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Experimental Investigation into Natural Convection Heat Transfer inside Triangular Enclosure with Internal Hot Cylinder

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Abstract

Natural convection air heat transfer and fluid movement currents around a hot circular cylinder inside an inclined triangular enclosure has been analyzed experimentally. Three different sizes of an enclosure with a long side of 20, 25, and 30 cm, the thickness of 1 mm, and depth of 50 cm were used in the present work to give three radius ratios. The effect of Rayleigh number, radius ratio, the rotation angle of triangle enclosure, and the inclination angle of the apparatus with horizontal axis θ on the heat transfer process was investigated. The ranges of these parameters were: Rayleigh number from 5×10^6 to 2.5×10^8, radius ratio (0.345, 0.455, and 0.618), rotation angle (0^o, 45^o, and 90^o), and inclination angle (0^o, 45^o and 90^o). The results show that the heat transfer rates increase with increase in Rayleigh number and as the rotation angle of enclosure is changed from 0^o to 90^o. Moreover, the heat transfer rate increases linearly with Rayleigh number at higher radius at rotation angle 0^o, 90^o only. While, it increases slightly with Rayleigh number at rotation angle 45^o. Additionally, the higher heat transfer rates occur at vertical position of enclosure inclination angle 90^o and rotation angle 0^o (the base of triangle at the bottom) and it decreases as inclination angle deviates from 90^o to 0^o. This behavior is reverse completely at higher radius ratio 0.618. Empirical correlations for the average Nusselt number has been found to depend on Rayleigh number., radius ratio, rotation angle and inclination angle.

Keywords: Laminar Flow, Natural Convection, Triangular Enclosure, Inclination Angle, Circular Cylinder.

التحقيق التجريبي في انتقال الحرارة بالحمل الطبيعي داخل حاوية مثلثة مع اسطوانة داخلية ساخنة

عقيل عبدالله محمد ، انسام عادل محمد ، Shylesha V Channapattanac

الخلاصة:

تم تحليل نقل حرارة الهواء بالحمل الطبيعي وتيارات حركة السوائل حول أسطوانة دائرية ساخنة داخل حاوية مثلثة مائلة بشكل تجريبي. تم استخدام ثلاثة أحجام مختلفة من حاوية ذات جانب طويل ٢٠ و ٢٥ و ٣٠ سم، وسمك ١ مم ، وعمق ٥٠ سم في العمل الحالي لإعطاء ثلاث نسب نصف قطرية. تم دراسة تأثير رقم رايلي، ونسبة نصف القطر ، وزاوية دوران العلبة المثلثية ، وزاوية ميل الجهاز مع المحور الأفقي على عملية نقل الحرارة. نطاقات هذه المعلمات هي: عدد رايلي من ٥ × ١٠٦ إلى ٢,٥ × ١٠٨ ، نسبة نصف القطر (٠,٣٤٥ ، ٠,٤٥٥ ، و ٠,٦١٨) ، زاوية الدوران (٠ ، ٤٥ و٩٠) درجة وزاوية الميل (٠ ، ٤٥ و ٩٠) درجة. أوضحت النتائج أن معدلات نقل الحرارة تزداد مع زيادة رقم رايلي وتغيرت زاوية الدوران للغلاف من ٠ درجة إلى ٩٠ درجة. علاوة على ذلك ، يزداد معدل نقل الحرارة خطيًا مع رقم رايلي عند نصف قطر أعلى بزاوية دوران ٠ درجة ، ٤٥ درجة فقط. بينما يزيد قليلاً مع رقم رايلي بزاوية دوران ٩٠ درجة. بالإضافة إلى ذلك ، تحدث معدلات نقل الحرارة المرتفعة في الوضع الرأسي لزاوية ميل العلبة ٩٠ درجة وزاوية الدوران ٠ درجة (قاعدة المثلث في الأسفل) وتتناقص مع انحراف زاوية الميل من ٩٠ درجة إلى ٠ درجة. ينعكس هذا السلوك تمامًا عند نسبة نصف قطر أعلى ٠,٦١٨. وجد أن الارتباطات التجريبية لمتوسط عدد نسلت تعتمد على رقم رايلي ونسبة نصف القطر وزاوية الدوران وزاوية

الميل.

1. Introduction

Free convection plays significant functions in a variety of home and industrial settings. The most crucial area of research for engineering design, operation, analysis, and development is the study of heat transfers for cooling or heating reasons. Using diverse methods to improve heat transfer leads to better design, which saves energy, materials, keeps people safe, and protects the environment in a variety of ways [1]. Controlling the convective heat transfer rate (q) can be accomplished either by making adjustments to the working fluid type, which changes the heat transfer coefficient (h), or by making adjustments to the geometry of the system, which changes the heat transfer area (A). In Newton's theory, forced convection refers to the process of heat transfer that takes place as a consequence of fluid free stream bulk movement. The free stream motion is created by forces from the outside, such as those supplied by pumps, blowers, fans, compressors, and so on. In the prior examinations [2-7], a brief mention was made of the phenomenon known as natural convection, which is an additional activity that only occurs sporadically.

The common interpretation of this phenomenon is that it is a buoyant force brought about by the internal density gradient of a system. Analytically, natural convection is handled the same way as any other external force expression in the equation for momentum conservation that Flack and colleagues developed [8]. At the end of the 1970s, research was published on experimental measurements of spontaneous convective Heat transmission in two-sided air-filled triangular enclosures the same temperature and a base that is not moving. This paper presents the findings of the first research study to investigate spontaneous convection in enclosures with three angles. Flack's [9] investigation into the same geometries was broadened to include a variety of boundary conditions. Numerous numerical and experimental studies of Natural laminar convection and heat transfer in horizontal triangular enclosures have been carried out since then [10]. Some of studies [11-23] have been carried out for a variety of aspect ratios, boundary condition combinations, and a wide range of Rayleigh and Prandtl numbers. Some of these studies were designed specifically to investigate convective flow patterns, bifurcation phenomena, and stability. Subsequently, localized non-uniform heating boundary conditions were established, and a triangular enclosure was outfitted with flush-mounted heaters [24] then projecting heaters [25]. Abu Nada [26], studied the effects of the Rayleigh number, the inclination angle, and the aspect ratio on the free convection heat transport in a concentric cylindrical enclosure.

The results showed that when the aspect ratio goes up, the amount of heat transfer goes down a lot. On other hand, the heat transfer rate increases somewhat with an increase in tilt, and it is closely connected to the Rayleigh number. Yesiloz and Aydin [27] investigated the effect of Ra and the degree of inclination relating to heat transfer by natural convection in a quadrant cavity. Optimal conditions were achieved between 0 and 270 degrees of inclination. Fand et al. [28] studied the free convection heat transfer from a horizontal cylindrical enclosure wall to bulk fluids of air, water, and various types of silicone lubricants in the regions of Ra of $2.5x10^2$ to $1.8x10^7$ and Pr of 0.7 to 30000. Nu=f(Ra) or Nu=f(Ra,Pr) .In 1983, Sparrow and Charmcill [29] investigated the effect of the effect of temperature on the free heat transfer coefficient from convection of the vertical annulus space between two concentric cylinders (the cylinder enclosure of the internal cylinder). The working fluid was air with a range of $1.5*10^3$ to 10^5 for the Rayleigh number. Hamad [30] investigated how the angle of inclination affected the convection of heat in an inclined annular cylinder. In this investigation, a pair of 900-mm-long aluminum cylinders with inner cylinder diameters of 44 mm and outer cylinder diameters of 72 mm were utilized.

The inner cylinder was set to 900 W of heat, which was supplied by a constant heat discharge from an electric radiator. Kitamura et al. [31] investigated the properties of free convection by observing the flow patterns around the exterior surface of the cylinder and quantifying the surface temperature distribution of the outer walls. The researchers observed that three-dimensional flow segregation initially occurred at the irregular edges of the cylinder before transforming into a turbulent transition. In regions of transitional and turbulent flow, the resident Nusselt numbers significantly increase, despite covering only a small portion of the cylinder's surface, according to the researchers' findings. Akeel et al. [32–35] investigated the effect of theoretically distinct parameters on the behavior of natural fluid flow and convection heat transmission within an enclosure. These parameters included porous media, corrugated enclosures and inner cylinder shapes, and nanofluid. A triangular insert surrounding a circular cylinder that is uniformly heated has been the subject of experiments on natural convection. Various parametric operating conditions, including the modified Rayleigh number, aspect ratio, eccentricity, and orientation angles, have been considered. The range of the Rayleigh number was between ($5 x 10^6$) and ($2.5 x 10^8$) On the basis of hydraulic diameters, the radius ratios were 0.34, 0.45, and 0.61. The orientation angles of the triangular cylinder were (0°, 45°, and 90°), whereas the enclosure's inclination angles were (0° and 45°). Air was the working fluid (Pr = 0.7). The experimental apparatus was installed to create all of these conditions in order to replicate the objective of this study.

2. Experimental Setup

Trendy In this section, the experimental configuration is discussed. Thus, the evaluated samples and on-site experimental apparatus are described in detail. The experimental rig of present work setup is illustrated in Fig.1 and Fig.2. The experimental investigation section is equilateral triangular enclosure containing a uniformly heated circular cylinder by using electric heater inserted inside it. The electric circuit was conducted to heater to regulate heat flux by changing the electric potential. The thermocouples were used to gauge the temperature. placed on the outer surfaces of circular and triangular cylinders.

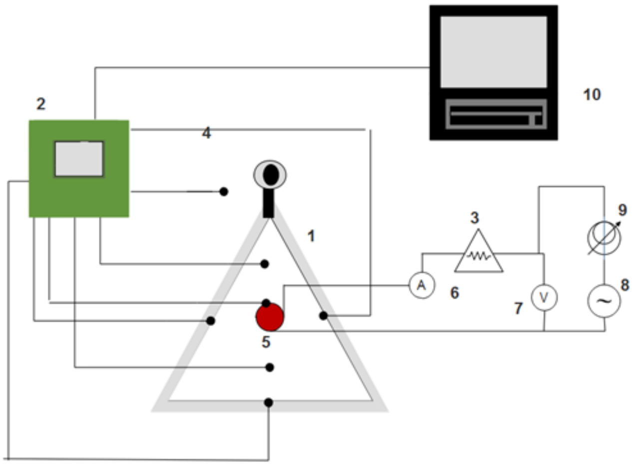

Figure (1): Experimental setup: 1. Triangular enclosure, 2. Microprocessor, 3. Resistor, 4. Thermocouple, 5. Heater, 6. Ammeter, 7. Voltmeter, 8. Electrical source. , 9. Variac adjuster and 10. Personal computer

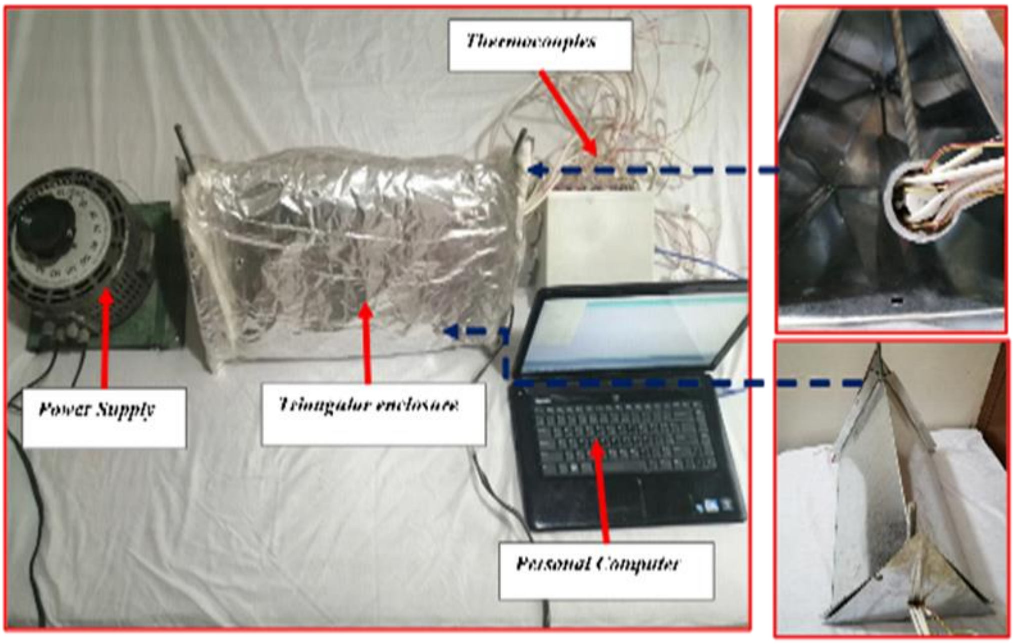

Figure (2): Photographical view of experimental setup.

The orientation of triangular enclosure can be changed by using angle adjusting regulator. Fig.3 shows the physical domain of the problem. The aspect ratio can be also adjusted by using different sizes of triangular enclosures.

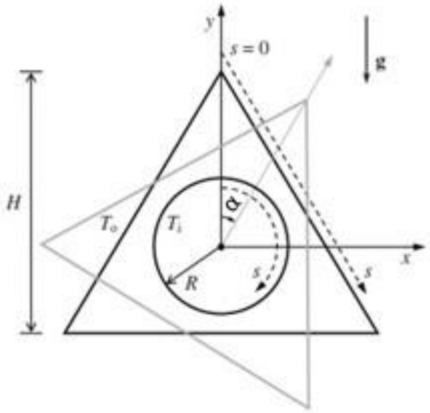

Figure (3): Schematic diagram of Triangular enclosure orientations.

Triangular enclosure was fabricated from aluminum as a triangular duct as shown in Fig.2. The ends of triangular duct were closed by triangular plate of aluminum to configure the enclosure. Then, the enclosure was carefully insulated using fiber glass with thickness of 2 mm. Three different size enclosures with length side of 20, 25 and 30 cm, thickness of 1 mm, and depth of 50 cm were used in the present experimental work to give three aspect ratios.

A heated cylinder of Aluminum with 5 cm diameter, 1 mm thickness and 50 cm length was placed inside the triangular enclosure. The heater with an electrical resistance of 49 Ω and maximum power of 1000 W was carefully wrapped around Teflon cylinder with5 cm diameter and 50cm length. This Teflon cylinder was inserted concentrically inside Aluminum cylinder. The central heater was used to generate a constant heat flux on the outer surface of inner circular cylinder. To carry out the experiments, initially selected the aspect ratio by choose one of the three triangular enclosures (faces lengths of 20, 25 and 30 cm). Then adjust the inclination angle (θ=0° horizontal, 45° inclined, and 90° vertical) and triangle rotation (α=0° base of triangle at the right,90° base of triangle at the bottom, 180° base of triangle at the top). Where we can control the amount of heat flux entering the system through the input voltage to the heater.

The device waited more than three hours for the situation steady-state at the lowest needed voltage before switching the following voltage, which was at the very least the lower one in order to cut down on the amount of time that is needed to attain the steady-state circumstance.

2.1. Data Analysis

The temperature distribution inside the triangular enclosure was used to obtain the data. The Surface heat flow can be continuous computed using the formula below :

$$q = \left(\frac{V * I - Q_{rad}}{A}\right) \quad \dots(1)$$

Where V denotes electrical potential, I denotes electric current, A is the outer surface area of the inner cylinder, and Q rad denotes the heat transfer through radiation between the circular and triangular cylinders. The experimental results show that the radiation heat flow is negligible. and can be neglected. The calculated heat fluxes for various voltages are shown in Table.1:

Table (1): Heat Fluxes Taken at Various Voltages used in Present Study.

V(Volt)	*I(Amp)*	*q(W/m²)*
12	*0.23*	*35.15924*
15	*0.31*	*58.47134*
18	*0.38*	*87.13376*
21	*0.43*	*114.7643*
24	*0.49*	*149.8089*
30	*0.53*	*202.5478*

The following expression may be used to determine the heat transfer coefficient:

$$H = \frac{q}{T_{in} - T_\infty} \quad \dots\dots (2)$$

Where T_∞ is the average temperature of the heated air inside the cage.

From the following, the Nusselt number may be calculated:

$$Nu = \frac{hL}{k} \quad \dots\dots (3)$$

The Rayleigh number can be calculated from the following:

$$Ra = \frac{g\beta(T_{in} - T_{out})L^3}{\vartheta_\alpha} \quad \dots\dots (4)$$

Where L is the characteristics length ($L = R_{out} - R_{in}$).

The linear relationship between the logarithmic Rayleigh number and the logarithmic mean Nusselt number may be fitted to the data to obtain the empirical correlation for each aspect ratio, angles of apparatus inclination, and angles of triangle rotation as reflected in the following:

$$Nu = cRa^n \quad \dots\dots (5)$$

c and n are empirical constants that may be altered by altering the enclosure. aspect ratio, and inclination and rotation angles.

At the average mean film temperature, all of the air's physical characteristics, including, v, and, were assessed (T_f).

$$T_f = \frac{T_{in} + T_{out}}{2} \quad \dots\dots (6)$$

2.2. Error Analysis

There is no doubt that the majority of calculation errors were primarily attributable to errors in the measured quantities. The Kline and McClintock method [43] is used in this field to calculate the error in the obtained results.

Let R be a function of one or more independent variables. v1, v2 … vn

$$R = R(v_1, v_2 \dots . v_n) \quad \dots (7)$$

Hence, the experimental errors that may well happen due to an implementation of the variables are set in Table (2) which is taken from the computing devices as follows:

Table (2): Uncertainties of Measuring Devices

*Independent variable **(v)***	*Uncertainty Interval "Taken from the Measuring Devices"**(w)***

Surface to bulk air temperature	*±0.16 °C*
Voltage of the heater	*±0.9 °C*
Current of the heater	*±0.05425 Amp*
Hydraulic diameter	*±0.0002 m*

The local Nusselt number equation can be written as follows

$$Nu = \frac{qL}{(T_{in}-T_{\infty})k} \quad \ldots (8)$$

$$Nu = \frac{VIL}{(T_{in}-T_{\infty})Ak} \quad \ldots (9)$$

$$\langle\frac{w_{Nu}}{Nu}\rangle^2 = \left[\langle\frac{w_v}{V}\rangle^2 + \langle\frac{w_I}{I}\rangle^2 + \langle\frac{w_L}{L}\rangle^2 + \langle\frac{w_{\Delta T}}{\Delta T}\rangle^2 + \langle\frac{w_A}{A}\rangle^2\right] \quad \ldots (10)$$

Where

A = inner cylinder surface area = $2\pi R_{in} l$

L= length of inner cylinder=50 cm

$\Delta T = (T_{in} - T_{\infty})$ …. (11)

3. Experimental Results

3.1. Effect of heat flux on temperature distribution

The heat flux term has direct effect on temperature distribution according to equation 3. The rising heat flux leads to temperature increasing in default proportion. Figure 4 shows the seeming temperature distribution sideways cylinder for various heat fluxes at rr=0.455, α=0° and θ=0°. The results show that there are no significant changes of temperature along the axial cylinder surface by a constant value of heat flux. While heat flux varies the temperature in linear proportion. The reason of the temperature being constant along the cylinder surface is the measured nature of the experimental data was at steady state, the heat will transfer to all cylinder surface by action of conductive heat transfer.

The heat flux varies linearly with Rayleigh number Ra as shown in Figure 5 for various rr in α=θ=0°. It is noticed that Rayleigh number decreases as aspect ratio rr increases. The decreasing of rr leads to increasing the enclosure hydrodynamic diameter which has proportion to power 4 within Ra.

3.2. Effect of Ra, rr and α on Nu at θ=0°

Figure 6 shows difference of Nusselt number against Ra on behalf of various α at rr = 0.345. The Nu increases by increasing Ra, the higher Nusselt number is observed at α=90°. The large enclosure at α=90° promotes the higher heat transfer area as comparing to 45°, hence the amount of cold air will drop to lower region of cylinder where the hot air will rise to the base of enclosure. The maximum increasing of Nu is observed at α=90° and Ra=2.06×10^8 about 19 % as compared to α= 45°. Moreover, the amount of contact cold air for higher Rayleigh numbers is higher at α= 90° than that at α= 45° due to geometry orientation and its relation to gravity action.

Fig 7 indicates a plot of Nusselt number as opposed to Rayleigh number aimed at various α at rr=0.455 and θ=0°. The power proportion between Nu and Ra is observe, the Nu range increases by increasing α in rr=0.455 in lower Ra but it will be maximum at α=45 when Ra>$5\text{x}10^7$. It seems to be the amount of dropped cold air is less to 0.35 cases, so the behavior of Nu by changing α is different. The heat transfer area of natural convection at α= 45° is higher than 90 ° at rr=0.455. In other worlds, the contact time of viscous forces is higher than gravitational forces in higher rr.

Figure 8 shows Nusselt number versus Rayleigh number for various α at rr=0.618. The Nu increases by increasing Ra, the higher Nu is observed at α=90° when Ra>$2\text{x}10^7$. The small enclosure at 90° promotes the higher heat transfer area as comparing to 45° and 0°, hence the amount of cold air will drop to lower region (cylinder region) where the hot air will rise to the base of enclosure. The maximum increasing of Nu is observed at α=90° and Ra>$2.0\text{x}10^7$ approximate to 33 % as compared to α= 45°. The Nu range at α=45° is higher than α=0 and becomes lower than 0° at Ra $2.6\text{x}10^7$. The non-uniform trend in small enclosure (rr=0.618) has been developed because the interaction between the cold and hot boundary layers may be happened as mentioned in [17]. Generally, the heat transfer rates increase as α deviates from 0° to 90°.

3.3. Effect of Ra, rr and α on Nu at θ=45°

Figure 12 shows Logarithmic Nusselt number versus Logarithmic Rayleigh number for various α at rr=0.345 at θ=45°. Generally, the behavior of heat transfer rate is the same as horizontal enclosure θ=0° at lower value of radius ratio only rr=0.346. The maximum Nu at 90° is higher by 20 % than that at 45°. It seems that the gravitational forces effect on heat transfer coefficient more viscous forces inside the enclosure when the larger enclosure length is high and inclined in 45 °. The linear fitting controls the relation between Nu and Ra.

Figure 13 shows Rayleigh number against the Nusselt number for various α at rr=0.618 at θ=45°. As shown in this figure that, at inclined angle of enclosure θ=45° and higher value of radius ratio rr=0.618, the behavior of heat transfer process is reversed. The heat transfer increases proportion by way of angle of enclosure rotation α declines. The maximum Nu value is achieved at α=90 and Ra=$6\text{x}10^6$ about 63 %. The decreasing in Nu will drop by increasing Ra and the Nu of other angles higher than 0 at Ra=$2.6\text{x}10^7$. It seems to be that the heat transfer is affected by gravitational forces and inertial forces in proportion according to radius ratio rr. The lower rr promotes the direct proportion with angle 0°, and the behavior has the opposite function in higher rr. Figures 14, 15 and 16 show the variation of Log Nu with Log Ra for various rr at θ=45° and α=0°, 45° & 90°. Lower rr presents higher ranges of

Nu and Ra in same heat fluxes and vice versa unlike to θ=0 case. In lower α, the increasing the Nu is lower than the higher α. The divergence between Nu values of higher rr and lower rr increases by α increasing.

Generally, the behavior of heat transfer rate is the same as horizontal enclosure θ=0° at lower value of radius ratio only rr=0.346. The rate of heat transmission quickens. Generally, with reduce the radius ratio and becomes linearly with Rayleigh number at higher radius ratio at α=0°, 45° only and increase slightly with Ra at α=90°.

3.4. Effect of Ra, rr and θ on Nu at α=0o

Figures 17, 18, and 19 show logarithmic Rayleigh number against the Nusselt number for different angles of inclination θ and at α=0° (the base of triangle at the bottom) and rr=0.35, 0.455, and 0.618; respectively. The increasing in θ increases the Nu values significantly. Generally, these figures show the higher heat transfer rates occur at vertical position of enclosure (θ=90°) and decrease as θ deviates from 90o to 0o. This behavior is reverse completely at higher radius ratio (rr=0.618). The maximum heat transfer rising at rr=0.356 and 0.455 is about 50 % and 54%; respectively, at Ra=5x107 and θ=90°. It looks that the viscous forces by the action of elevated hot air is higher than the gravitational forces which has less effect on heat transfer rate at higher θ. Figures 20, 21 and 22 show the disparity, of logarithmic Nusselt number versus Rayleigh number for various rr at α=0 and θ=0, 45° &90°; respectively. The lower radius ratio gives higher ranges of Rayleigh number and Nusselt number at the same heat fluxes and vice versa. It's found that the increasing in radius ratio for same Ra Increases Nu.

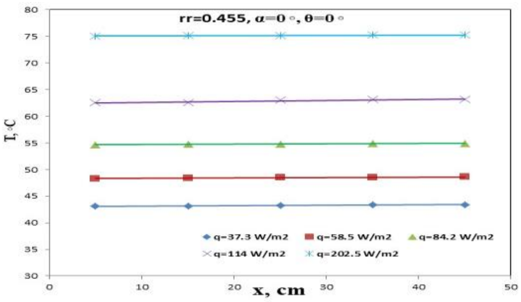

Figure (4): Cylinder surface temperature variation along x-axis for various heat fluxes at rr=0.455, α=θ=0.

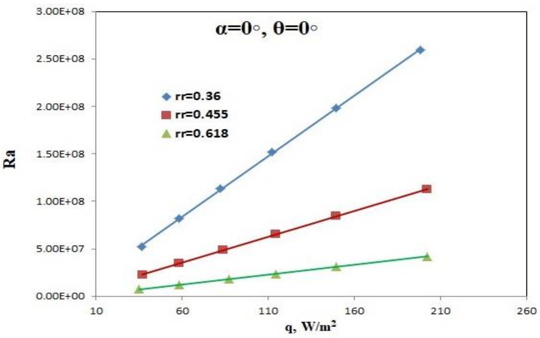

Figure (5): Modified Rayleigh number Ra versus heat flux for various rr at α=θ=0°.

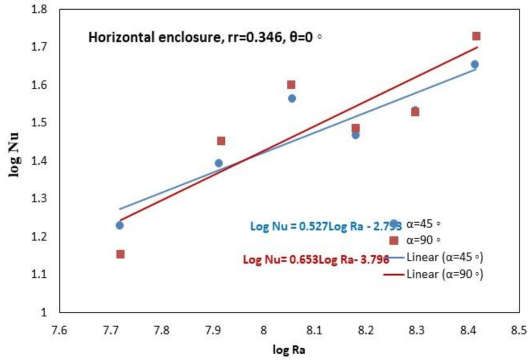

Figure (6): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various α at θ=0° and rr=0.35.

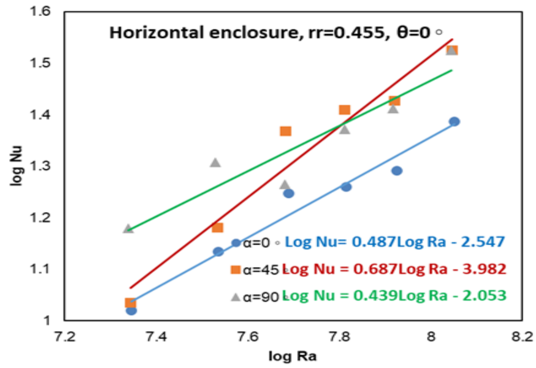

Figure (7): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various α at θ=0 ◦ and rr=0.455.

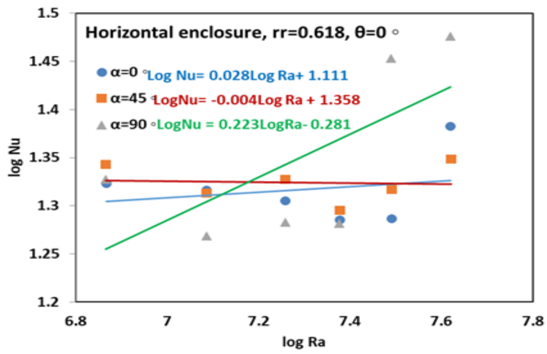

Figure (8): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various α at θ=0 ◦and rr=0.618.

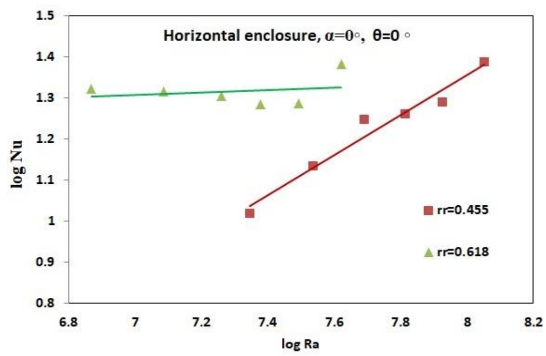

Figure (9): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various rr at θ=0 and α=0.

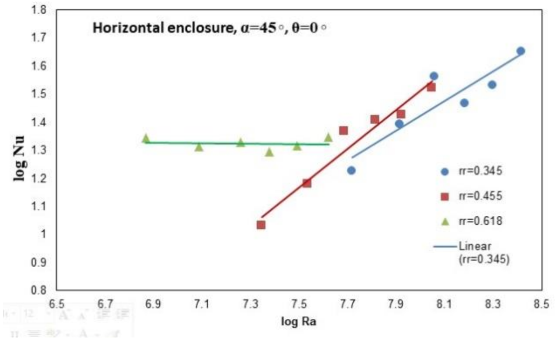

Figure (10): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various rr at θ=0 ◦and α=45.

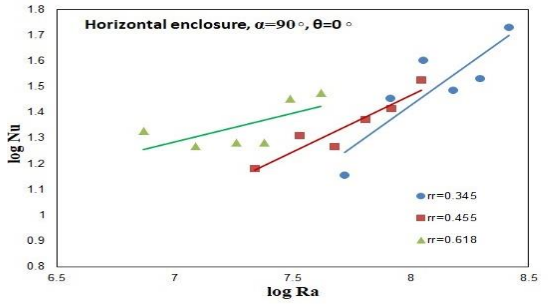

Figure (11): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various rrat θ=0◦ and α=90◦.

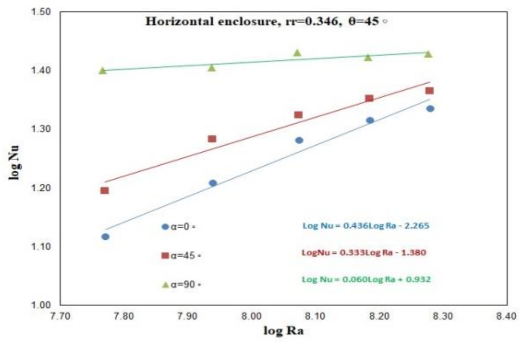

Figure (12): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various α at θ=45◦ and rr=0.35.

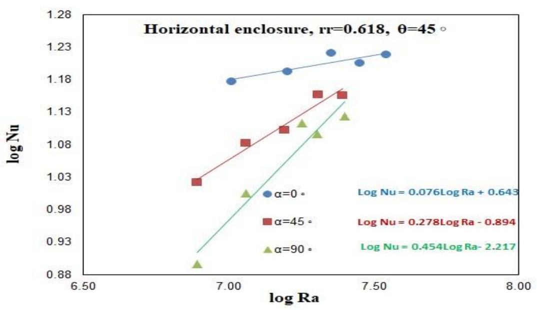

Figure (13): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various α at θ=45 ◦ and rr=0.618.

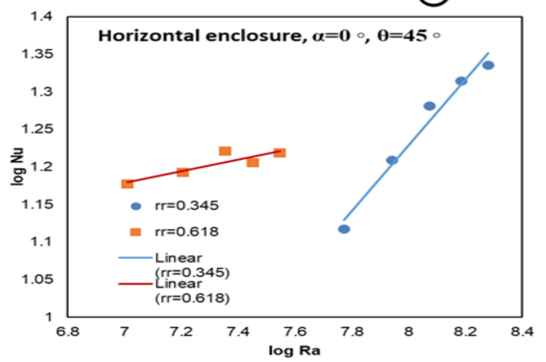

Figure (14): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various rr at θ=45◦ and α=0.

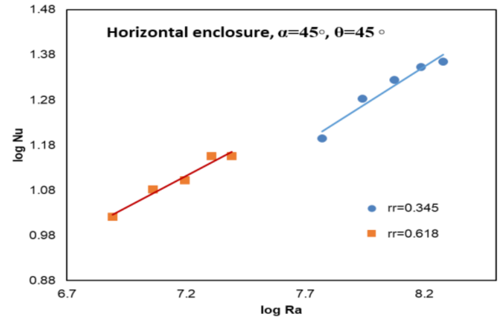

Figure (15): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various rr at θ=45◦ and α=45◦.

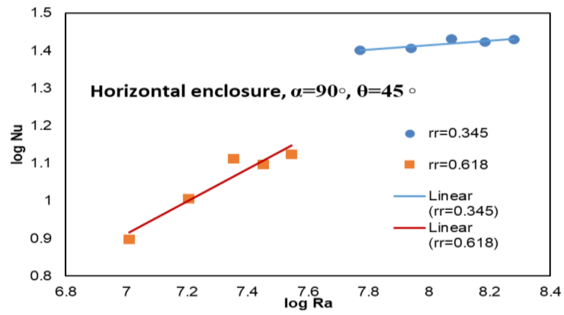

Figure (16): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various rr at θ=45o and α=90◦.

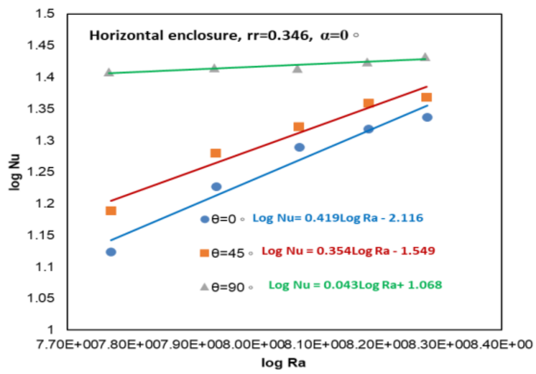

Figure (17): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various θ at α=0◦ and rr=0.345

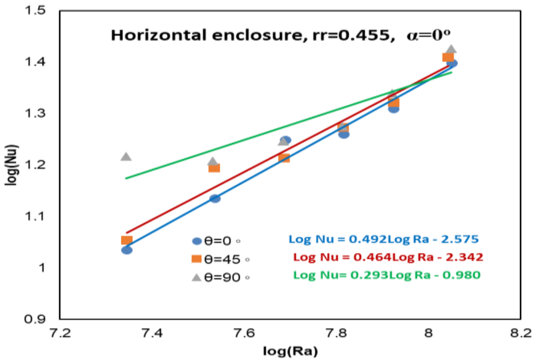

Figure (18): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various θ at α=0 and rr=0.455.

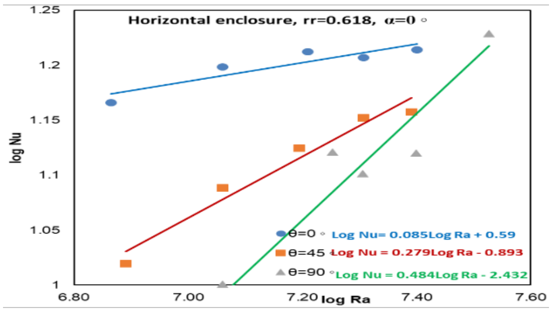

Figure (19): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various θ at α=0°and rr=0.618.

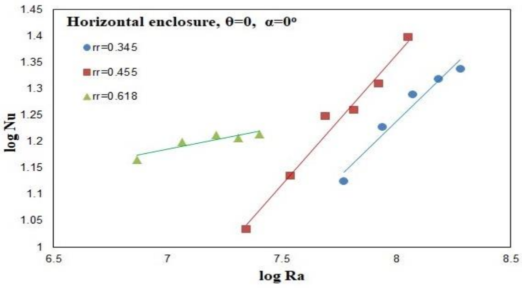

Figure (20): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various rr at α=0 and θ=0.

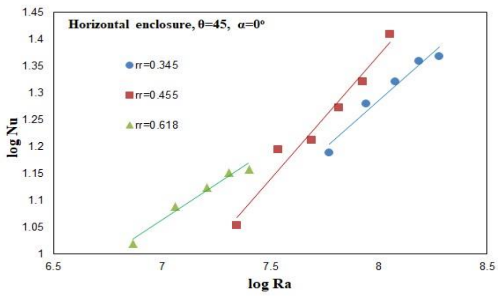

Figure (21): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various rr at α=0o and θ=45°.

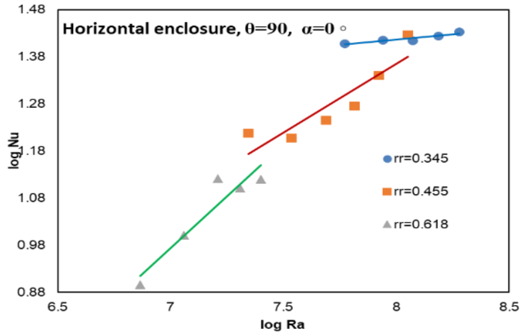

Figure (22): Logarithmic Nusselt number versus Logarithmic Rayleigh number for various rr at α=0 and θ=90°.

3.5. Empirical Correlations

The empirical correlation that describes the heat transfer, by normal convection in triangular enclosure at different angles of inclination and orientation is the same equation 6. The constants of equation 6 resulted from experimental work can be seen in table 3.

The general empirical correlation of present work has been deduced from figures 23, 24, and 25 as follows:

$$Nu = b\left[Ra\frac{\left(\frac{\pi}{2}+\theta\right)^{0.5}}{rr^{0.8}}\right]^{n} \quad ......(12)$$

Where b and n are empirical constants certain in table 4.

Table(4): Constants' Values of Equation 5 for Exerimental Work.

a	*B*	***n***
0°	*-0.291*	*0.19*
45°	*-0.297*	*0.199*
90°	*-0.281*	*0.267*

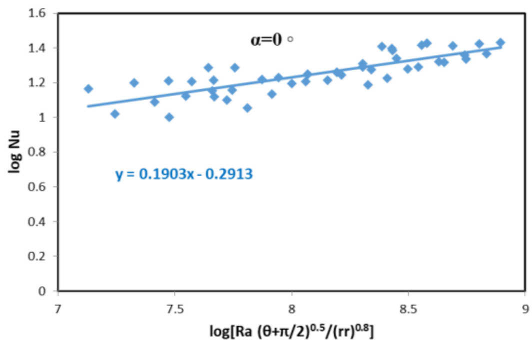

Figure (23): Logarithmic Nusselt number versus logarithmic $\left[Ra\frac{\left(\frac{\pi}{2}+\theta\right)^{0.5}}{rr^{0.8}}\right]$ for α=0°.

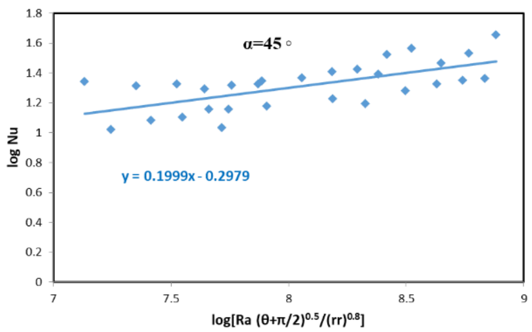

Figure (24): Logarithmic Nusselt number versus Logarithmic$\left[Ra\,\frac{\left(\frac{\pi}{2}+\theta\right)^{0.5}}{rr^{0.8}}\right]$ for α=45°

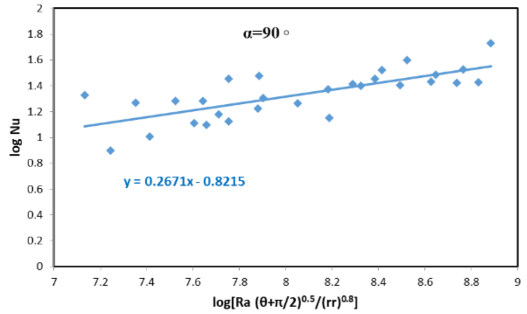

Figure (25): Logarithmic Nusselt number versus Logarithmic$\left[Ra\,\frac{\left(\frac{\pi}{2}+\theta\right)^{0.5}}{rr^{0.8}}\right]$ for α=90.

5. Conclusions

The free convection inside the triangular enclosure containing uniformly heated circular cylinder for various parameters has been successfully investigated experimentally under laminar constant surface heat flux conditions. From the present work, we conclude that:

1- There are no significant changes of temperature along the axial cylinder surface at the constant significance of heat flux. While the heat flux varies the temperature in linear proportion.
2- The Nusselt number climbs linearly along with the Rayleigh number.
3- Generally, the heat transfer rates increase with decrease in radius ratio and as rotation angle α is rotated from 0° to 90°.
4- Higher heat transfer rate occurs at vertical position of enclosure (θ=90o), α=0o (the base of triangle on the bottom), and lower radius ratio and it decreases as θ deviates from 90o to 0o. This behavior is reverse completely at higher radius ratio (rr=0.618).
5- General empirical equations of the relationship between the Nusselt number and the Rayleigh number Ra, radius ratio rr, and angle of inclination θ were deduce for three rotation angles α.

Nomenclature

A	Surface area (m^2)
Cp	Heat capacity (J/Kg. K)
g	Gravity acceleration (m/s^2)
Gr	Grashof number
h	Individual heat transfer coefficient ($W/m^2.K$)
L	Characteristics length (m)
K	Thermal conductivity (W/m. K)
Nu	Nusselt number
q	Heat flux (W/m^2)
T	Temperature (°C)
Ra	Rayleigh number
*Ra**	Modified Rayleigh number
rr	Radius ratio
Pr	Prandtle number
U	Velocity components in x and y direction (m/s)
W	Uncertainty
	Greek letters
ν	Kinematic viscosity (m^2/s)
β	Thermal expansively (1/K)
α	Radial angle of enclosure
ϱ	Fuid density (kg/m^3)

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