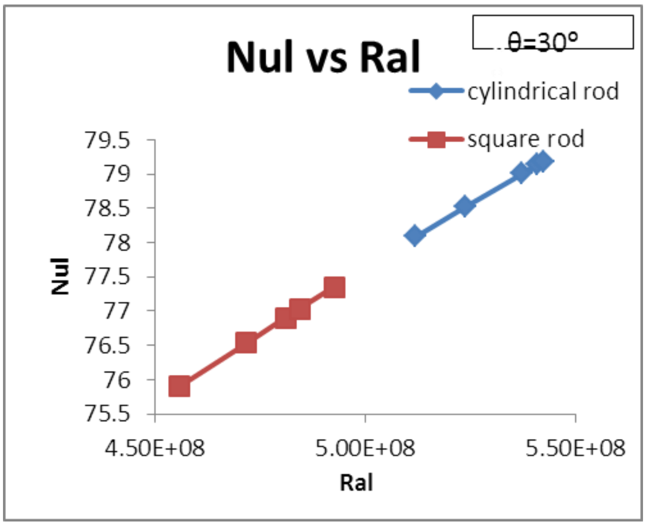
Graph. ***4.3.1*** *Effect of geometries on average Nusselt number at 0^{0}*

The above Graph .4.3.1 shows that average Nu_l Vs Ra_l at θ=0^{0}, heat transfer rate in square test section is more than cylindrical test section at 6.08E+08 $Ra_l$and at 126.36 Nu_l



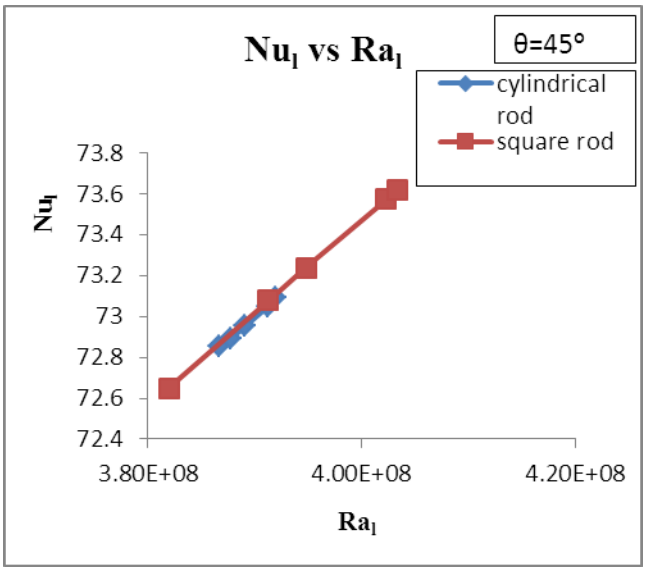
Graph. *4.3.2* *Effect of geometries on average Nusselt number at 15^0*

The above Graph .4.3.2 shows that ,average Nu_l Vs Ra_l at $\theta=15^0$,heat transfer rate in cylindrical test section is more than square test section at 5.73E+08 $Ra_1$and at 80.3 Nu_1



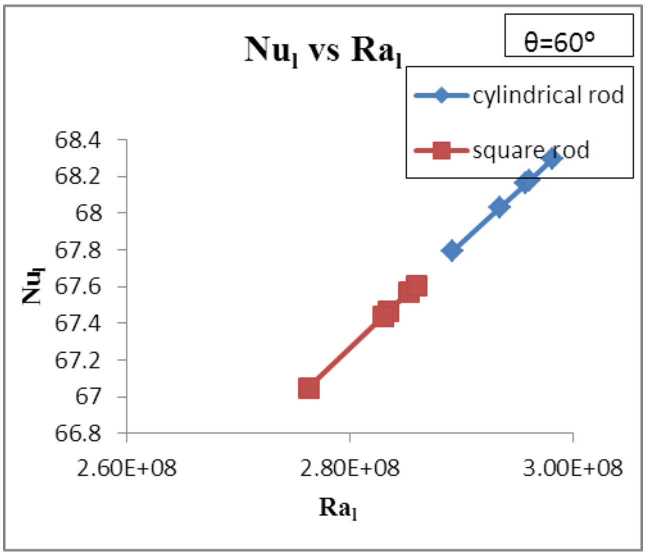
Graph. *4.3.3* *Effect of geometries on average Nusselt number at 30^0*

The above Graph .4.3.3 shows that , average Nu_l Vs Ra_l at $\theta=30^0$,heat transfer rate in cylindrical test section is more than square test section at 5.42E+08 $Ra_1$and at 79.19 Nu_1



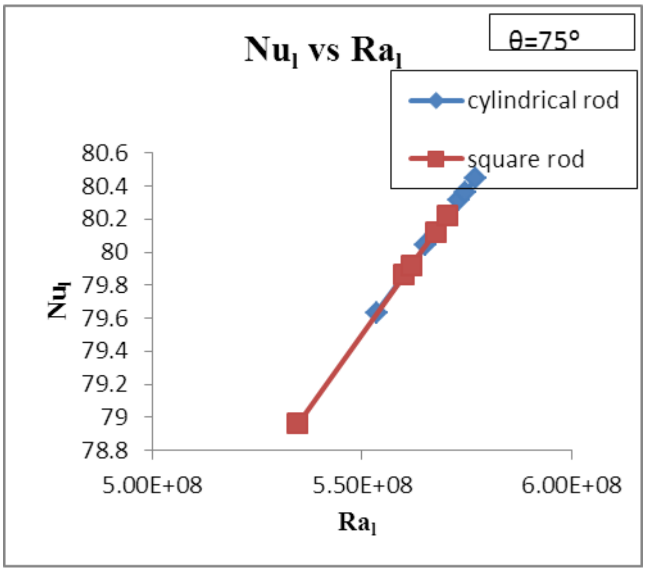
Graph.*4.3.4* *Effect of geometries on average Nusselt number at 45^0*

The above Graph .4.3.4 shows that, average Nu_l Vs Ra_l at $\theta=45^0$, heat transfer rate in square test section is more than cylindrical test section at 4.03E+08 Ra_1 and at 73.6 Nu_1



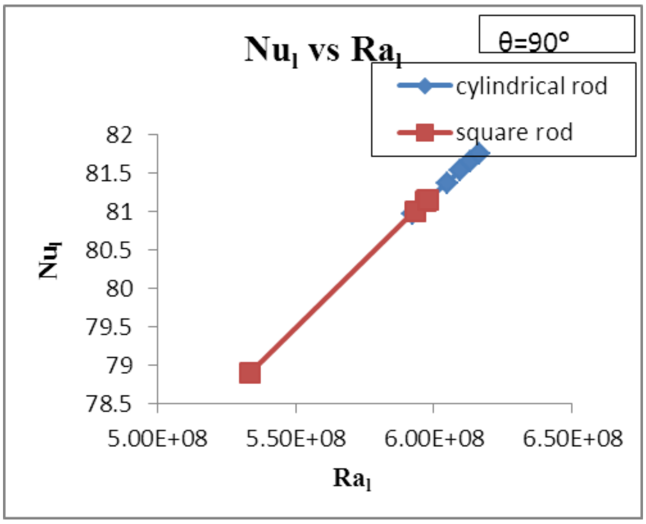
Graph. 4.3.5 *Effect of geometries on average Nusselt number at 60^0*

The above Graph .4.3.5 shows that , average Nu_l Vs Ra_l at $\theta=60^0$, heat transfer rate in cylindrical test section is more than square test section at 2.98E+08 Ra_1 and at 68.29 Nu_1 and at 80.4 Nu_1



***Graph.4.3.6** Effect of geometries on average Nusselt number at 75^0*

The above Graph .4.3.6 shows that, average Nu_lvsRa_l at $\theta=75^0$,heat transfer rate in cylindrical test section is more than square test section at 5.77E+08 $Ra_l$and at 80.4 Nu_l



***Graph.4.3.7** Effect of geometries on average Nusselt number at 90^0*

The above Graph .4.3.7 shows that, average Nu_l Vs Ra_l at $\theta=90^0$,heat transfer rate in cylindrical test section is more than square test section at 6.16E+08 Ra_l and at 81.7 Nu_l

Result-

1.Nusselt number increases proportionately with the power of the Rayleigh number.
2.The average heat-transfer coefficient obtained for cylindrical rod is 45% more than that obtained from square rod.
3.Natural convection heat transfer in cylindrical rod and square rod has been studied experimentally for Ra number between 0.00E+08 to 6.00E+08 for inclination angle between 0^0 to 90^0
4. At $\theta=0^0$, heat transfer rate in square test section is more than cylindrical test section at 6.08E+08 $Ra_l$and at 126.36 Nu_l
5. At $\theta=15^0$,heat transfer rate in cylindrical test section is more than square test section at 5.73E+08 $Ra_l$and at 80.3 $Nu_{l.}$
6. At $\theta=30^0$,heat transfer rate in cylindrical test section is more than square test section at 5.42E+08 $Ra_l$and at 79.19 $Nu_{l.}$
7. At $\theta=45^0$, heat transfer rate in square test section is more than cylindrical test section at 4.03E+08 Ra_l and at 73.6 $Nu_{l.}$
8. At $\theta=60^0$, heat transfer rate in cylindrical test section is more than square test section at 2.98E+08 Ra_l and at 68.29 Nu_l and at 80.4 $Nu_{l.}$
9. At $\theta=75^0$,heat transfer rate in cylindrical test section is more than square test section at 5.77E+08 $Ra_l$and at 80.4 $Nu_{l.}$
10. At $\theta=90^0$,heat transfer rate in cylindrical test section is more than square test section at 6.16E+08 Ra_l and at 81.7 $Nu_{l.}$

5. Conclusion

1. It is good agreement of results of heat transfer coefficient with empirical correlation for both cylindrical as well as square rest section.
2. Nusselt number, for angles (15^0, 30^0, 60^0, 75^0, 90^0) have 40% more heat transfer rate for cylindrical test section and for angle (45^0) have 50% more heat transfer rate for square test section
3. As the inclination angle increases, it increase the heat transfer rate, as it increases coefficient of heat transfer.
4. Set up is validated for vertical position of test section using ANSYS .

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