Convective Heat Transfer in Vertical Eccentric Annulus with Rotating Inner Cylinder

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Abstract: -

The natural, forced and mixed convection heat transfer process in vertical eccentric open ended and opened upper end annulus with stationary and rotating inner shaft and uniformly heated stationary outer cylinder has been experimentally investigated. The study covers a wide ranges of heat flux 80≤q≤500 W/m^2, rotational Reynolds number Re_{Ω}=0 (stationary inner shaft) and 50≤ Re_{Ω} ≤100 for laminar Couette flow ($Re_{\mathrm{cr}} = 110$), and 150≤ Re_{Ω} ≤300 for turbulent Taylor vortex flow, and Richardson number 0.23≤Ri≤30. The results showed that the heat transfer process improves as the eccentric ratio increases, and the heat transfer process in open ended eccentric annulus is better than that in opened upper end eccentric annulus by **7.5 %.** Four empirical correlations for average Nusselt number (Nu_m) have been deduced as a function of ε and Ra for pure natural convection and as a function of ε and Ri, for mixed convection. The results are compared with previous works and show a good agreement.

Key words: Annulus; convection heat transfer; rotating inner cylinder.

1. INTRODUCTION

Heat transfer in concentric annulus with rotation either or both tubes has wide applications in rotating electric machines, bearings, gas or oil exploration drills, boring rings, rotors of helicopters, etc. Because of these applications, many researchers have been evincing interest in this fields.

Much of the experimental work reported concerns with convection heat transfer in open ended or enclosure annulus at vertical and horizontal positions. **[12]** studied the natural convection heat transfer in an annulus with rotating inner cylinder and stationary outer cylinder and deduced an empirical equation for Ta>40 $Nu_m = 0.21(T_a^2.P_r)^{1/4}$. **[5]** examined the natural convection heat transfer process in an annulus with rotating uniformly heated inner cylinder and outer cooled cylinder and deduced an empirical equation

$Nu = 0.11(G_rP_r)^{0.29}$. **[8]** concluded that the eccentricity of the inner cylinder enhances the overall heat transfer coefficient change by less than 10% over the range of eccentricities. **[13]** concluded that the maximum local Nusselt number at inner and outer cylinder occurs at the inclination angle equals to 60^0 and 75°, respectively.**[3]** studied Taylor vortex flow under uniformly heated rotating inner tube and cooled stationary outer tube as boundary conditions and deduced an empirical correlation $Nu = 0.22\, Ta^{0.25} Pr^{0.3}$. **[7]** examined the heat transfer process in concentric annulus at a radius ratio of 0.52, with rotating inner cylinder and stationary outer cylinder. **[6]** concluded that the rate of heat transfer is greater for eccentric pipes in comparison with the concentric and single pipes. **[11]** concluded that, for the case of rotation of the inner cylinder, the heat transfer process will increase with the increasing rotational Reynolds number under the interaction of the centrifugal buoyancy force. **[10]** submitted an empirical equation for average Nusselt number on the rotating inner cylinder as a function of Rayleigh number, aspect ratio, and rotational Reynolds number $Nu=0.668 Ra^{0.27} AS^{-0.264} R^{0.14}$.**[1]** showed the vortex centers displace towards the outer cylinder wall when the radius ratio increases.**[13]** examined the helical flow in the region between two concentric cylinders during operations of drilling oil. The correct prognostic drilling fluid flow in an annular space between the well-bore wall and the drill cylinder is important to determine the variation in fluid pressure within the wellbore. The results showed that the introduction of the inner tube rotation rise the pressure drop of these fluids. **[9]** studied the vibration effect on the mixed convection heat transfer at the entrance region of concentric vertical annulus with rotating inner cylinder and outer stationary heated cylinder with radius ratio of 0.365. They concluded that the local Nusselt number increases as forced frequency increases. **[2]** examined the heat transfer process by mixed convection in the entrance region of horizontal and vertical concentric annulus with uniformly heated stationary outer cylinder and rotating inner shaft. They concluded that the local Nusselt number values increase as Taylor number increases.

As can be seen from the above literature, there are little available literatures concerning the heat transfer by natural, forced and mixed convection in vertical open ended and opened upper end with seal lower end eccentric annulus with rotating inner cylinder. The present study covers this lack and gives a clear view to actual physical behavior in the heat transfer process.

2. EXPERIMENTAL SETUP

The test section as shown schematically in **Fig.1** consists of two eccentric cylinders made from aluminum. The stationary outer cylinder has dimensions of 60 mm diameter and 100 cm length; and the rotating inner shaft has dimensions of 3 mm wall thickness, 30 mm diameter, 125 cm length. The outer cylinder is heated at constant heat flux by heater wire made of Nickel-Chromium wire which has resistance of 4.0 ohm per length. The wire is insulated by means of ceramic beads to prevent electric contact between heater and surface cylinder. The outer cylinder is insulated with asbestos layer with 45mm thickness to reduce the heat losses. To determine the heat loss from the surface of test section, three thermocouples were fixed on the asbestos insulation surface to measure the average temperature at the insulation surface. Knowing the thickness and thermal conductivity of the asbestos insulation, the heat loss thus can be calculated. Two Teflon rings are fitted at the upper and lower ends of outer cylinder with the same of its inner diameter and 100 mm outer diameter. The Teflon was chosen due to its low thermal conductivity (0.25 W/m.K) to reduce the heat loss from the aluminum outer cylinder ends. The outer cylinder surface temperatures were measured by twenty five thermocouples (type K) arranged along the outer cylinder. All thermocouples were used with leads and calibrated using the melting point of ice made from distilled water as a reference point, the boiling point of distilled water and the boiling points of several pure chemical substances. The rotating of inner shaft with different rotating speeds was achieved by electric motor and the angular velocity is controlled by using dimmer (change voltage). The angular velocity Ω is measured by handheld digital tachometer in revolution per minute (r.p.m.). The heater electric circuit consists of electric transformer type Phillips to adjust the heater input power as required, and a digital multimeter to measure the heater voltage and current.

The inlet air temperature was measured by one thermocouple located through the lower piece of Teflon, while the outlet bulk air temperature was measured by two thermocouples located through upper piece of Teflon.

The experiments were carried out through the following procedure: (1) the electric heater and the motor for the rotating inner shaft are switched on. (2) The apparatus left more than four hours to reach the steady state condition at maximum required voltage then switched to the next voltage at least the lower one (to reduce the required time to reach the steady state condition). (3) The measuring parameters collected during each test are; thermocouples temperatures in °C, the heater current in amperes, the heater voltage in volts, the angular velocity of inner shaft. (4) Same steps were recurred with applying new values of heat flux, and angular velocity for inner shaft.

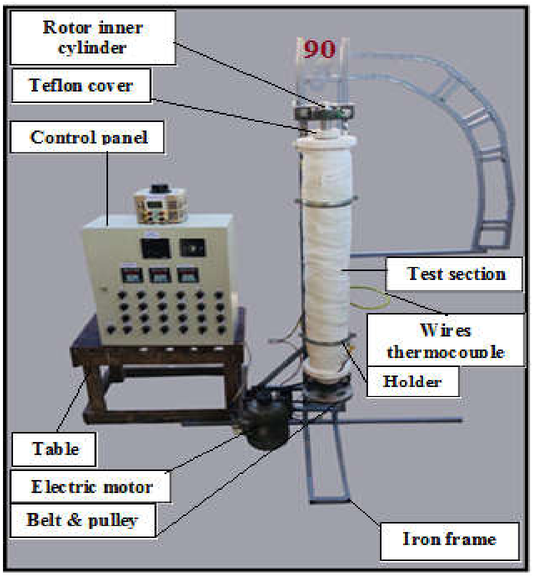

Fig. 1 Schematic Diagram of Experimental Setup.

3. Data Analysis

Simplified steps were used to analyze the heat transfer for the air flow in an annulus when the outer cylinder is subjected to a uniform heat flux. The total input power supplied to the outer cylinder can be calculated by **[2]**:

$$Q_t = V \times I \quad (1)$$

The convection and radiation heat transferred from the outer cylinder is:

$$Q_{cr} = Q_t - Q_{cond} \quad (2)$$

where Q_{cond} is the conduction heat loss which was found from the following equation:

$$Q_{cond} = \frac{\Delta T_{oi}}{ln\frac{r_o}{r_2}/2\pi K_a L} \quad (3)$$

The convection-radiation heat flux can be represented by:

$$q_{cr} = Q_{cr}/A_o \quad (4)$$

Where: A_o is outer surface area of cylinder and equal $(2\pi r_2 L)$.

The local radiation heat flux can be calculated from the expression [16]

$$q_r = F_{1-2}\, \varepsilon\, \sigma[((T_s)_x + 273)^4 - ((\overline{T_s}) + 273)^4] \quad (5)$$

Hence, the convection heat flux at any position is:

$$q = q_{cr} - q_r \quad (6)$$

The radiation heat flux is very small and can be neglected. Therefore, the convection-radiation heat can be equated to the convection heat flux, q.

The local heat transfer coefficient can be obtained by:

$$\mathrm{h_x} = \frac{\mathrm{q}}{(\mathrm{T_s})_\mathrm{x} - (\mathrm{T_b})_\mathrm{x}} \quad (7)$$

The local Nusselt number (Nu_x) can be calculated:

$$Nu_x = \frac{\mathrm{h_x}\ \mathrm{D_h}}{\mathrm{K}} \quad (8)$$

The average values of Nusselt number Nu_m can be calculated by:

$$Nu_m = \frac{1}{L}\int_0^L Nu_x dx \quad (9)$$

The average values of the other parameters can be calculated based on calculation of average outer cylinder surface temperature and average bulk air temperature as follows:

$$\bar{T}_s = \frac{1}{L}\int_{x=0}^{x=L} (T_s)_x dx \quad (10)$$

$$\bar{T}_b = \frac{1}{L}\int_{x=0}^{x=L} (T_b)_x dx \quad (11)$$

$$\bar{T}_f = \frac{\bar{T}_s + \bar{T}_b}{2} \quad (12)$$

All thermodynamics air properties of ρ, μ, v, and k were evaluated at the average mean film temperature ($\bar{T}_f$) **[4].**

4. Results And Discussions

A total of 60 test runs have been carried out on a test rig which consists of an annulus with a rotating inner cylinder and uniformly heated stationary outer cylinder. Eccentricity plays a significant role in the flow and heat transfer behavior in the vertical annulus with rotating or stationary inner cylinder and stationary outer cylinder. The eccentric case is much more complicated than its concentric counterpart, because the flow field is in fact strongly influenced by the geometry of the annulus as well as its orientation. The temperature distribution, local Nusselt number, and mean Nusselt number have been ploted with values of Rayleigh number equal to 14544, 21220, 32675, 43671, 48976, 16534, 28231, 38692, and 49543, and rotational Reynolds number ranges from 0 to 300 at vertical position (θ=90^0) for different eccentric ratio Ɛ=0.33 and 1.6.

4.1 Local Nusselt Number (Nu_x)

The local Nusselt number variation along the outer cylinder surface of open ended annulus is plotted for selected runs in **Figs. 2 to 5**. The **Figs. 2 and 3** show the effect of Rayleigh number on the variation of the Nusselt number along the outer cylinder at Ɛ=0.33, Ɛ=1.6 Re_{Ω}=0 and 300; respectively. The figures show that the local Nusselt number increases with increasing Rayleigh number due to the buoyancy effect. The **Figs. 4 and 5** show the effect of rotation Reynolds number on the variation of the Nusselt number along the outer cylinder at Ɛ=0.33,Ɛ=1.6 Ra=14544 and 48976; respectively. The figures show that the local Nusselt number increases with increasing the rotational Reynolds number as Rayleigh number kept constant because of the strong mixed convection currents which increase with increasing rotational Reynolds number.

As can be seen from these figures that at the inlet annulus, the local Nusselt number decreases sharply to reach a minimum value at a certain point after which the local Nusselt number gradually increases. It is shown that there is no effect on the behavior of local Nusselt number when the value of rotational Reynolds number exceeds the critical value (Re_{cr}=110) because of higher values of Richardson number which indicate the dominant of mixed and natural convection in the heat transfer process.

The local Nusselt number variation along the outer cylinder surface of opened upper end annulus is plotted for selected runs in **Figs. 6 to 9**. The behavior of local Nusselt number variation in the opened upper end eccentric annulus is similar to that in open ended eccentric annulus shown in **Figs 2 to 5**; respectively.

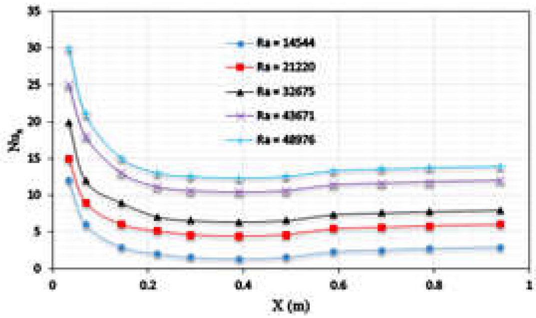

Fig. 2 Variation of local Nusselt number with the axial distance for different Rayleigh numbers, at Re_{Ω}= 0, Ɛ= 0.33.

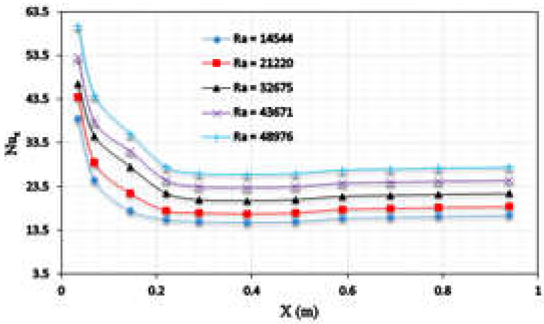

Fig. 3 Variation of local Nusselt number with the axial distance for different Rayleigh numbers, at Re_{Ω}= 300, Ɛ=1.6.

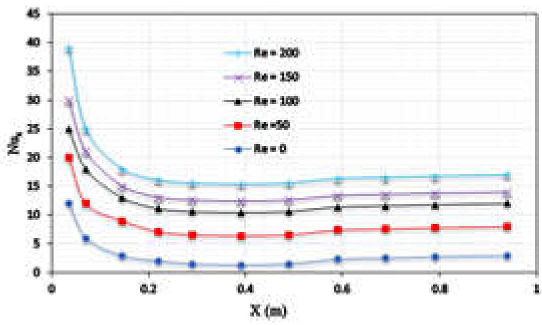

Fig. 4 Variation of Nusselt number with the axial distance for different rotational Reynolds numbers, at Ra= 14544, Ɛ=0.33.

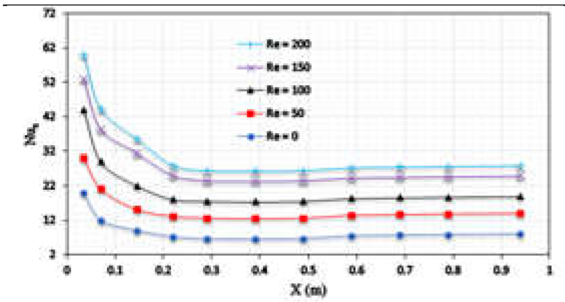

Fig. 5 Variation of Nusselt number with the axial distance for different Reynolds numbers, at Ra= 48976, Ɛ=1.6.

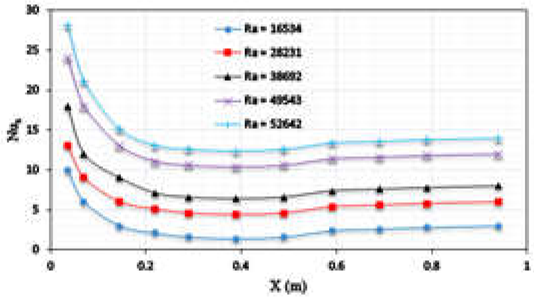

Fig. 6 Variation of local Nusselt number with the axial distance for different Rayleigh number values, at Re_{Ω} = 0, Ɛ= 0.33.

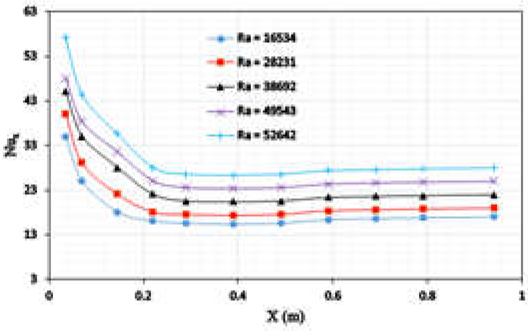

Fig. 7 Variation of local Nusselt number with the axial distance for different Rayleigh number values, at Re_{Ω} = 300, Ɛ=1.6.

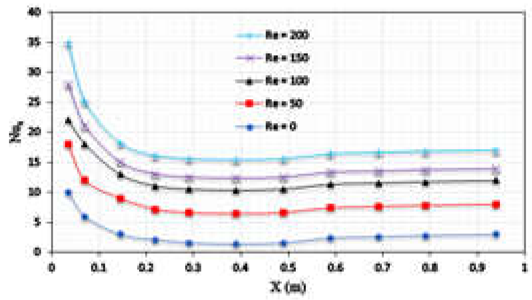

Fig. 8 Variation of Nusselt number with the axial distance for different rotational Reynolds number, at Ra= 16534, Ɛ=0.33.

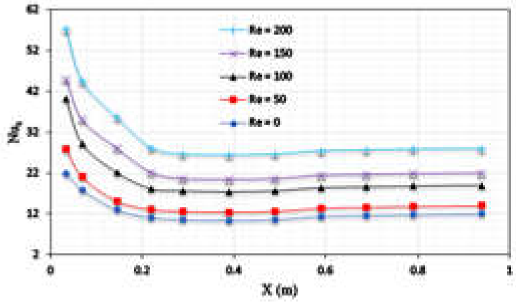

Fig. 9 Variation of Nusselt number with the axial distance for different rotational Reynolds number, at Ra= 52642, Ɛ=1.6.

4.2 MEAN NUSSELT NUMBER

The effect of eccentric ratio on mean Nusselt number (Nu_m) versus Rayleigh number for open ended annulus with stationary inner cylinder and versus Richardson number for open ended annulus with rotating inner cylinder for selected runs is shown in **Figs. 10 and 11**. It is noticed that the mean Nusselt number values increase with the increasing eccentric ratio. Because of the eccentric ratio causes an increase in cross-section area in one side and reduction in the cross-section area in opposite side. So the circulation increases in wide side, temperature gradient increases in narrow side and both increase the heat transfer.

The effect of eccentric ratio on mean Nusselt number (Nu_m) versus Rayleigh number for stationary inner cylinder and versus Richardson number for rotating inner cylinder in opened upper end and seal lower end eccentric annulus is shown in **Figs. 12 and 13**. As can be seen from these figures that the behavior and trend of mean Nusselt number are the same that in open ended eccentric annulus shown in **Figs. 10 and 11**; respectively.

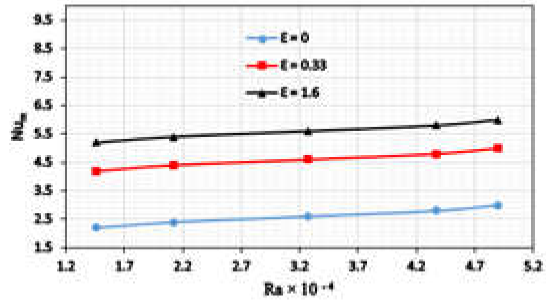

Fig. 10 Variation of average Nusselt number versus Rayleigh number for different values of eccentric ratio.

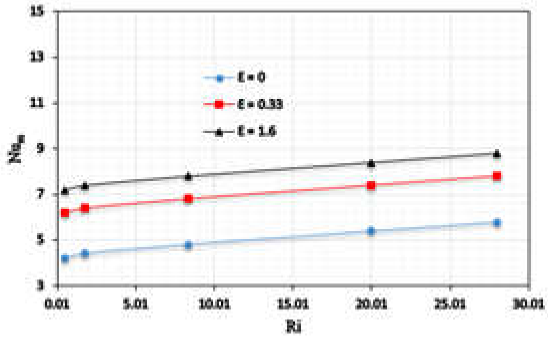

Fig. 11 Variation of average Nusselt number versus Richardson number for different values eccentric ratio.

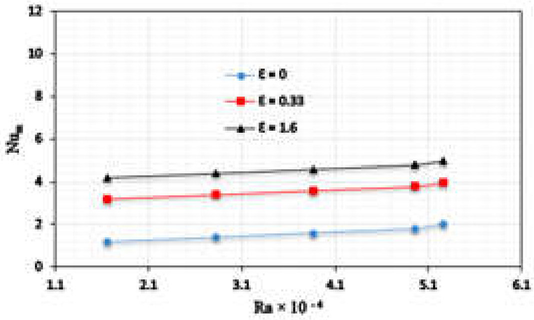

Fig. 12 Variation of average Nusselt number versus Rayleigh number for different values eccentric ratio.

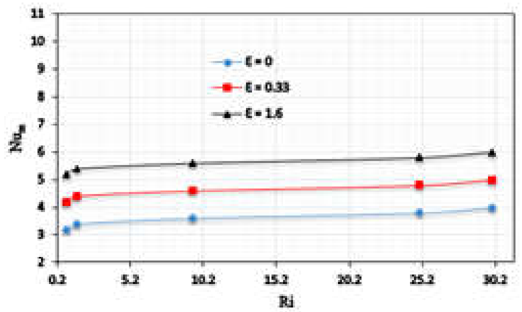

Fig. 13 Variation of average Nusselt number versus Richardson number for different values eccentric ratio.

4.3 Comparison Between Open Ended and Opened Upper End Annulus

The variation of average Nusselt number versus Rayleigh number for open ended and opened upper end eccentric vertical annuli with stationary inner and outer cylinders is shown in **Fig.14**. It is noticed that the Mean Nusselt numbers for open ended annulus are higher than that in opened upper end and closed lower end annulus at the same value of Rayleigh number.

The variation of average Nusselt number versus Richardson number for open ended and opened upper end eccentric annuli with rotating inner cylinder and stationary outer cylinder is shown in **Fig. 15**. It is noticed that the Mean Nusselt numbers for open ended annulus are higher than that in opened upper end and closed lower end annulus at the same value of Richardson number.

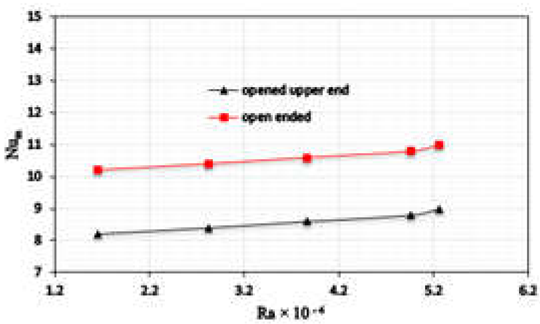

Fig. 14 average Nusselt number versus Rayleigh number at Re_{Ω}=0.

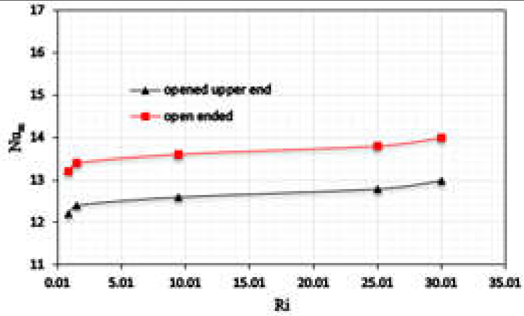

Fig. 15 average Nusselt number versus Richardson number.

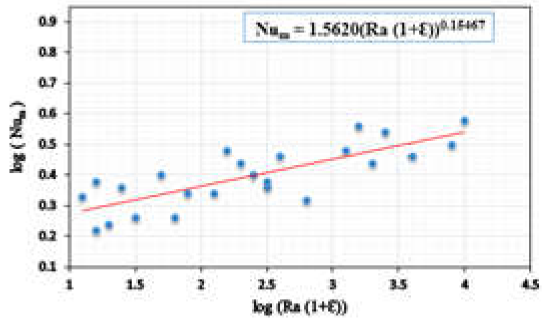

Fig. 16 Logarithmic mean Nusselt number versus logarithmic (Ra (1+Ɛ)).

4.4 Empirical Equations

Four empirical equations for average Nusselt number at the outer stationary cylinder of vertical eccentric open ended and opened upper end annulus with stationary and rotating inner cylinder have been deduced as a function of Rayleigh number and eccentricity ratio for stationary inner cylinder case and as a function of Richardson number and eccentricity ratio for rotating inner cylinder case as shown in **Figs 16 to 19**; respectively. The error resulted from these equations if the measured value of average Nusselt number is compared with the calculated value are 8%, 9%, 7%, and 8%; respectively. All the data as can be seen in these figures are reduction by linearization to give the following equations for stationary and rotating inner cylinders for both cases open ended and opened upper end annulus; respectively:

$$Nu_m=1.5620(Ra(1+Ɛ))^{0.15467} \quad (13)$$

$$Nu_m=1.4652(Ra(1+Ɛ))^{0.1367} \quad (14)$$

$$Nu_m=1.8779(Ri(1+Ɛ))^{0.1324} \quad (15)$$

$$Nu_m=1.7939(Ri(1+Ɛ))^{0.1134} \quad (16)$$

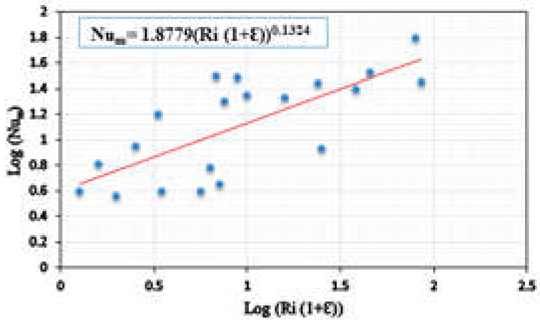

Fig. 17 Logarithmic mean Nusselt number versus logarithmic ((Ri)(1+Ɛ)).

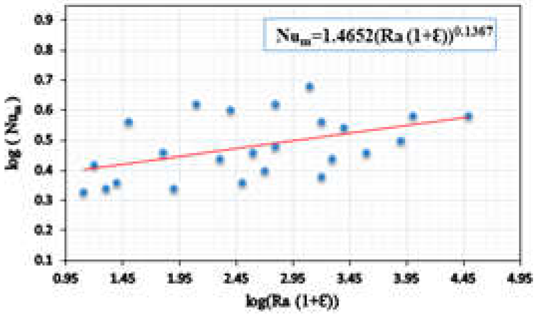

Fig. 18 Logarithmic mean Nusselt number versus logarithmic (Ra (1+Ɛ)).

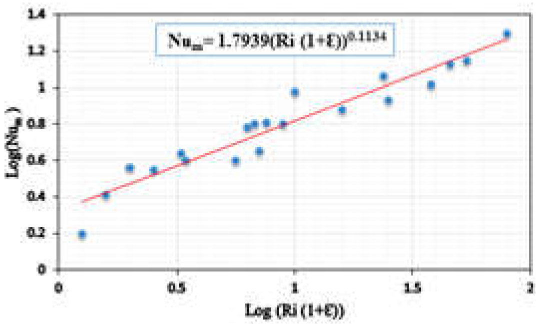

Fig. 19 Logarithmic mean Nusselt number versus logarithmic (Ri(1+Ɛ)).

4.5 Validation the Present Work

The average Nusselt number variations versus $((Re_\Omega/Ra)\times(1+\varepsilon))$ for vertical eccentric annulus with uniformly heated stationary outer cylinder and rotating inner cylinder resulted from the present work is compared with that of the experimental work of **[10]** as shown in **Fig.20**. It is obvious that the behavior and trend of Nu_m for the two works are the same. The difference between the two experimental works may be referred to the difference in thermal boundary condition and the values of ratio L/D_h for both works.

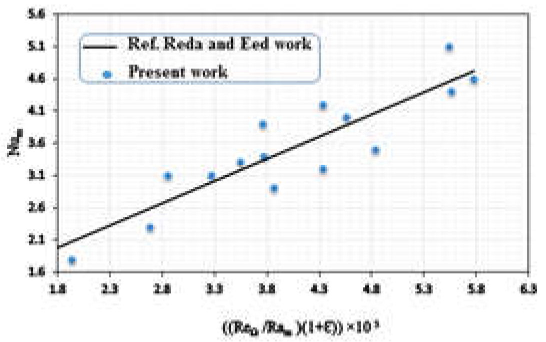

Fig. 20 Comparison between the present work and the Reda and Eed work [9] for the mean Nusselt number versus $((Re_\Omega/Ra)\times(1+\varepsilon))$.

5. CONCLUSIONS

This investigation is concerned with the experimental of Couette and Taylor vortex flows in open ended and opened upper end vertical eccentric annulus with the inner cylinder rotating at constant speed. The forced, natural, and mixed convection were maintained according to the value of Richardson number. For all ranges of rotational Reynolds number, Rayleigh number, and Richardson number taken in the present study, $50 \leq Re_\Omega \leq 100$ for laminar Couette flow ($Re_{cr} = 110$), and $150 \leq Re_\Omega \leq 300$ for turbulent Taylor vortex flow, Rayleigh number $14544 \leq Ra \leq 52642$, and Richardson number $0.23 \leq Ri \leq 30$, the following remarks were concluded.

1. There is no effect on the behavior and trend of local Nusselt number where the rotation Reynolds number exceeds the critical value ($Re_\Omega=110$) because of higher values of Richardson number, this means the dominant mixed convection in the heat transfer process.
2. The heat transfer process in open ended eccentric annulus is better than that in opened upper end annulus by **7.5%.**
3. The mean Nusselt number increases as Richardson number increases.
4. The heat transfer process improves as Rayleigh number increases at constant value of Reynolds number.
5. The heat transfer process improves as rotational Reynolds number increases at constant value of Rayleigh number.
6. Five empirical equations have been deduced for mean Nusselt number as a function of rotational Reynolds number and Rayleigh number for each angle of inclination, in addition to general empirical equation contains all above effective parameters.

NOMENCLATURE

A_o outer surface area of cylinder, (m^2)
C_p specific heat at constant pressure, (kJ/kg.K)
D_h, hydraulic diameters, $2(r_2-r_1)$, (m)
g gravity acceleration, (m/s^2)
F_{1-2} radiation view factors ≈ 1.

h heat transfer coefficient, (W/m^2.ºC)
I current, (ampere)
L length of annulus, (m)
k wave number
q heat flux, (W/m^2)
$\dot{Q}_{conv}$ convection heat loss, (W)
Q_{cond} conduction heat loss, (W)
Q_t total heat power, (W)
T temperature, (ºC)
ΔT_{oi} Difference between average inner and outer lagging surface temperature, (ºC)
r_o radius of outer lagging surface, m
r_1 Inner radius of cylinder, (m)
r_2, outer radius of cylinder, (m)
V voltage, (volt)
e Displacement of inner cylinder axis from outer cylinder axis

Creak Symbols

ν kinematic viscosity, (m^2/s)
Ω Angular velocity of inner cylinder, (rad./s)
η Radius ratio, r_1/r_2
μ Dynamic viscosity, (kg/m.s)
β Coefficient of volume expansion, (1/K)
ρ Density, (kg/m^3)
Δ Difference between two values
σ Stefan-Boltzmann constant $= 5.66\times10^{-8}$ W/m^2 K^4
ε emissivity of the polished aluminum surface=0.09.
κ thermal conductivity of air, (0.6099 W/m.ºC)
κ_a thermal conductivity of asbestos, (0.161 W/m.°C)
λ wave length, m

Dimensionless Groups

Pr Prandtl number $(\mu.Cp/K)$
Nu Nusselt number $(h.D_h/K)$
Gr Grashof number $\left(g\beta(T_w - T_b)D_h{}^3/\nu^2\right)$
Ta Taylor number($\frac{\Omega r_m^{0.5}(r_2-r_1)^{1.5}}{\nu}$)
Re_Ω rotational Reynolds number $(\Omega r_1 D_h/\upsilon)$
Ra Rayleigh number $(Gr.Pr)$
Ri Richardson number (Gr/Re^2)
Ɛ Eccentric ratio (2e/d$_i$)

Subscript

b bulk
f film
I entrance
x local
o exit
m mean
s surface of outer cylinder

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انتقال الحرارة باالحمل في تجويف حلقي غير متمركز عمودي بقضيب معدني أسطواني داخلي دوار

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الخلاصة: تم التحري عمليا لعملية انتقال الحرارة بالحمل الحر , القسري والمختلط في تجويف حلقي غير متمركز عمودي بنهايتين مفتوحتين وبنهاية عليا مفتوحة، ذو اسطوانة خارجية ثابتة مسخنة تسخين منتظم وقضيب معدني اسطواني الشكل داخلي ثابت ودوار. غطت الدراسة مديات الفيض الحرارية ($80\leq q\leq 500$) w/m^2، رقم Re الدوراني $Re_\Omega=0$ (قضيب معدني داخلي ثابت) ويتراوح ($50\leq Re_\Omega \leq 100$) لجريان طباقي ($Re_{cr}=110$) و ($150\leq Re_\Omega \leq 300$) لجريان دوامي مضطرب ، ورقم Ri يتراوح ($0.23\leq Ri\leq 30$). بينت النتائج العملية بان عملية انتقال الحرارة تتحسن كلما زادت نسبة اللاتمركز ,وعملية انتقال الحرارة بالتجويف الحلقي اللاتمركزي ذو النهايتين المفتوحة افضل منها في التجويف الحلقي اللاتمركزي ذو النهاية العليا المفتوحة بنسبة 7.5 % .استنبطت اربعة علاقات تجريبية لمعدل رقم Nu_m كدالة ل ε و Ra للحمل الحر النقي وكدالة ل ε و Ri للحمل المختلط.قورنت النتائج مع البحوث السابقة وقد اعطت نتائج مقبولة جيدة.

الكلمات المفتاحية: تجويف حلقي، انتقال الحرارة بالحمل، قضيب معدني أسطواني، داخلي دوار.