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Heat Transfer Characteristics in a Horizontal Annulus Enclosure with Uniformly Heated Stationary Inner Cylinder and Rotating Outer Cylinder

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ABSTRACT: An experimental and numerical investigations have been carried out to show the effect of rotating outer cylinder on heat transfer and fluid flow for a horizontal annulus filled with air. The rotating outer cylinder was cooled by ambient while the inner cylinder was kept stationary and subjected to uniform heat flux. The dimensionless parameters ranges that studied for this work were rotational Reynold's number ($0 \leq Re_{\Omega} \leq 4000$), Rayleigh number ($2 \times 10^4 \leq Ra \leq 5 \times 10^5$), radius ratio η =3 and aspect ratio Ar=11.59. A computational fluid dynamic package ANSYS fluent (R.15) has been used to achieve the numerical simulations. The validation of present results shows a good agreement with previous works. The results show that the effect of increasing Rayleigh number more than the influence of rotational Reynolds number on average heat transfer coefficient since the increasing percentages were 16.33 and 0.0445 respectively. Empirical correlations have been deduced for the average Nusselt number Nu as a function of Ra only for pure natural convection and as a function of Re_{Ω} and Ra for mixed convection.

KEYWORDS: mixed convection; rotating outer cylinder; annulus; rotational Reynold's number.

INTRODUCTION

Heat transfer convection in a cylindrical annulus configuration with and without axial flow as well as rotating one or both of cylinders became important subject of many researchers because a huge applications that work on base of it. The complexity increases with rotating one or both cylinders wall. The rotating devices that have application importance in a lot of systems among them the rotating heat pipe, heat exchangers, electric machines, clinical oxygenators, pumps, turbines and others applications that use rotating configurations. A lot of papers directed to understand the heat transfer and fluid flow in rotating annulus have been reported in the literature. The researchers Ahmadbadi and Karrabi [1]examined empirically the heat transfer in a channel with non-annular configuration between rotor and stator with grooves distributed circumferentially. The results showed that by increasing rotor speed from 300 rpm to 1500 rpm, the heat transfer of the stator and rotor increased by 30% and 45% respectively, as well as they observed that the axial distribution of local Nusselt number on the rotor's surface was uniformly more than stator's surface. Dellil and Azzi [2] worked a code CFX1 and SST model to study the heat transfer in turbulent flow for annulus between stator and rotor. The investigation covered cases according to rotation speed of the rotor and magnetic gap value. They observed that the Nusselt number of the rotor rises with increasing number of revolutions and air gap. Mauwafak et al.[3] studied the vibration effect on the mixed convection heat transfer in the entrance region of concentric vertical annulus with rotating inner cylinder and stationary heated outer cylinder of 0.365 as radius ratio with a heated outer cylinder of 1.2 m. The ranges of Reynold number, Taylor number, heat flux, and frequency were $514 \leq Re \leq 1991$, $10.44\times10^4 \leq Ta \leq 82.23\times10^4$, ($468 \leq q \leq 920$) W/m^2, and Fr=32 & 77 Hz; respectively. They showed that the values of local Nusselt number increase as the heat flux increases at the natural frequency. Nair et al.[4] used model of finite volume for two dimensional steady state coupled with approach of pressure-velocity coupling to simulate natural, mixed convection heat transfer in an annulus with rotating inner cylinder at variable angular velocities to satisfy the natural and mixed convection. Three temperature differences 5K, 50K and 90K were represented the cases of Ra. They showed that with increasing Ra number, the heat transfer was enhanced but increasing power consumption. Through the range $1 \geq Ri \geq 0.01$, heat transfer reduces with increasing rotation velocities and increasing power consumption. For the range $Ri \leq 0.1$, Ra number increasing centrifugal force because centrifugal forces become dominant. Akeel et al.[5] investigated the heat transfer characteristics in an annulus with rotating inner cylinder and outer cylinder kept stationary and uniformly heated. The parameters were studied as follow, rotational Reynolds number (Re_{Ω}) for inner cylinder, and Rayleigh number (Ra), annulus angle of inclination from horizontal to the vertical and eccentric ratio. They showed that Nusselt number improves as the angle of inclination deviates from horizontal to vertical position. Rajkumar et al [6] carried out an experimental and a simulation by ANSYS CFX 14, for a vertical annulus with outer cylinder rotating and the stationary inner cylinder was heated uniformly. The results showed that the Nusselt Number increases progressively with increasing rotational parameter, thereafter a strong decreasing in Nusselt number. Osman [7] worked a Numerical investigation of laminar mixed convection in a square cross-sectioned cylindrical annular enclosure at variable ranges of Richardson number, ri/d ratio and Reynolds number at Prandtl number value equal to 1. He found that the mean Nusselt number $\overline{Nu}$ demonstrates a monotonically decreasing manner with increasing Ri whereas the mean Nu number increases with increasing ri/d and Re which is constant for scaling estimation.

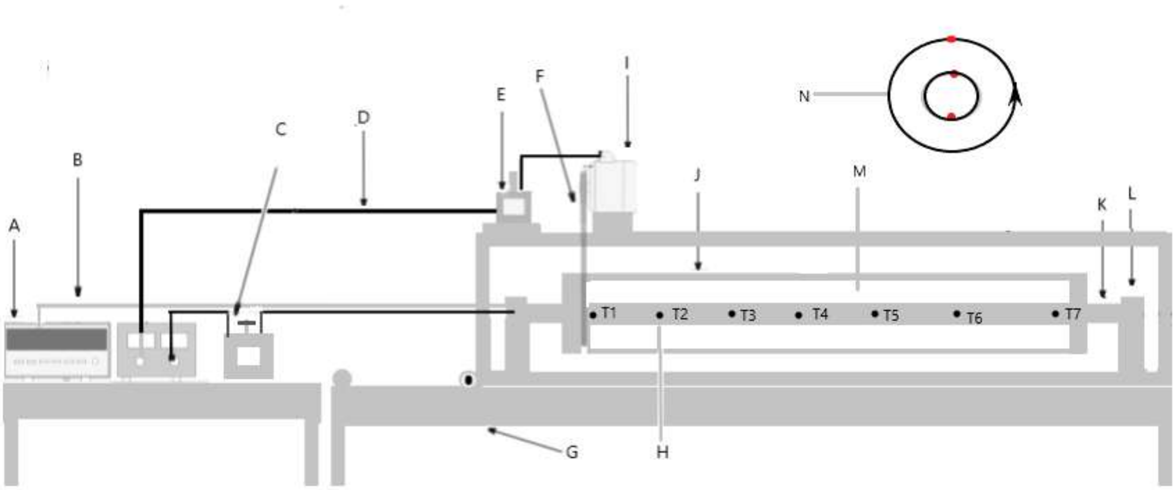
From the above literatures review it can be noticed that most of the studies directed to describe the fluid flow and heat transfer in rotating inner cylinder in an annulus horizontally or inclined and vertically with axial flow (open ended) in comparison to rotating outer cylinder in horizontal annulus with closed ends as studied by present work. Furthermore, the previous problems were studied only experimentally or numerically. The principle attention of the current work is to investigate the effect of rotating outer cylinder experimentally as well as numerically on convection heat transfer and fluid flow in horizontal enclosure formed by two concentric cylinders filled by air, in which the fixed inner cylinder is heated uniformly by heat flux while the rotating outer cylinder is cooled by ambient temperature.

EXPERIMENTAL PART

The experimental rig schematic is shown in Fig.1. The rig includes two horizontal concentric cylinders formed the annular gap. A polished hollow steel cylinder with 33 mm for outer diameter of the inner with a thickness of 4 mm and length of 400 mm. The material of enclosure cylinder is Aluminum with inner radius 51 mm, cylinder thickness of 5 mm and length 400 mm. The radius ratio (η) of gap between concentric cylinders is 3 which mean the size of annulus is large gap . A cartridge heater is used to heat inner cylinder uniformly, which put inside it. Ends of heater are insulated by Teflon rods of same dimeter of inner cylinder walls to prevent heat loss from heater faces. The inner cylinder and their parts have been bolted to a frame with fixing bearing (see Fig.1). The inner steel cylinder is aligned with the outer cylinder by two bearings inside the end cups. The voltage supply to heater regulated by electrical transformer (TDGC CHINA SHANGHAI) to adjust the input heat flux as required.

In one ends of outer cylinder a pulley installed to be driven by belt drive to convey motion from three phase AC motor (HEW HERFORD Co.). The control of speed rotation of the AC motor is achieved by VFD Variable Frequency Drive, (Telemecanique, Electronic, Japan). Care is taken to arrange the driver and driven pulleys close to each other such that the central distance between the pulleys is 100 mm. The measured speed rotation of the rotating enclosure is achieved by a stroboscope (Lutron Technologies Taiwan Ltd).

The number calibrated thermocouples of type T are thirteen. They are eight thermocouples distributed in opposite lines in upper and lower surface of inner cylinder in equidistantly locations. The thermocouple wires were taken out from one ends of the inner cylinder. Four thermocouples are located on surface of rotating outer cylinder and one thermocouple used to measure ambient temperature. The estimated error for calibrated thermocouples are 0.175 ◦C. A multi-channel temperature meter system (Applent AT4524 China Ltd) is used to record temperature data.



A:Data Acquisition System, B: Thermocouple Wires, C: Dc Power Supply (UPS), D: Electric Supply Wires, E: Variable Frequency Drive, F:Pully and Belt Drive, G:Iron Frame , H: Thermocouple locations , I: Three Phase AC Motor, J:Cross section for rotating outer cylinder, K:Stationary Heated Inner Cylinder, L: Fixed Bearing, M:Annular gap , N: thermocouples in annulus (side view).

Figure 1. Experimental apparatus Schematic

EXPERIMENTAL ANALYSIS PROCEDURE

Mean temperature difference of heated cylinder is calculated by equations[6]:

$$\Delta \mathrm{T_{avg}} = \frac{\sum_{j=1}^{N} \mathrm{Tj}}{\mathrm{N}} - T_f \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

Where N=8

Where T_f is the fluid temperature evaluated from:

$$T_f = \frac{\bar{T}si + \bar{T}so}{2} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

Where

$\overline{\mathrm{T}}\mathrm{si}, \overline{\mathrm{T}}\mathrm{so}$ are average surface temperature of inner and outer surface respectively , K

The rotational Reynolds number Re_Ω is computed from:

$$Re_\Omega = \frac{\omega r_0 d}{\nu} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

Where

ω : Rotational speed, rad/s

r_o, r_i: Outer radius and Inner radius respectively, m

d : Annular gap, $(r_o - r_i)$, m

ν : Kinematic viscosity, $\left(\frac{\mu}{\rho}\right)$, m^2/s

ρ : Density, kg/m^3

μ: dynamic viscosity, Kg/m. s

The average convection heat transfer coefficient are defined respectively as:

$$\bar{h} = \frac{Q}{A_i * \Delta \mathrm{T_{avg}}} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

Where

Q: Power input, =Ampere*Voltage , W

A_i : Inner cylinder surface area of heated cylinder, m^2

Mean Nusselt Number ,

$$\bar{N}u = \bar{h} * \frac{d}{k_f} \quad \text{.................................} \quad (5)$$

kf : Fluid Thermal conductivity, $W/m\,.K$

NUMERICAL PART

The governing equations of continuity, momentum, and energy transport in cylindrical coordinates are [6]:

$$\nabla\,.\,u = 0 \quad \text{..............................} \quad (6)$$

$$\frac{\partial u}{\partial t} + (u.\nabla)u = -\nabla P + \frac{1}{Re}\nabla^2 u + \frac{Ra}{prRe^2}\theta[cos\emptyset e_r - sin\emptyset e_\emptyset] \quad \text{..............................} \quad (7)$$

$$\frac{\partial u}{\partial t} + (u.\nabla)\theta = \frac{Ra}{prRe^2}\theta \quad \text{..............................} \quad (8)$$

Where

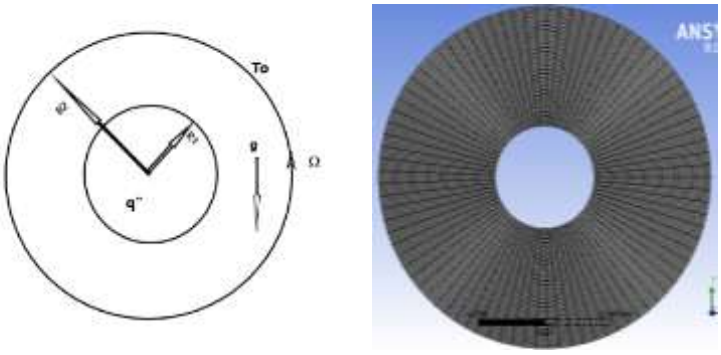
u , v: dimensionless velocity vector in radial and angular directions respectivly
P : dimensionless Pressure
Pr : Prantdl number ν/α
$\emptyset$: Angular coordinate
θ : Dimensionless temperature
$e_r, e_\emptyset$: Unit vector in the radial and angular directions respectivly
g : Acceleration due to gravity, m/s^2
Gr : Grashof number, $g\beta\Delta Td^3/\nu^2$
Ra : Rayleigh number = $Gr.Pr$
α : Thermal diffusivity, m^2/s
β : Isobaric cubic expansivity of fluid, $1/K$

The boundary conditions for normal pressure gradient and velocity components are zero and no slip conditions respectively, which are assumed for interfaces of fluid-wall. The boundary conditions can be written in mathematic form, as bellow:

$$u = v = 0\,, \theta = 1\,, at\ \ r = ri \quad \text{..............................} \quad (9)$$

$$u = 0\,, v = 1\,, \theta = 0\,, at\ r = r_o \quad \text{..............................} \quad (10)$$

The boundary conditions and equations above which are solved numerically, by using the Computational Fluid Dynamics package which is known ANSYS Fluent R.15 [8]. The used physical problem in the simulation and corresponding to experimental work as well as computational mesh are shown in Fig. 2 a,b. The two-dimensional computational domain was discretized using structured non-uniform quadrilateral cells created [9]. Fine grids were used to get the required accuracy for temperature and velocity profiles in boundary layers for both cylinders.



a **b**

Figure 2. (a) The physical domain (b) sample of the computational mesh

NUMERICAL SIMULATION PROCEDURE

The assumptions for physics model for two-dimensional horizontal annulus that the inner cylinder is subjected to a condition of uniform heat flux as thermal condition and fixed as momentum condition. The outer cylinder is assumed to rotating at uniform angular velocity and cooled by ambient air temperature [10]. Density change in the air as working substance is neglected everywhere except in the buoyancy, and all the other physical properties of the air are considered constant (Boussinesq approximation). Viscous dissipation in the energy equation is also neglected. The validity of Boussinesq approximation for this kind of problem has been reported earlier in [1^{1}]. SIMPLE method is used for velocity and pressure coupling. The factors of under relaxation method that used in the current work are 1 for energy, 0.3 for pressure and 0.7 for momentum. The solution Convergence less than 1×10^{-5} was set to be achieved when maximum of all the residues reaches this value. As validation part, the numerical studied work solution of heat transfer convection in concentric cylinders assumed in the same boundary condition for the experimental work except in two-dimension horizontal location. The numerical solutions that worked in a problem geometry without flow inlet and exit to the annulus (enclosure). The numerical simulation of local temperature compared to experimental local temperature showed that the percentage error less than 2.5% as summarized in Table1. A grid dependency test has been worked to reach to optimum grid for simulations program as shown in fig 3.

Table 1. The numerical local temperature compared to experimental results.

Speed (RPM)	Theoretical Temperature (K)	Experimental Temperature (K)	Error percent
0	300	294	-2%
30	302.5	295	-2%
60	303	296	-2%
120	300	295.55	-1%
180	298	296.4	-1%
300	296	293	-1%

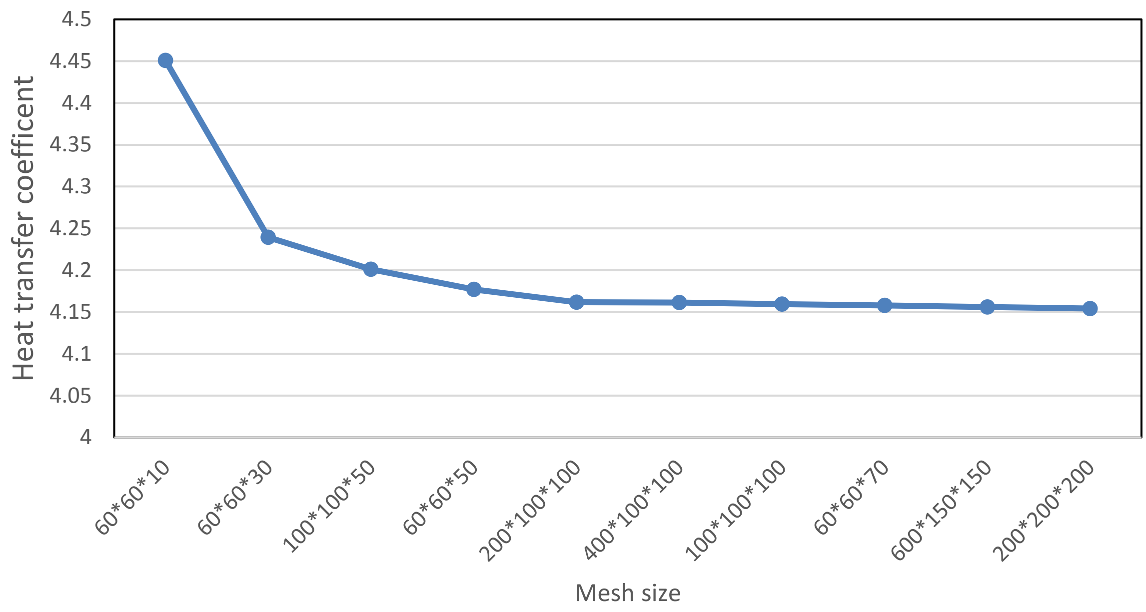


Figure 3. Grid dependency test

EXPERIMENTAL RESULTS

As validation part for study, the experimental results for natural convection and impact of rotation are compared with results of Zhidao work [12]. The results comparison are graphed in Fig. 4 which are showed a good agreement. The investigation worked with variables outer cylinder rotational Reynolds numbers (0 - 1200) as ranged by Zhidao.

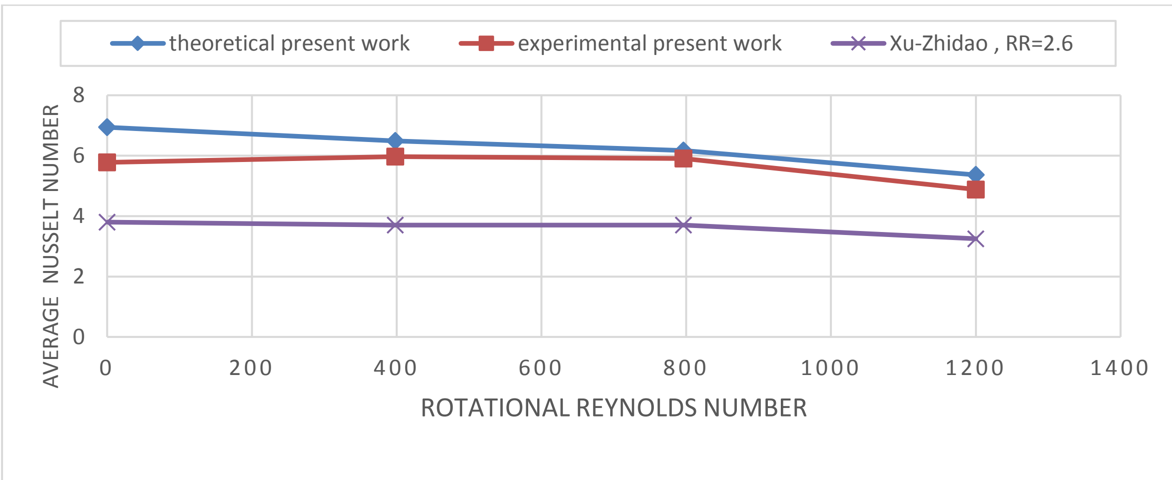


Figure 4. Comparison the present work with work of Zhidao at Ra=10^5

Figure 5, shows the effect of outer cylinder rotational Reynolds number on temperature difference at different Rayleigh numbers to get understanding for trend of Nusselt number with outer cylinder rotational Reynolds number in fig.4. From fig.5 observed that when temperature difference is increase that leads to drop in heat transfer coefficient(fig.4) at low rotational speed.

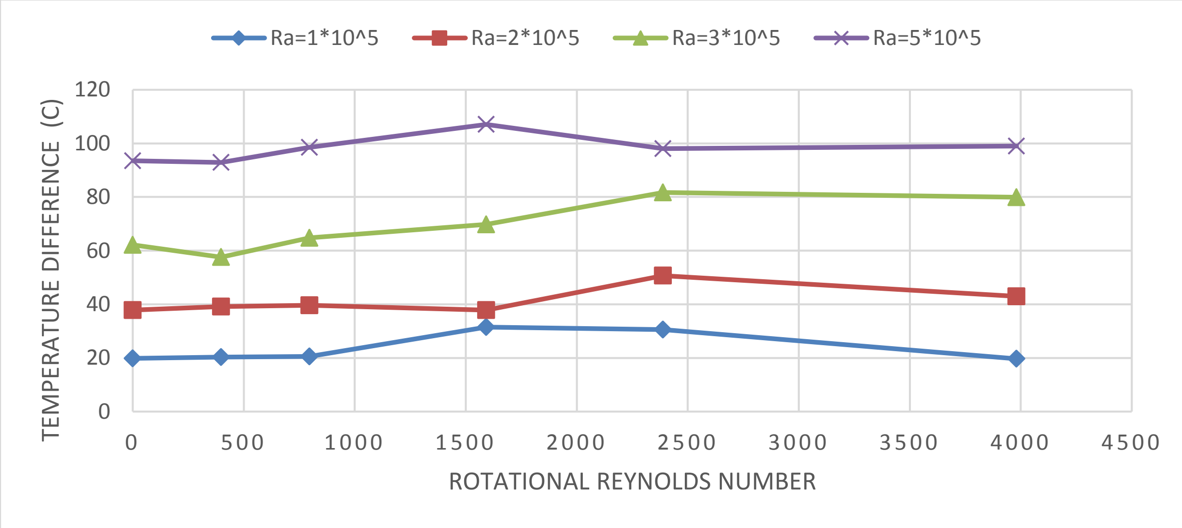


Figure 5. Effect of rotational Reynolds number on temperature difference at different Rayleigh numbers

The effect of heat flux of inner cylinder on average Nusselt number at $Re_{\Omega} = 0$ is shown in Fig.6. It is observed that as heat flux increasing the average Nusselt number increases directly.

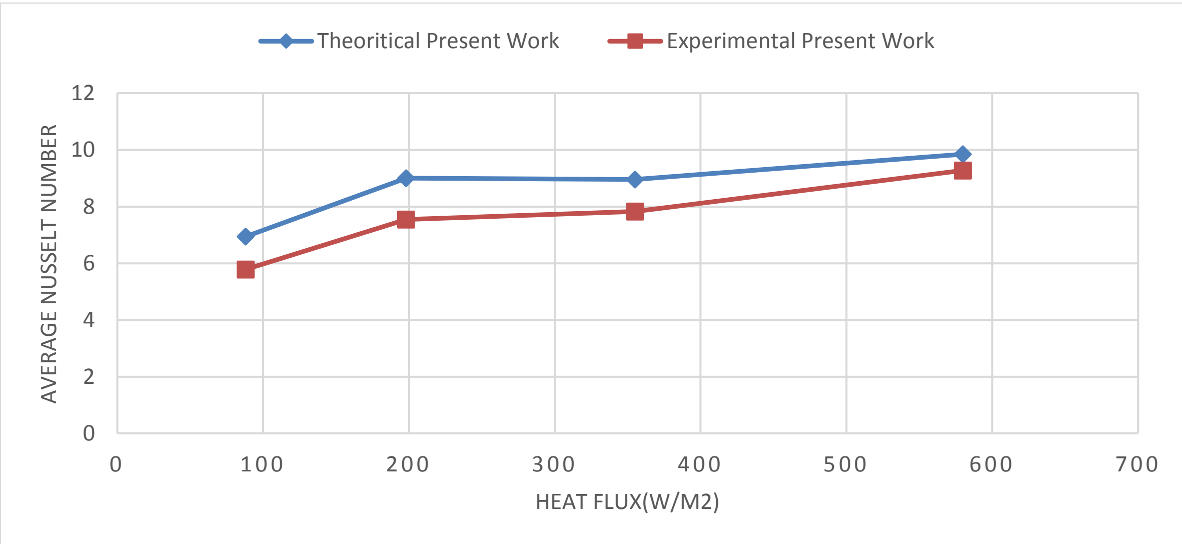


Figure 6. Effect of heat flux for inner cylinder on Nusselt number at $Re_{\Omega} = 0$

The trend of heat transfer during changing outer cylinder rotational Reynolds number, is shown in Fig.7 there are three trends, the first, free convection heat transfer at $Re_{\Omega} = 0$. The second, at low rotational Reynolds number, nearly heat transfer is close to that of first trend, but above certain value of Re_{Ω} number it drops until reach a certain value of rotational Reynolds number $Re_{\Omega} = 2388$ due to competition between buoyancy and centrifugal force, and third, there is increasing in heat transfer after $Re_{\Omega} = 2388$, due to centrifugal force dominated.

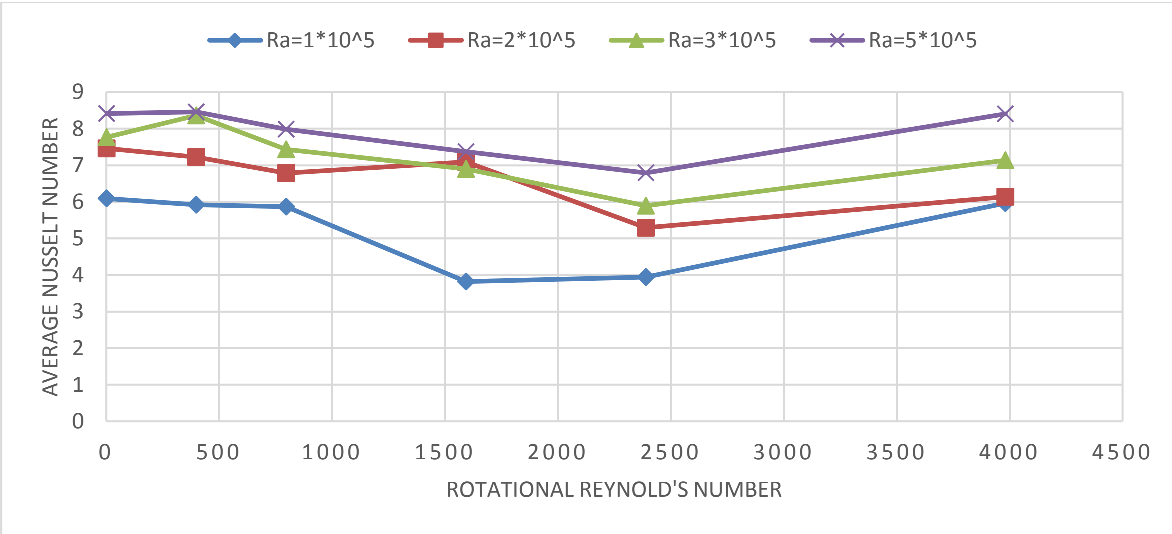


Figure 7. Nusselt number versus rotational Reynolds number at different Rayleigh numbers, Radius ratio=3

The experimental correlation that conducted for range $\boldsymbol{Re}_{\Omega}$=0 is

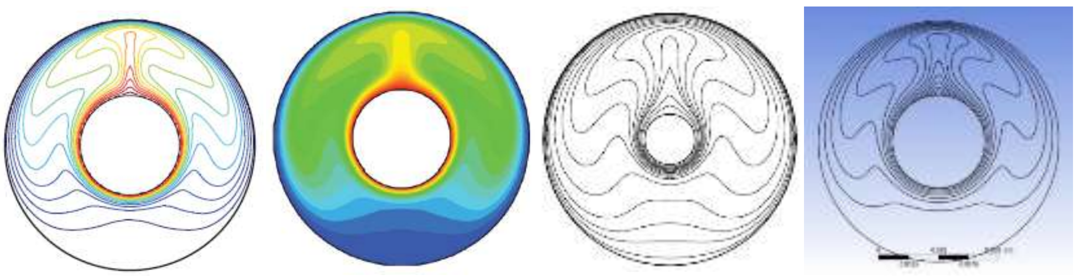
$$Nu= 0.6359(Ra)^{0.2479} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (11)$$

And the correlation that conducted for range $0 < Re_{\Omega} < 4000$ is given as follows:

$$Nu=1.705\ (Ra/Re_{\Omega})^{0.1314} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (12)$$

THEORETICAL RESULTS

In numerical investigation for predicting the effect of rotation outer cylinder on heat transfer and fluid flow characteristics, the studied ranges of the dimensionless parameters as follow, rotational Reynolds number ($0 \leq Re\Omega \leq 4000$), and Rayleigh number ($2 * 10^4 \leq Ra \leq 5 * 10^5$). Firstly, numerical simulation of stationary cylinders (Re_{Ω} =0) was worked and compared with simulations results of Abu Nada et al [13], Habibis Saleh et al [14] and Zhidao as in Fig. 8. The results showed a good agreement with these simulations, so the numerical model is appropriate for the present physic problem.



Abu Nada et al | Habibis Saleh et al | Zhidao | present work

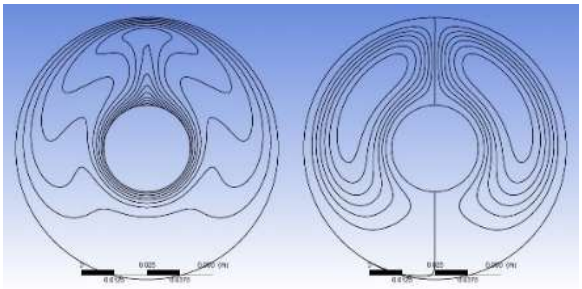
Figure 8. Validation of theoretical part of study isotherms contours at Re_{Ω} =0, Ra =10^5

Figure 9 shows the effect of outer cylinder rotational Reynolds number Re_{Ω} and Rayleigh number Ra on the contours of isotherms and streamlines. For natural convection Re_{Ω}=0, as shown Fig 9 a and b, the velocity pattern is known as two-kidney for both Ra number. As Re_{Ω} increasing, the streamlines contours begin to move in the direction of the motion of the outer cylinder (counter clockwise), as shown Fig 9 c,d,e,f. However, any cell on the right of the inner cylinder is moving upwards, and any cell on the left of the inner cylinder is moving down by viscous action of the rotating outer cylinder. Thus, the isotherms contours move in the opposite direction to the rotating outer cylinder. This shift continues as rotational Reynolds numbers increase until the two-eddy flow at (Re_{Ω}=400)as in Ra=2*10^4 are transformed into one-eddy flow at (Re_{Ω}=800) as in Ra=2*10^4

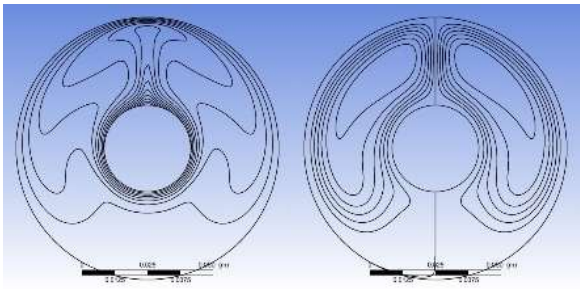
and then to zero-eddy flow at (Re_{Ω} =1600)as in Ra=2*10^4.While at Ra=1*10^5 observe some delay at Re_{Ω} in transforming from two eddy to one eddy and to zero eddy. Thereafter there are formation like concentric circles approaching Couette flow around the inner cylinder at $Re_{\Omega} = 2400$ at Ra=2*10^4 while observe eccentric circles around the inner cylinder and small plume for isothermal plots due to high Re_{Ω} as shown in Fig.k,l.

Ra= 2*10^4 Ra= 1*10^5

Isotherms **Streamlines** **Isotherms** **Streamlines**

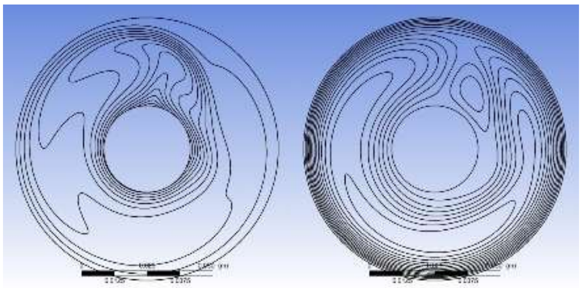


a

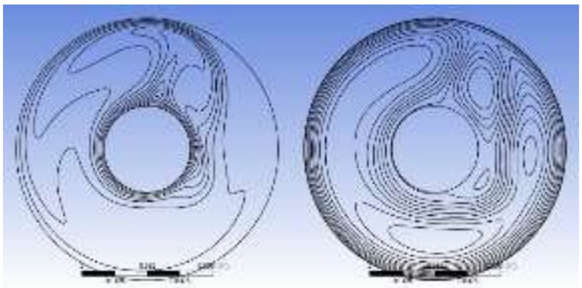


b

$Re_{\Omega} = 0$

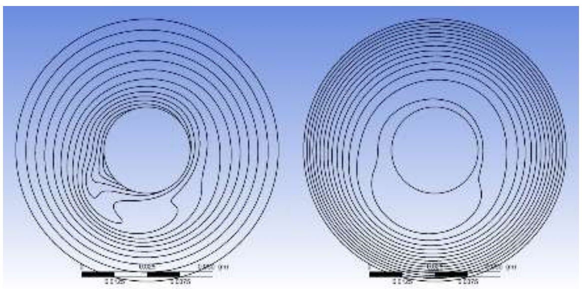


c

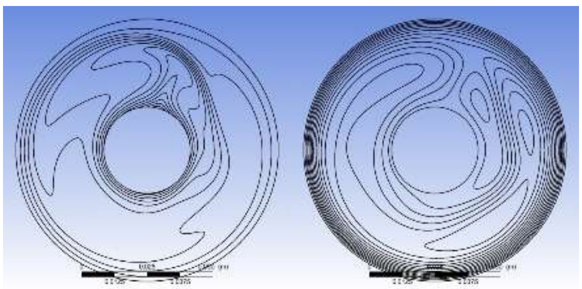


d

$Re_{\Omega} = 400$

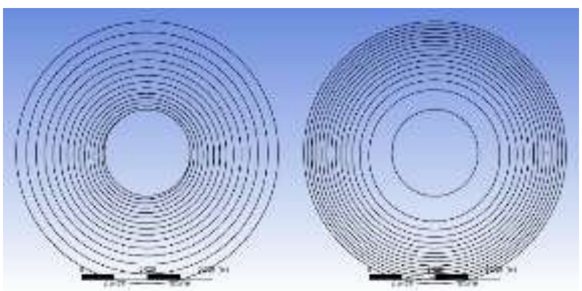


e

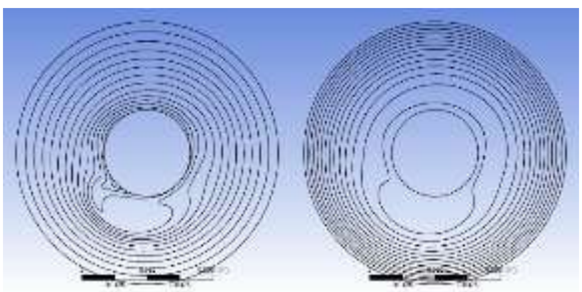


f

$Re_{\Omega} = 800$

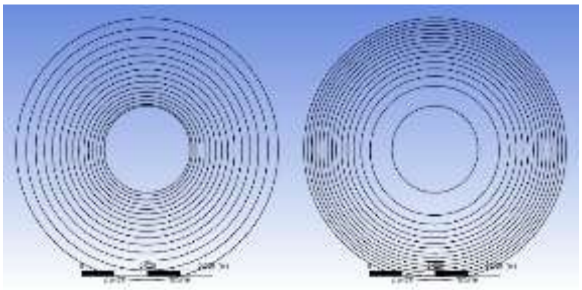


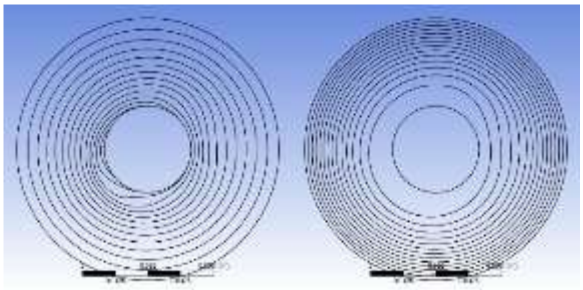
g



h

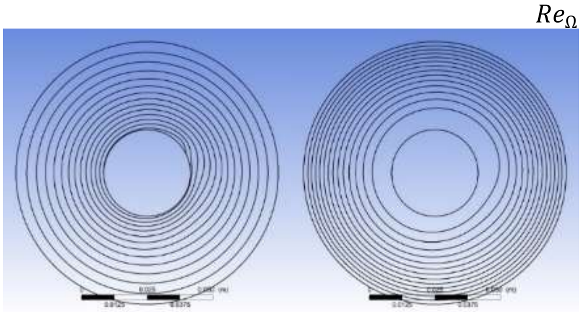
$Re_{\Omega} = 1600$

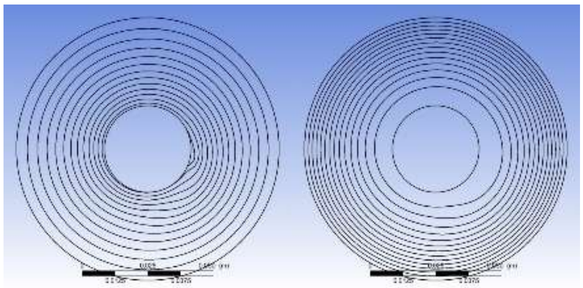




i j

$Re_{\Omega} = 2400$





k l

$Re_{\Omega} = 4000$

Figure 9. Isotherms and Streamlines contours at radius ratio=3

CONCLUSIONS AND RECOMMENDATION

Numerical and experimental investigations have been worked to knew the heat transfer characteristics in a cylindrical horizontal annulus of closed ends with a stationary heated uniformly inner cylinder while the outer cylinder was rotating and cooled by ambient. The concluding remarks from the study of rotating speed outer cylinder on convection heat transfer were as below:

1. The trend of heat transfer and fluid flow results agreed well with that of the previous works.
2. The effect of increasing Rayleigh number more than the influence of rotational Reynolds number on average heat transfer coefficient since the percentages were 16.33 and 0.0445 respectively.
3. Three regions of Streamlines plots can be observed through increasing rotating speed of outer cylinder according to eddies numbers from rest to maximum speed : two eddy flow, one and zero-eddy flows.
4. The numerically simulation results have provided a good explanations to the experimental results.
5. The correlation that conducted for range Re_{Ω}=0 is Nu= 0.6359(Ra) $^{0.2479}$
6. The correlation that conducted for range $0 < Re_{\Omega} < 4000$ is given as follows: Nu=1.705 (Ra/Re_{Ω}) $^{0.1314}$

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