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Transient Free Convection between Two Horizontal Concentric Cylinders

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ABSTRACT

Natural convection was simulated in this work in a concentric ring between two horizontal concentric cylinders with an inner radius Ri and an outer radius Ro, with a radius ratio of RR = Ro / Ri (1 / 1.5, 1/2) the blank is filled Toroid with an air of Prandtl number Pr = 0.71. Interfering with the coordinates fitted to the object using the time-lapse method. An analytical expression is used to prepare the coordinate conversion from the physical domain to the arithmetic field. Rayleigh numbers 10^4, (1, 2, 3, 4, 5, 6, 7, 8)10^5 and various inclination angles of the inner cylinder, ($0 < \theta^{\circ} < 360$), are also described numerically. The measured flow and temperature fields are depicted as streamlined, equal lines with Nusselt number changes. Inflow patterns and heat fields, both the diameter ratio and the Rayleigh number have been discovered to be extremely important. The angle of inclination of Rayleigh numbers does, without a doubt, have a nominal effect on heat transfer.

Keywords: convection; concentric ring; square inclined enclosure; horizontal concentric cylinders.

1. INTRODUCTION

Natural heat transfer rate in annuli occurs in a variety of industrial settings, including nuclear, solar, and thermal storage systems, as well as various fields of electric energy. When a fluid is present between two circular cylinders, either concentric or eccentric, at different fixed temperatures, a complicated buoyancy-driven flow is created in the presence of a constant gravitational force. If the temperature difference between the cylinders is minimal, diffusion can control the energy transfer. As a result, in such conductive flow regimes, the average Nusselt number remains typically constant for all Rayleigh numbers under a given critical value. Convection occurs over this important Rayleigh number, causing heat plumes to appear in the annular gap. Natural convection within annular vacuoles has been studied extensively and given much clarification in the literature as explained by the authors in [1-5]. Looked into this issue in the concentric rings, while some are [6-15] investigated the effect of deflection on heat transfer for different conditions and numerical methods. Using a conduction boundary-layer model, Briefly, Kuehn and Goldstein [6] Relation to the prediction of heat transfer with normal convection from a horizontal cylinder to a liquid within a cylindrical container under semi-static conditions. Ratzel et al. [7] used a finite element method to investigate free convection in concentric and eccentric annular areas. Yao [8] used turbulence techniques to investigate unconstrained convection in eccentric rings. Projahn et al. [9] used the implicit finite difference approach with curvilinear coordinate transformations appropriate for the body natural convective flows in eccentric annuli numerically. The recorded findings were compared to Kuehn and Goldstein's [6] Both vertical and horizontal differences yielded similar results. Feldman et al. [10] used bipolar coordinate transformations to estimate the evolving an eccentric annular vertical duct's flow and thermal fields. Using the one-dimensional finite element approach with a bipolar coordinate transformation and pseudospectral algorithms, Darrell and Roger [11] Within an eccentric toroidal distance between two thermally heated circular cylinders, the free convection heat transfer was numerically investigated. Ho et al. [12] Heat transfer by free convection in eccentric rings with mixed boundary conditions was investigated, and a connection between the Nusselt and Rayleigh numbers was proposed. Making use of the variable separation method, Hwang and Jensen [13] The eccentric ring's convective laminar dispersion flow was investigated. Koichi et al. studied natural convection heat transfer in eccentric horizontal loops between a heated outer oval tube and an inner cylindrical tube cooled with different orientations [14]. Hosseini et al. [15] studied convection in an open-ended vertical anomaly loop and discovered that the best possible eccentricity for optimum heat removal exists. Within porous cylindrical rings, natural convection occurs is critical in a variety of industrial applications, including underground cable networks, thermal energy storage, and thermal insulation. In the literature, great focus and attention was given to the condition of the concentric porous ring. Caltagirone [16] was the first to use the Christiansen effect to visualize isotherms in a porous cylindrical layer in an experimental analysis of a porous ring to a constant radius ratio. Caltagirone can only control the two-cell flow mechanism, so it was inferred that multicellular structures do not occur with increasing Rayleigh count. Later, Fukuda et al. [17] used the finite difference approach to show three-dimensional results for an inclined annulus.

Rao et al. [18, 19] used the Galerkin the horizontal annulus in two and three dimensions is investigated using the Galerkin method. Standard perturbation expansion methods are used. and the Galerkin process, Himasekhar and Bau [20] investigated the action of bifurcation phenomena. Caltagirone's [16] Charrier-Mojtabi et al. [21] replicated the visualization experiments and verified the presence of two-dimensional four-cell flow structures.

Barbosa Mota and Saatdjian [22, 24] tested the effect of radius ratio on the conversion of the flow system and stability by using numerical methods. They demonstrated that the device relies on the Rayleigh number to converting a two-cell flow to a four-cell flow, regardless of whether it rises or falls. Bau et al. [25] used the finite difference technique to investigate the reduction in heat transfer by using an eccentric geometry, and Bau, H. H. [26] used the standard perturbation expansion technique to investigate the reduction in heat transfer by using an eccentric geometry. Himasekhar, K., and Bau, H. H. [27] developed Nusselt number-Rayleigh number and geometrical parameter correlation using a boundary-layer technique. The flow pattern in these experiments was limited to a bicellular two-dimensional flow pattern, which is clearly not the only flow pattern that can occur in practice. Barbosa Mota and Saatdjian [28] studied four-cellular flow conditions in an eccentric cylinder and discovered that eccentricity decreases heat for a given radius ratio and Rayleigh number, there is a loss, which can significantly alter the flow system. Using a high-order compact finite difference technique, Barbosa Mota et al. [29] investigated heat transfer by natural convection in an elliptic annulus with a porous medium. Hussein et al. [30] Studied the enclosure heat transfer characteristics numerically using a rectangular model enclosure. Meanwhile Alguboori et al. [31] studied the elliptic enclosure with circular heat source. Both free and forced convection in enclosure was studied as well [32, 33]. Enclosure fitted with porous medium was studied to show the effect of porosity on the heat transfer characteristics of enclosure [34, 35]. Several researchers studied the heat convection [36-42] there work includes a practical investigation of natural convection heat transfer in a concentric vertical cylinder and the effect of forced vibration on heat transfer augmentation. The findings reveal that the local heat transfer coefficient is clearly affected by the amount of heat input and axial distance of the cylinder, indicating a positive relationship with the first and an inverse relation with the latter, while an obvious increase is observed in the local Nusselt number along the cylinder axis from bottom to top.

In this study, the physical problem is addressed. Simply put, it is a horizontal concentric ring system with a radius ratio and an inner radius Ri and an outer radius Ro. The annular void is filled with air. First, when time t = 0, the inner and outer cylinder walls are thought to be at similar temperatures. Over time the temperature increases and thus, this temperature change causes the system to become unstable, resulting in a laminar flow.

2. PHYSICAL PROBLEM

The physical problem considered in this study is shown in Figure (1-2). Simply, it is a system of the concentric with an inner radius of Ri and an outer radius of Ro, a horizontal annulus is formed, with radius ratio RR = R_o/R_i. The annular space is filled with an air of Prandtl number $P_r = 0.71$. Firstly, the inner and outer cylinder walls are believed to be at equal temperatures in time t = 0. Then, as the time progresses t > 0, the inner wall is suddenly heated up to Th, the temperature was then maintained at that stage, with the assumption that Th > Tc. Because of the temperature rise, the system becomes unsteady, resulting in a flow buoyant.

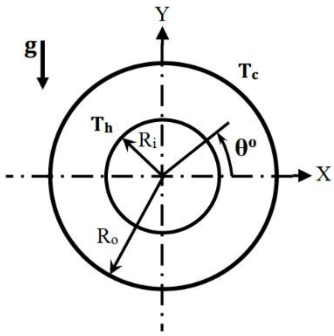


Figure 1: Schematic diagram of the physical problem.

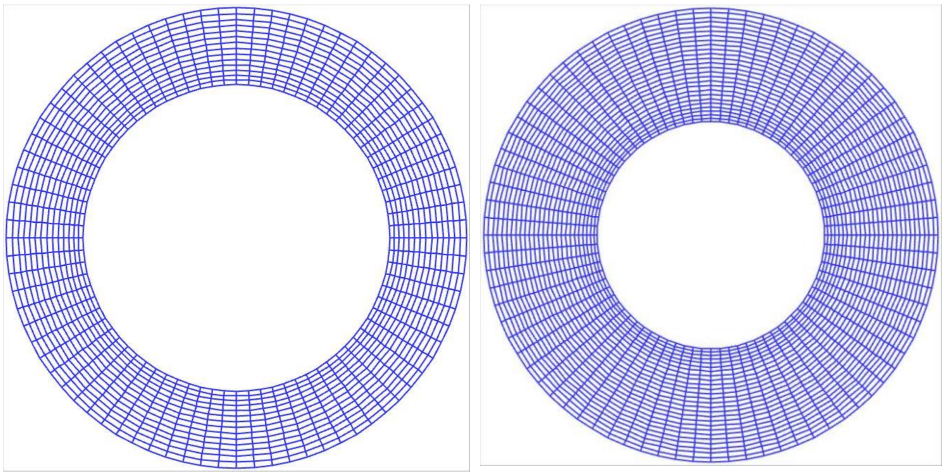


Figure 2: Computational domains: (Left) RR=1/1.5 and (Right) RR=1/2.

3. MATHEMATICAL FORMULATION

The two cylinders analyzed in this paper are entangled in a horizontal position and vary in diagonals, as seen in the problem diagram in Figure 1. For numerical analysis, the Galerkin finite element method was used, and the non-dimensional ruling equations solved by the model were as follows:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \tag{1}$$

$$\rho_f\left(\frac{\partial u}{\partial t}\right)+\rho_f\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=-\frac{\partial p}{\partial x}+\mu_f(\nabla^2 u), \tag{2}$$

$$\rho_f\left(\frac{\partial v}{\partial t}\right)+\rho_f\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}\right)=-\frac{\partial p}{\partial y}+\mu_f(\nabla^2 v)+\rho_f\beta_f g\left(T_f-T_c\right), \tag{3}$$

$$\left(\rho c_p\right)_f\left(\frac{\partial T_f}{\partial t}\right)+\left(\rho c_p\right)_f\left(u\frac{\partial T_f}{\partial x}+v\frac{\partial T_f}{\partial y}\right)=\nabla\cdot\left(k_f\nabla T_f\right), \tag{4}$$

where, t' is the dimensional time. To transform the above dimensional system of equations into non-dimensional one for doing an analysis in general scale, the following parameters are used:

$$X=\frac{x}{D_i},\ Y=\frac{y}{D_i},\ U=\frac{uD_i}{\alpha_f},\ V=\frac{vD_i}{\alpha_f},\ \theta_f=\frac{\left(T_f-T_c\right)}{\left(T_h-T_c\right)},\ P=\frac{pD_i^2}{\rho_f\alpha_f^2} \tag{5}$$

where, Di is the diameter of the inner cylinder, k, and α are the density, the thermal conductivity, and the thermal diffusivity of fluid. By substituting the parameters in equation 5 into the above set of governing equations (1 - 4), we can get the following non-dimensional equations:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \tag{6}$$

$$\left(\frac{\partial U}{\partial t}\right)+\left(U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}\right)=-\frac{\partial P}{\partial X}+(\nabla^2 U) \tag{7}$$

$$\left(\frac{\partial V}{\partial t}\right)+\left(U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}\right)=-\frac{\partial P}{\partial Y}+(\nabla^2 V)+\frac{Ra}{Pr}\theta_f, \tag{8}$$

$$\left(\frac{\partial \theta_f}{\partial t}\right)+\left(U\frac{\partial \theta_f}{\partial X}+V\frac{\partial \theta_f}{\partial Y}\right)=\frac{1}{\mathrm{Pr}}\left(\nabla^2\theta_f\right) \tag{9}$$

where, U and V are the non-dimensional velocities in X and Y non-dimensional directions, respectively. The main non-dimensional variable of the problem is Rayleigh and Prandtl numbers, which can be defined as:

$$R_a = \frac{g\,\rho\,f\,B_f\,D_i^3\,(T_h - T_c)}{\alpha\,f\,\mu_f}, \quad P_r = (\frac{\mu}{\rho\,\alpha})f' \qquad (10)$$

4. BOUNDARY CONDITIONS

The velocities in both X and Y directions are presumed U = V = 0 on all solid surfaces of both inner and outer cylinders. Whereas, the thermal boundary conditions are assumed as follows:

$$\theta_f = 1 \text{ at } (\text{r} = R_i) and\ (0 < \theta^\circ < 360)$$

$$\theta_f = 0 \text{ at } (\text{r} = R_0) and\ (0 < \theta^\circ < 360)$$

The Prandtl number is taken as 0.71 for air. The following formula is used to measure the mean rates of heat transfer from the inner cylinder surface in terms of Nusselt number mean:

$$N_{um} = \frac{1}{\pi}\int_0^\pi \frac{\partial \theta_f}{\partial N}.d\theta^\circ \ \vdots\ r = R_i' \qquad (11)$$

where, N is the normal direction at the inner cylinder surface.

5. VERIFICATION

Comparison of the current program's findings with those of Pop et al. and Kumari and Jayanthi for the local angular Nusselt number Nu$\sqrt{R_a}$.

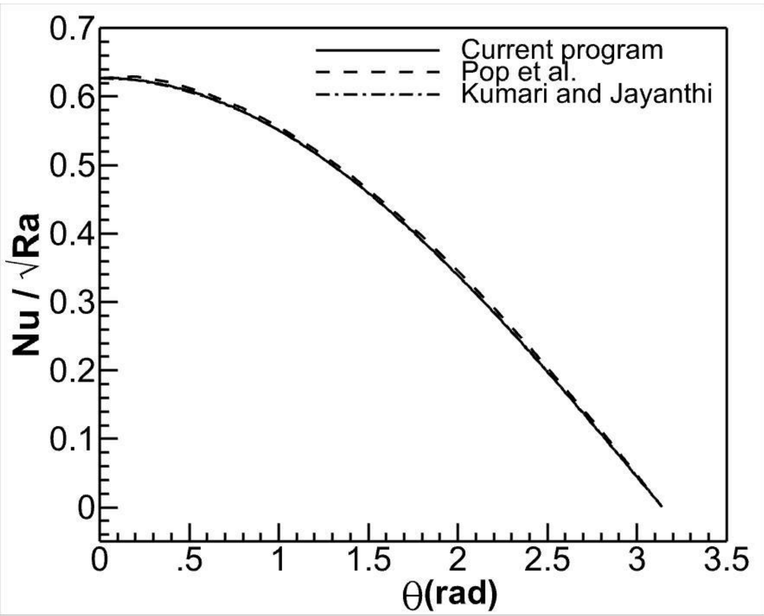


Figure 3: Verification of the current program.

6. RESULTS AND DISCUSSION

The current research revolves around the natural convection of a heated circular cylinder placed in another cylindrical layer of less diameter and the vacuum filled with air. The main goal is to examine the effect of the engineering parameter namely; with radius ratio RR = Ro / Ri, on convective and conductive heat transfer in all liquid and solid phases, and more generally in their thermal domains, for several Rayleigh numbers (Ra). The geometrical and thermal properties of the porous material are kept constant, for example the ratio of solid-to-fluid thermal conductivity (kR) = 1.0, the ratio of the diameter of the two cylinders RR = 1/2, RR = 1 / 1.5, and air is chosen as a working fluid with (Pr) = 0.71.

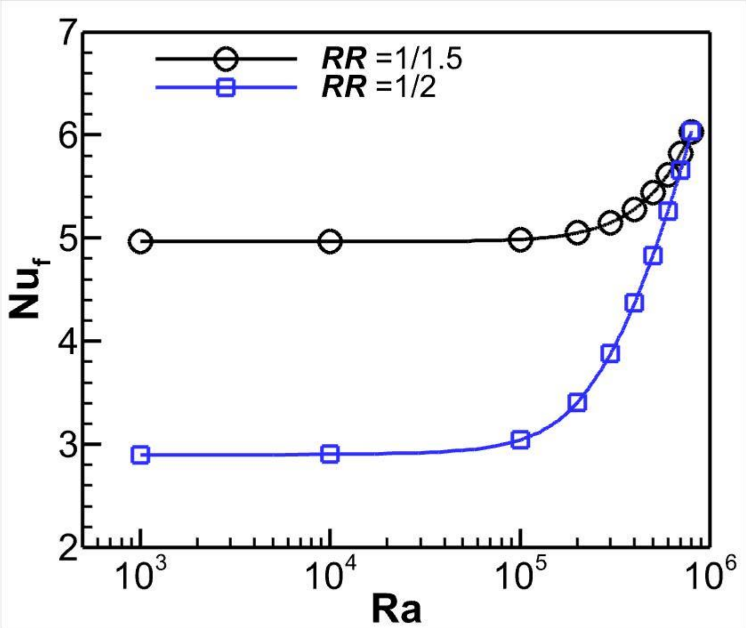


Figure 4: Variation of average Nusselt number against Rayleigh number at various annular radius ratio.

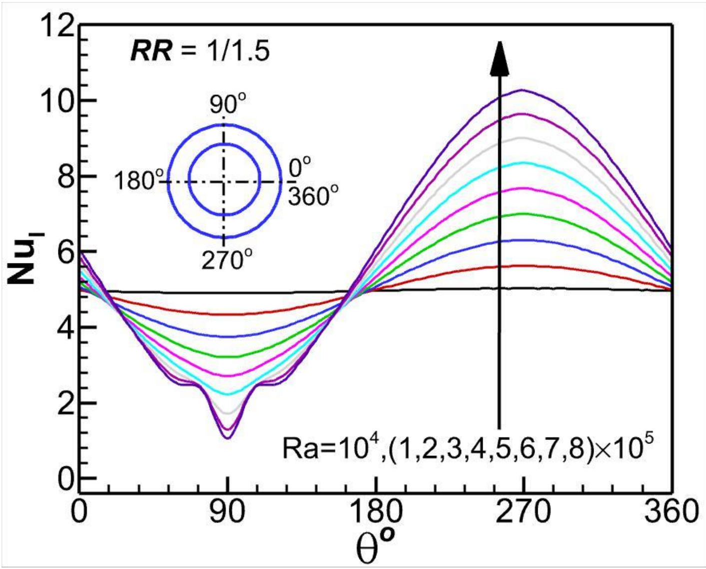


Figure 5: Variation of local Nusselt number around the internal cylinder wall, at annular radius ratio RR=1.5, and at various Rayleigh number.

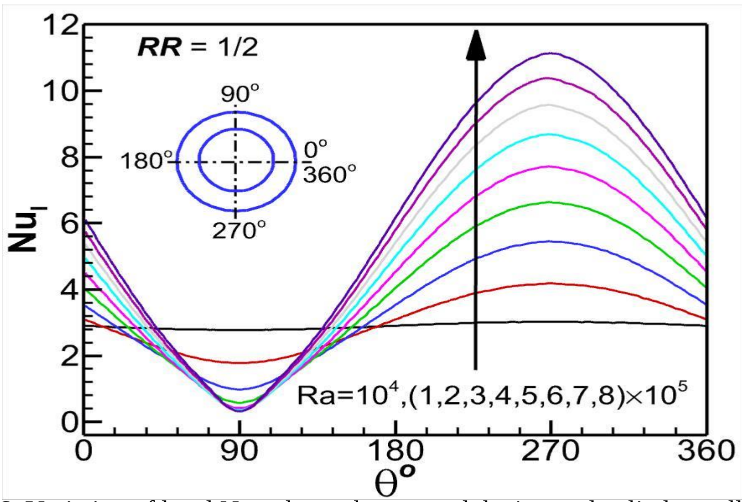
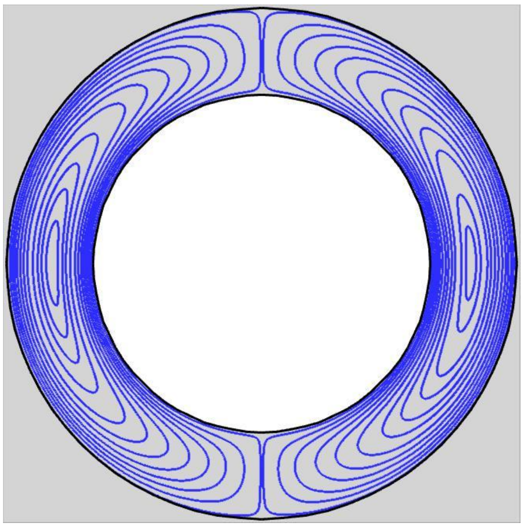
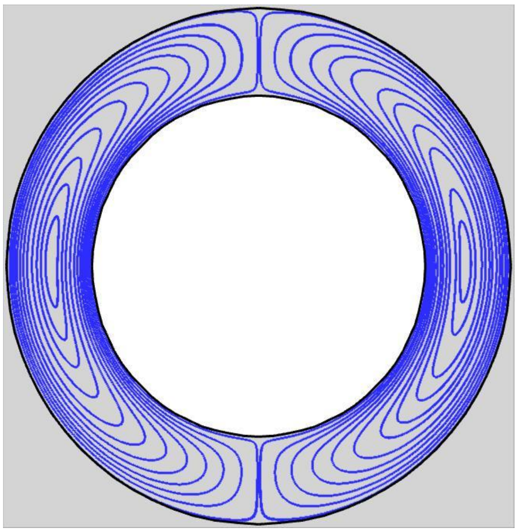


Figure 6: Variation of local Nusselt number around the internal cylinder wall, at annular

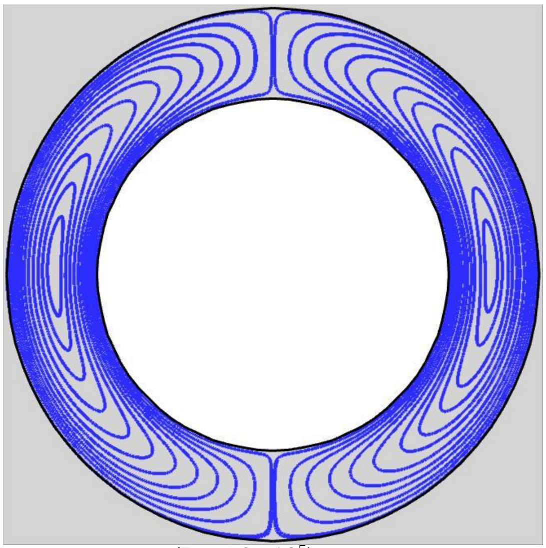
radius ratio RR=2, and at various Rayleigh number.

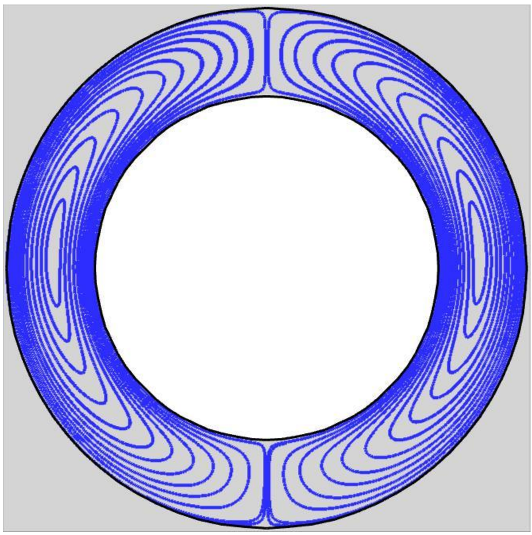
(Ra = 10^4) (Ra = 10^5)



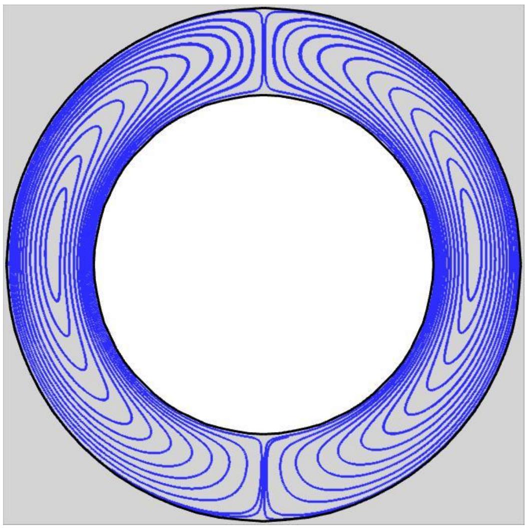


(Ra = 2 x10^5) (Ra = 4 x $10^{5)}$





(Ra = 6 x10^5) (Ra = 8 x 10^5)



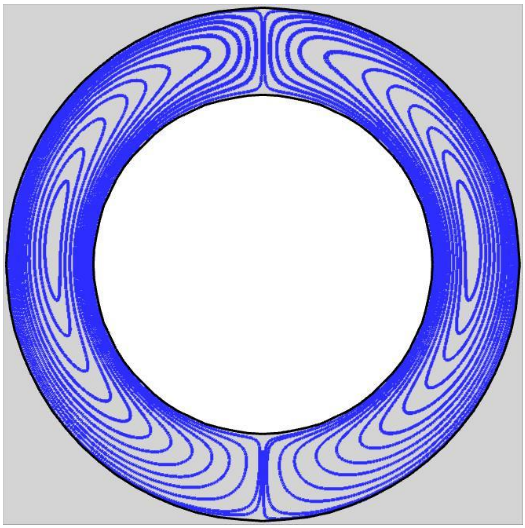


Figure 7: Streamlines patterns at radius ratio RR=1/1.5, and at various Rayleigh numbers.

(Ra = 10^4) (Ra = 10^5)

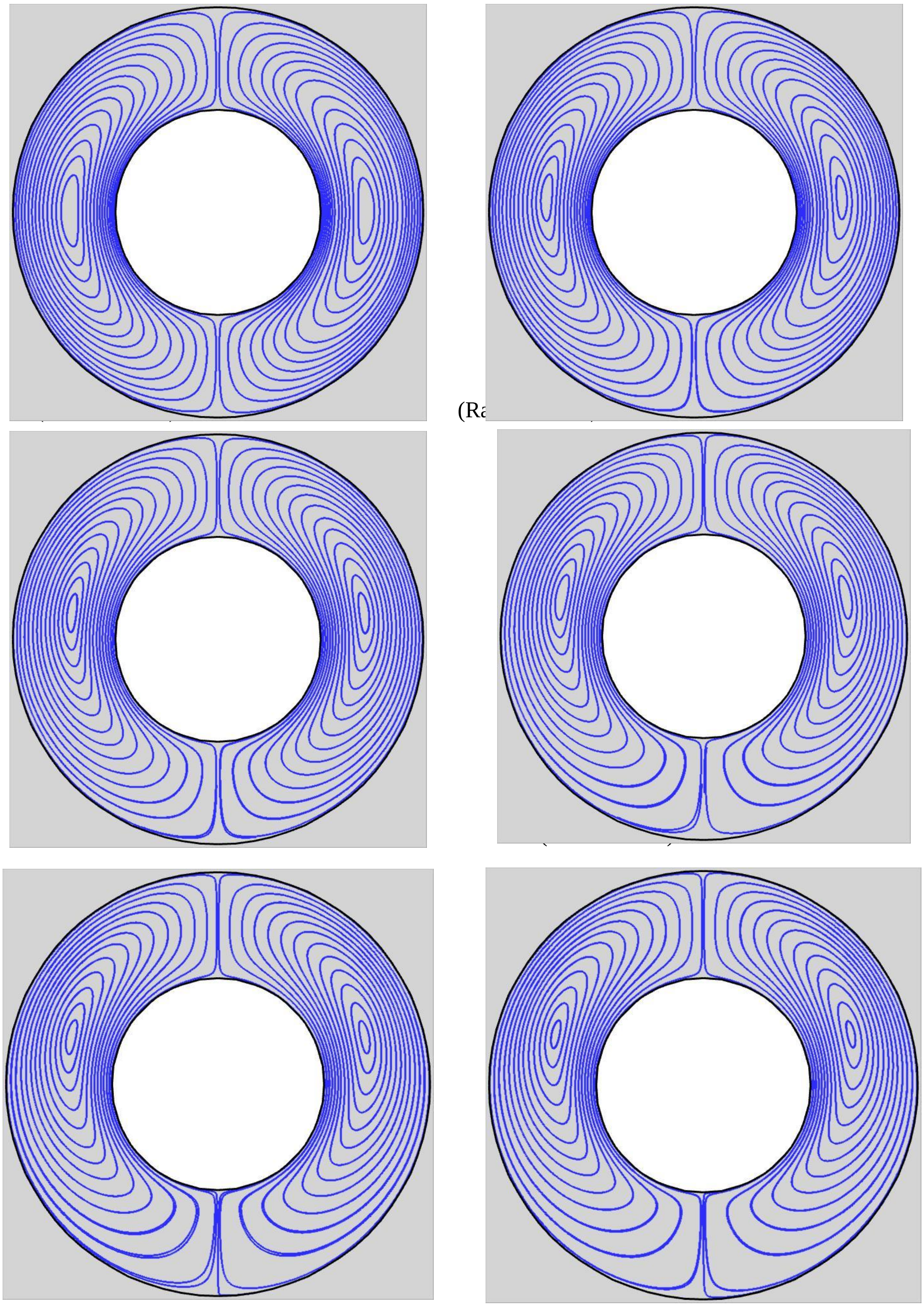


Figure 8: Streamlines patterns at radius ratio RR=1/2, and at various Rayleigh numbers.

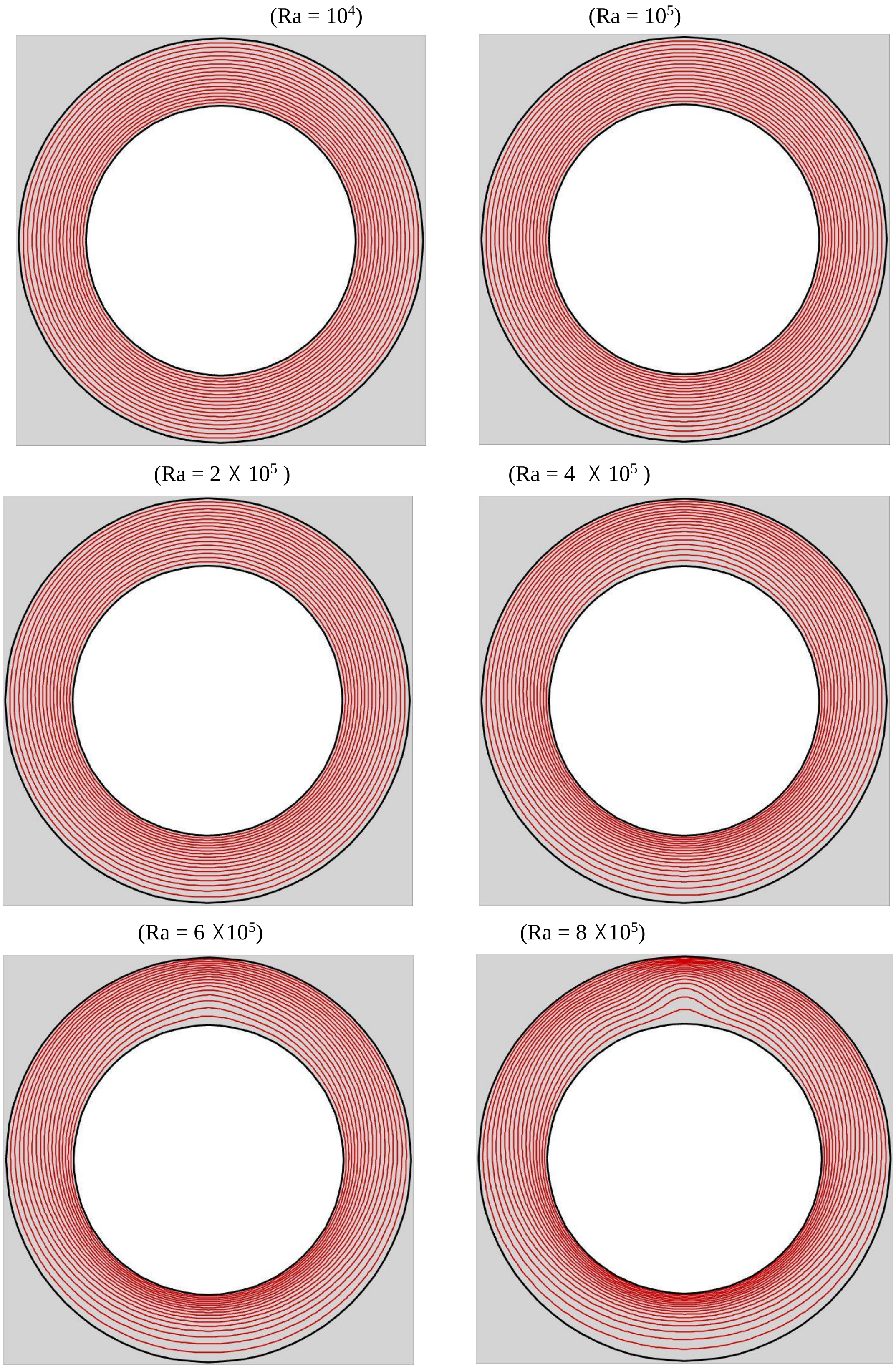


Figure 9: Isotherms patterns at radius ratio RR=1/1.5, and at various Rayleigh numbers.

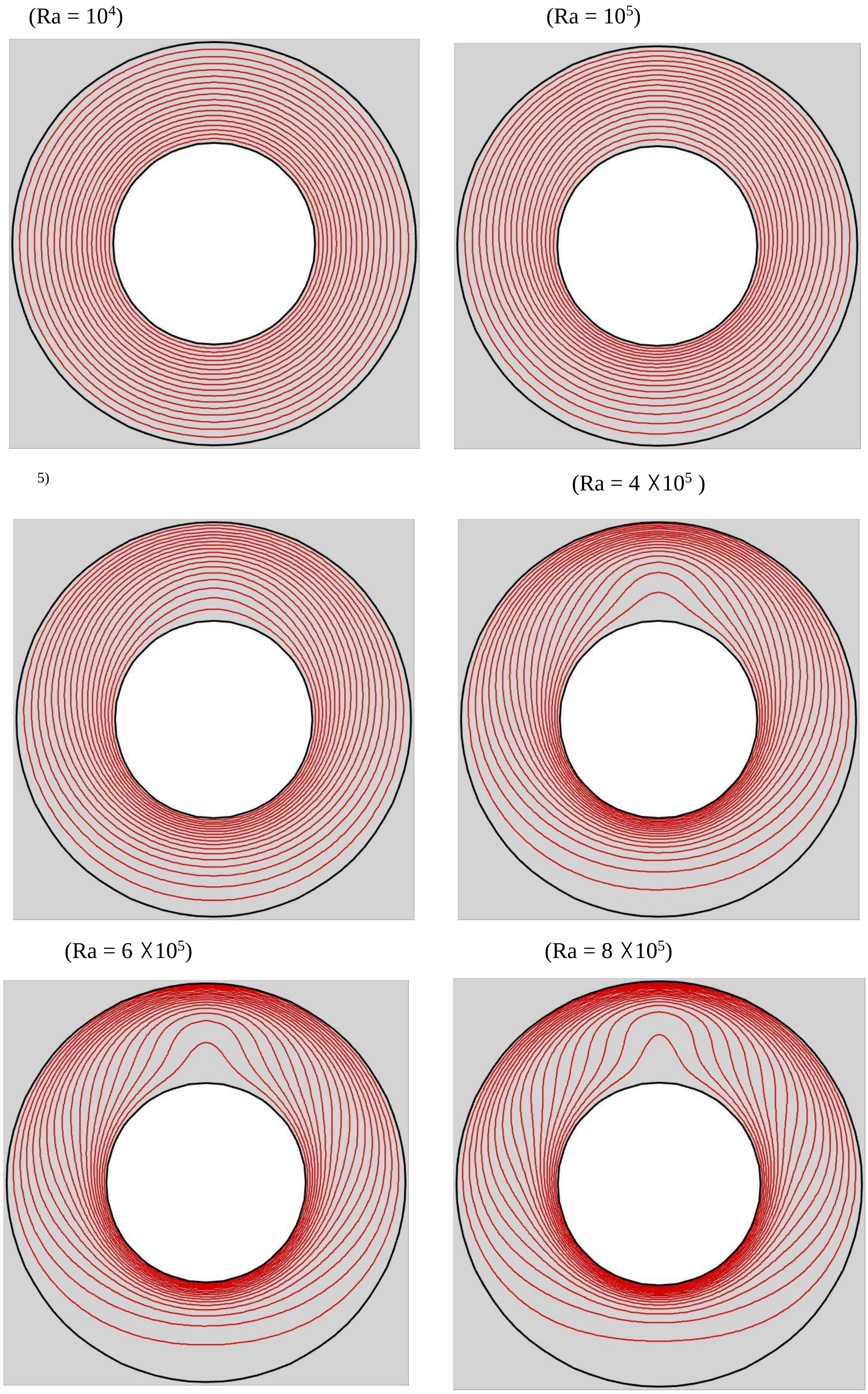


Figure 10: Isotherms patterns at radius ratio RR=1/2, and at various Rayleigh numbers.

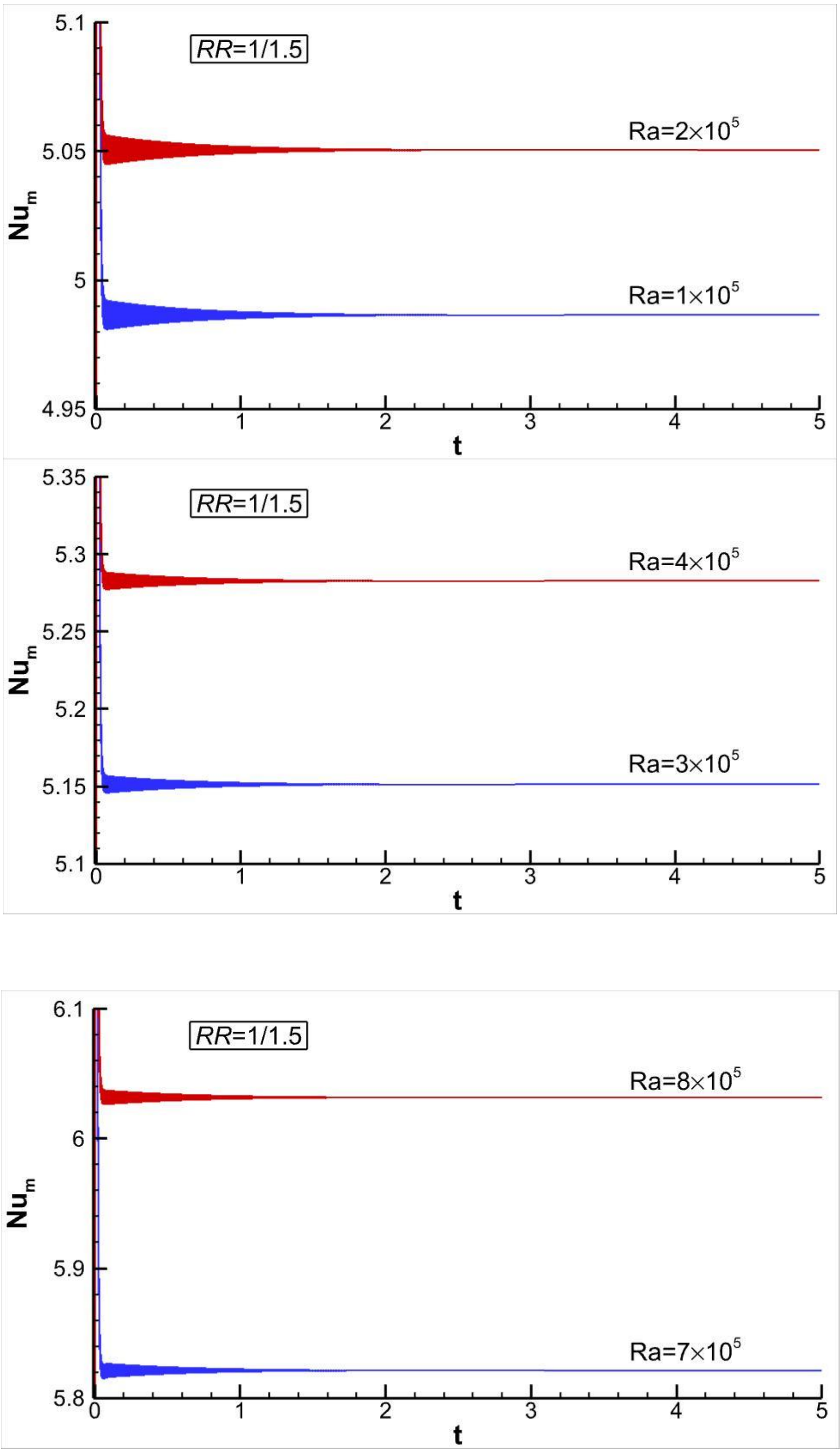
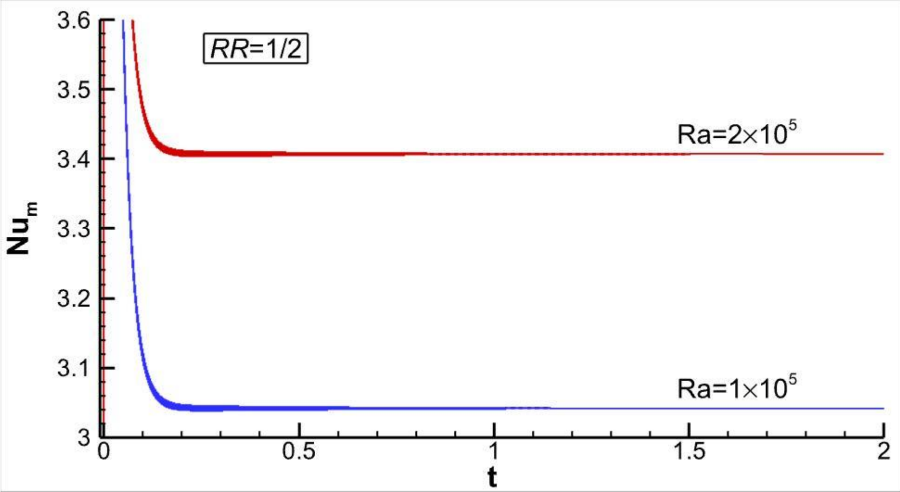
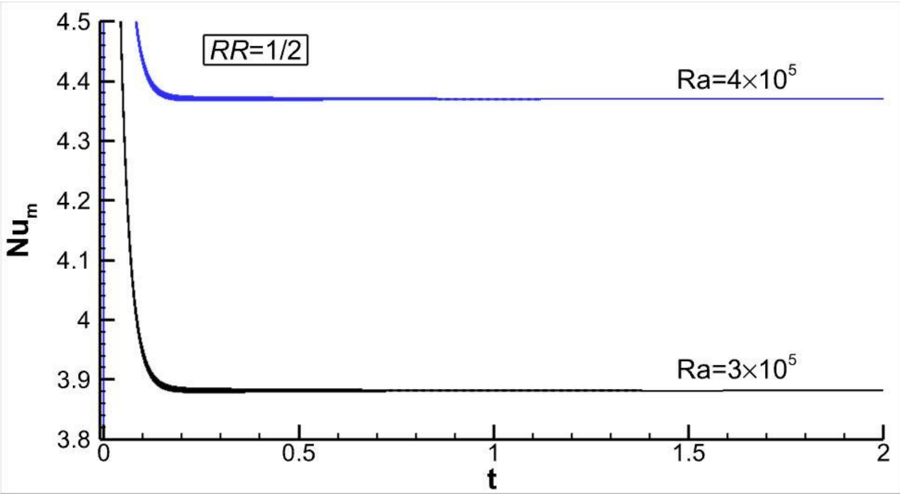


Figure 11: Variation of average Nusselt number with time for annular radius ratio RR=1/1.5 and at various Rayleigh numbers.





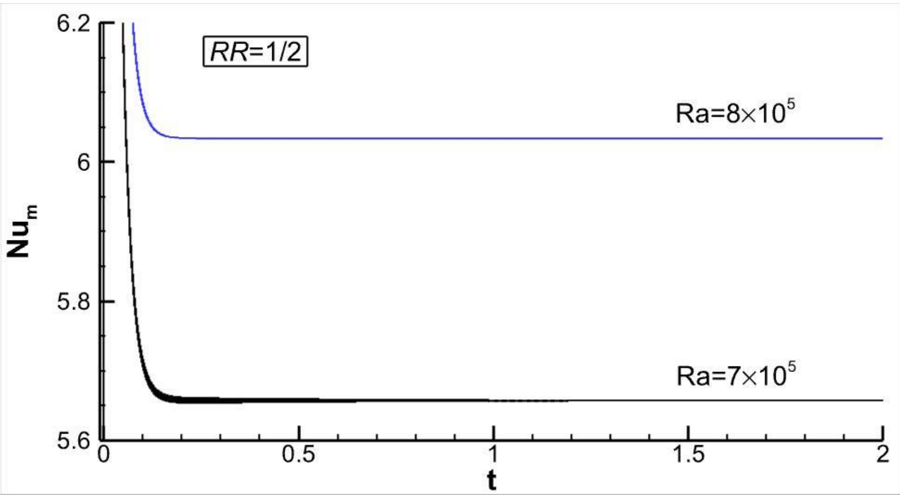


Figure 12: Variation of average Nusselt number with time for annular radius ratio RR=1/2 and at various Rayleigh numbers.

Figure (4) depicts the average variance of the Nusselt number with the Rayleigh number for various values of the radius ratio. Figure (4) indicates that the Nusselt number increases as the radius ratio value increases. For the radius ratio number 1/2, the Rayleigh number was 10^3 and the Nusselt number was 3, but as the Rayleigh number increased to 10^6, the Nusselt number increased to 3.9 for the same radius ratio. It also indicates the effect of the radius ratio on the Nusselt number; when the radius ratio number increased, the Nusselt number increased as well; for example, for the same value of the Rayleigh number at the radius ratio number of 1/1.5, the Nusselt number was 5, and when the Rayleigh number increased to 10^6, the Nusselt number increased to 6. Heat transfer is affected, and the Nusselt number (and thus Rayleigh number) varies dramatically with velocity. As the velocity of the cylinder increases, so does the amount of heat transmitted to convection, as seen in the diagram.

The local Nusselt number along the heated cylinder wall of various Rayleigh numbers with values of 10^4, (1, 2, 3, 4, 5, 6, 7, 8) 10^5, with a radius of (1/1.5), is shown in Figures 5-6. For the lowest Rayleigh number of 10^4, these figures indicate that the Nusselt number increases in the wall, reaches a peak, and then starts to decrease, it was a simple Nusselt number with a deflection angle of approximately 20° from the beginning of the wall to its end. At a constant value for the radius ratio, change 1 0^4, (1, 2, 3, 4, 5, 6, 7, 8) 10^5 Rayleigh number, for each number from (5 - 6). These numbers also demonstrate that as Rayleigh's number increases, Nusselt number also increases. As the diameter ratio changes with values of (1/1.2), Fig.6, a peak value begins to appear faster around the cylinder bore wall, followed by another peak value as the Rayleigh number rises. The first peak in Nusselt number 11 was at 270° to the wall, according to Figure (6), with a diameter ratio of (1/ 1.2) and a Rayleigh number of 8×10^5. A similar effect can be seen for a different Rayleigh number, Figure (5) Nusselt number was approximately equal from the beginning of the hot wall to its end, as seen in the 10^4 Rayleigh number. The Nusselt number increases from 4 at the beginning of the wall to 11 at the top, then falls to 5.9 at the end of the wall at 8×10^5 Rayleigh number. For mixed convection flow, streamlines and different heat patterns are shown in Figures (7-10). In the radius ratio RR = 1 / 1.5, and at various Rayleigh numbers small eddies are observed in the simplified image of Fig. As the Rayleigh number increased, a large vortex developed, at radius ratio RR = 1/2, and at various Rayleigh numbers. Increased vortices were observed further as in Fig.8, for moderate temperature and elevated temperature, radius ratio 1/1.5 and 1/2 respectively, As Rayleigh increases, vortices begin to develop near the cylinder wall. The heat begins to concentrate on the hot wall as an increase in the Rayleigh number means that the convection in the flow begins to decrease as the velocity of the fluid increases. The variation of the total Nusselt number in the cylindrical shape and the average Nusselt number of the hot upper wall versus the radius ratio of the various Rayleigh numbers are shown in Figures 11 and 12 respectively. It is evident that the total and mean heat transfer rate increases with increasing radius ratio and Rayleigh number increasing due to forced convection control in heat transfer process with change of time. The average values of a hot wall number for the radius ratio (1/ 1.5)are more than twice the total Nusselt number in the radius ratio (1/2)at the beginning of the heating time by t = 0, while the radius ratio (1/2) takes 0.35 time for the beginning of the heat transfer.

7. CONCLUSIONS

This study investigates the numerical properties of the two-dimensional natural convection problem between two horizontal concentric cylinders with an inner radius Ri and an outer radius Ro. A detailed analysis of the distribution process of the flow line, isothermal, and Nusselt number was applied to verify the effect of the proportions of the inner cylinder dimensions with the outer and the effect of the angles and inclination of the inner shell on fluid flow and heat transfer for a different set of Rayleigh numbers. The Nu shapes were not symmetrical due to the presence of secondary vortices and triple vortices on the upper surface of the inner cylinder resulting from the height of the column from the inner cylinder. The Nu value in the ratio of diameters 1 / 1.5 is greater than that of Nu in the ratio of 1/2 because the isothermal forms more densely on the inner cylinder surface and promotes heat transfer. As a result, the difference in the ratio of the diameters and the angle of inclination of Rayleigh numbers has a nominal effect on the heat transfer of different dimensional proportions. The effect of the ratio of the diameters and the angle of inclination is especially strong in the upstream regions on isothermal walls.

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