ISSN No. 2455-5800
Journal of Scientific Research in Allied Sciences

Original Research Article

THERMAL ANALYSIS ANNULUS DURING NATURAL CONVECTION USING ANSYS

Dushyant Singh Chandel, Dr. S. K. Nagpure

Mechanical Engineering Department, Scope College of Engineering Bhopal, India

Article history:
Submitted on: February 2017
Accepted on: February 2017
Email: info@jusres.com

ABSTRACT

Heat transfer and fluid flow through natural convection in annulus are significant phenomenon in various engineering system due to having technological relevance in thermal storage system, nuclear reactor, nuclear reactor technology, cooling of electronic apparatus such as heat sink, heat exchanger, and numerous other practical applications. In the work, at steady state condition the natural convective heat transfer has been investigate between concentric annulus computation using finite volume tool ANSYS Fluent 14.5. The inner radial wall of Annulus has been subjected to elevated temperature and the outer radial wall is subjected to room temperature. The effect of buoyancy due to free convection has been seen under the influence of various parameters such as radius ratio, varying Rayleigh number, Prandlt number, Nusselt number.

The equation of motion for free convection of concentric annulus has been solved by using SIMPLE algorithm where Bossious or incompressible boundary condition has been assigned the obtained result has been compared with various result and the results are within acceptable limit. It has been found that varying Rayleigh number and radius ration significantly affects the temperature profile and flow field. However, the Prandlt number has in-significant effect.

KEYWORDS: Annulus, Natural convection, Prandlt number, Rayleigh number.

INTRODUCTION

Convective heat transfer and fluid flow in annulus are significant phenomena in engineering systems as a consequence of their technological applications in nuclear reactors, heat exchangers, aircraft fuselage insulation to underground electrical transmission cables, thermal storage systems, cooling of electronic devices, solar energy systems, boilers, compact heat exchangers, cooling systems, cooling core of nuclear reactors, thermal insulation, gas-cooled electrical cables, and electrical gas-insulated transmission lines [1–7]. Therefore, researches on the heat transfer improvement in annulus are essential. Several researchers have been conducted on the improvement of heat transfer characteristics in natural-convection relevance. Nevertheless, the heat transfer enhancement in natural-convection applications has gained little attention.

LITERATURE SURVEY

Mack and Hardee in 1968 numerically investigate the natural convection Natural convection between concentric spheres at Low Rayleigh numbers under steady axi-symmetric boundary condition. Various parameters such as velocity, temperature, and streamline distribution on over all heat-transfer rates are briefly demonstrated.

Yin et al. 1973 examined the flow patterns in spherical annuli during natural convection. The flow pattern are compared with previous available literature for steady state and unsteady condition

Douglas et al. 1978 examined the steady forced convection of rotating spherical annuli. The governing equations are solved by using Galerkin approach. The obtained results are presented in form of temperature profile, velocity distribution and other heat-transfer characteristics.

In 1977 Ralph studied the flow of liquid in between a sphere and its cubical enclosure during free convection. A unique flow pattern has been found at the interference between up-flow and down-flow layers.

In 1988 sanjay and sengupta numerically analyzed the free convection between isothermal vertically eccentric spheres. The results show that the positive eccentricities have the adverse effect on heat transfer and conclude that the heat transfer enhances for very high positive eccentricities.

Chu and lee 1993 theoretical investigation of natural convection heat transfer of fluids between two concentric isothermal spheres The transient behaviour of the flow field and its subsequent effect on the temperature distribution for different Rayleigh numbers and radius ratios are analyzed by finite difference methods.

Asan 2000 focused on steady-state, laminar, two-dimensional natural convection in an annulus between two isothermal concentric square ducts. Solutions are obtained up to Rayleigh number of 106. Three different dimension ratios, L∗, namely 15, 310, and 35, are considered. The effects of dimension ratio and Rayleigh number on the flow structure and heat transfer are investigated.

Wu and tsai 2004 analyzed the transient natural convection for the temperature dependent viscosity of fluids in spherical annulus and between two vertically eccentric spheres using MSM, The results of this analysis show that heat and flow patterns vary with the Rayleigh number and the eccentricity; besides, the effect of variable viscosity is investigated.

Chen 2007 investigates the effects of height and radius ratio with a Newtonian fluid to determine heat transfer by natural convection between the sphere and vertical cylinder with isothermal boundary conditions. The inner sphere and outer vertical cylinder were heated and cooled in a steady change of temperature.

Jung, and Tanahashi 2008 performs numerical analyses for two different working fluids. One is an analysis of natural convection using a model with air that is compared with the results of applied experiments. Then, it can be used to investigate the pattern of natural convection in concentric spheres

Reddy and narasimhan 2010 studied interplay between internal heat generation and externally driven natural convection inside a porous medium annulus. The axisymmetric domain is bounded with adiabatic top and bottom walls and differentially heated side walls sustaining steady natural convection of a fluid with Prandtl number, Pr = 5, through a porous matrix of volumetric porosity, $\phi = 0.4$, . A correlation is proposed to predict the overall annulus Nu as a function of RaE□, RaI□, Da and γ. It predicts the results within ± 20% accuracy.

Jahanshahi et al. 2010 Heat transfer enhancement has been investigated in a square cavity subject to different side wall temperatures using water/SiO2 nanofluid. An experimental setup has been used to extract the conductivity value of nanofluid.

Mehrizi et al. 2013 in his paper the lattice Boltzmann method is used to investigate the effect of nanoparticles on natural convection heat transfer in two-dimensional horizontal annulus. The study consists of an annular-shape enclosure, which is created between a heated triangular inner cylinder and a circular outer cylinder.

Shekholeslami and ganji 2014 in his study magneto hydrodynamic effect on free convection of nanofluid in an eccentric semi-annulus filled is considered. The effective thermal conductivity and viscosity of nanofluid are calculated by the Maxwell–Garnetts (MG) and Brinkman models, respectively.

Ben,erkhi et al. 2016 analyze the forced convection in a foam-filled elliptic annulus. The results are evaluated with those of the empty annulus to review the thermal performance of the used aluminum foam. The detailed mathematical and physical characteristics of the convection are presented and discussed.

MATHEMATICAL MODELLING

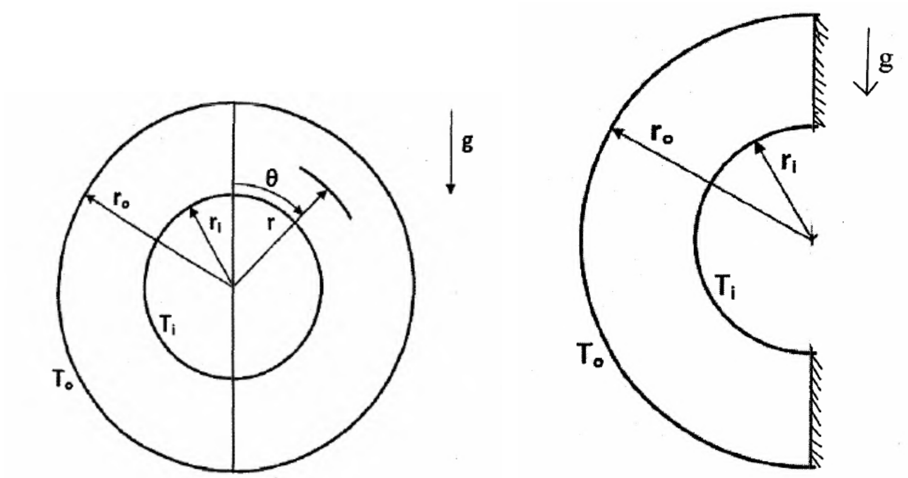

Figure 1 Physical model and coordinate system and Sectional view

figure 1 Here θ is the polar angle and varies from 0 to π The governing equations in polar coordinate are described in dimensional form specifying boundary conditions for velocity and temperature as below:

$$\frac{\partial}{\partial r}(r^2 u \sin\theta) + \frac{\partial}{\partial \theta}(r^2 v \sin\theta) = 0$$

$$u\frac{\partial u}{\partial r}+\frac{v\partial u}{r\partial\theta}-\frac{v^2}{r}=-\frac{1}{\rho_m}\frac{\partial p}{\partial r}$$
$$+\mu\left[\frac{1}{r^2}\frac{\partial}{\partial r}\left(r^2\frac{\partial u}{\partial r}\right)+\frac{1}{r^2\sin\theta}\frac{\partial}{\partial\theta}\left(\sin\theta\frac{\partial u}{\partial\theta}\right)-2\frac{u+\frac{\partial v}{\partial\theta}+v\cot\theta}{r^2}\right]-\frac{1}{\rho_m}\rho g\cos\theta \qquad (2)$$

$$u\frac{\partial u}{\partial r}+\frac{v\partial u}{r\partial\theta}+\frac{uv}{r}=-\frac{1}{r\rho_m}\frac{\partial p}{\partial\theta}$$
$$+\mu\left[\frac{1}{r^2}\frac{\partial}{\partial r}\left(r^2\frac{\partial v}{\partial r}\right)+\frac{1}{r^2\sin\theta}\frac{\partial}{\partial\theta}\left(\sin\theta\frac{\partial v}{\partial\theta}\right)+\frac{2}{r^2}\frac{\partial u}{\partial\theta}-\frac{v}{r^2(\sin\theta)^2}\right]+\frac{1}{\rho_m}\rho g\sin\theta \qquad (3)$$

$$u\frac{\partial T}{\partial r}+\frac{v}{r}\frac{\partial T}{\partial\theta}=\alpha\left(\frac{\partial^2 T}{\partial r^2}+\frac{1}{r^2}\frac{\partial^2 T}{\partial\theta^2}+\frac{2\partial T}{r\partial r}+\frac{\cot\theta}{r^2}\frac{\partial T}{\partial\theta}\right) \qquad (4)$$

$$\rho=\rho_m\left[1-\beta(T-T_m)\right]$$

Boundary Condition

u=0, v=0, T=Ti at r=ri for 0≤θ

u=0, v=0, T=Ti at r=ro for 0≤θ≤π

u=0, v=0, dt/dθ=0 at θ=0 for ri≤r≤ ro

u=0, v=0, dt/dθ=0 at θ=π for ri≤r≤ ro

Non Dimensional Parameters

$$U=\frac{u}{a/d} \qquad T^*=\frac{T-T_c}{T_h-T_c} \qquad V==\frac{v}{a/d}$$

$$P=\frac{\rho}{\mu\alpha/d^2}$$

$$R=r/d \qquad \bar{\rho}=\frac{\rho}{\rho_m} \qquad rr=\frac{r_o}{r_i}$$

Where rr (radius ratio)

$$rr=\frac{r_o}{r_i}$$

Pr (Prandlt Number)

$$\Pr = \frac{\mu}{\rho_m \alpha} = \frac{\vartheta}{\alpha}$$

Ra (Rayleigh Number)

$$Ra = \frac{g\beta \Delta T d^3}{\vartheta \alpha}$$

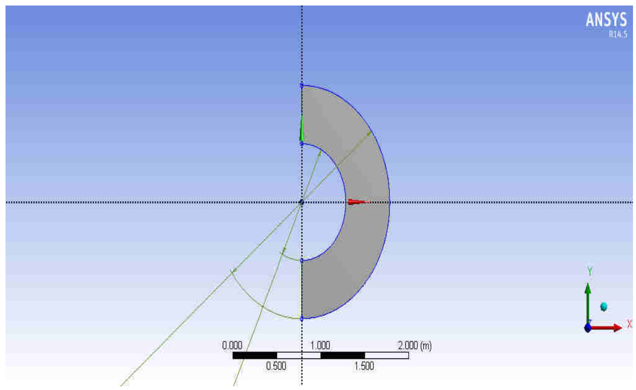

Figure 2 Model Geometry

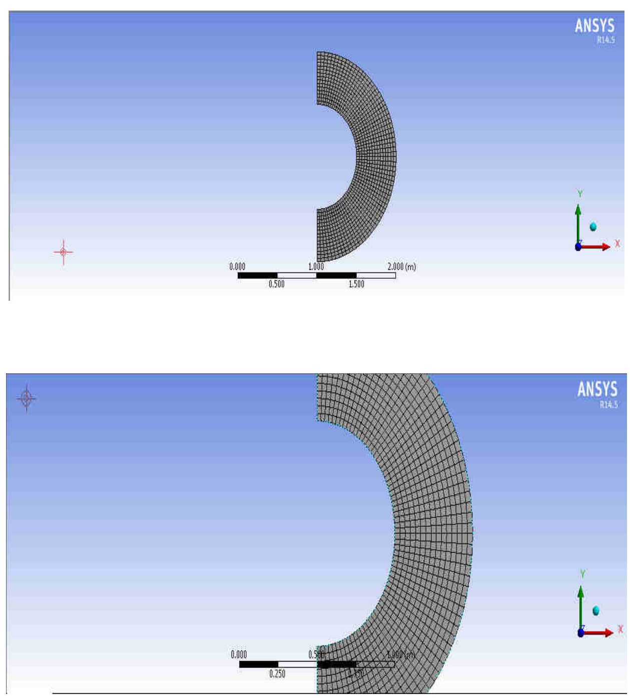

Figure 3 Mesh Model

Table 1 the Properties of Fluid

Gravitational acceleration	g	9.81
Density	ρ	1.0137
Specific heat	Cp	1007.5
Thermal Conductivity	k	0.02917
Dynamic viscosity	μ	0.00002074
Mean Temperature difference	tm	$3.48\text{x}10^2$
Kinematic viscosity	γ	0.0000205
Beta	β	0.00287
Delta	Γ	5
Radius	r	2,2.2,3
Thermal Diffusivity	α	0.00002856
Prandtl Number	Pr	0.716 and 0.01<Pr<10
Rayleigh number	Ra	$>10^3$

METHODOLOGY

The ANSYS 14.5 finite element program was used for analyzing the heat transfer with 2D annulus. For this purpose, a 2 dimensional flow region has been developed which includes inner and outer region which is subjected to different temperature. A 20-node three-dimensional structural solid element was selected to model the annulus surface. The annulus surface was discretized into 990 elements with 1072 nodes. The annulus surface boundary conditions can also be provided in mesh section through naming the portion of modeled annulus i.e Inner, Outer, Top region, Bottom region. For simulating annulus the Properties of Fluid are tabulated in table 1.

RESULT AND DISCUSSIONS

The governing of equation of natural convection for concentric sphere/annulus has been solved by using FEV tool ANSYS FLUENT. Where, SIMPLE algorithm has been used along with least square iterative approach. Parametric analysis has been carried out such as effect of non dimensional parameters such as Prandlt number (Pr), Rayleigh number (Ra), radius ratio (rr) on various thermal performance characteristics i.e. temperature field, flow field and the Nusselt number has been illustrated in this chapter.

GRID SENSITIVITY ANALYSIS

Using grid sensitivity test the effect of grid size on Nusselt number has been investigated and optimal value for the some parameters has been evaluated during FEV simulation. In table sensitivity analysis of radius ratio 22 at Pr 0.7 and Rayleigh number 10^3 has been tabulated in table 2 6.1. Four different grid size has been used where values of Nusselt number has been determined at inner and outer portion through which average Nusselt number has been calculated

Table 2 Table Sensitivity analysis for rr = 2, Pr = 0.7 and Ra = 103

Grid Size	Nu_h	Nu_c	Avg. Nusset Number
50x50	1.96	1.96	1.96
40x40	1.96	1.96	1.96
30x30	1.97	1.97	1.97
20x20	1.99	1.99	1.99

From the table 2 it has been observed that for grid size 20x20 and 30x30 the variation has been seen but for the 40x40 and 50x50 there is no such variation in magnitude of average Nusselt number. Therefore, 40x40 grid sizing has been selected for the further calculation.

Validation of FEV scheme

On table 3 exclusive comparisons has been tabulated for the present work with previous work. The parameter such as average Nusselt number and maximum Stream function has been calculated using FEV.

Table 3 Comparison of Present work with reference papers at Ra= 10^3rr= 2, Pr= 0.7,.

Reference	Ψ_{max}	Avg. Nu
Present	3.23	1.1
Mack and Harde [1]	3.21	1.12
H.S. Chu, T.S.Lee [6]	3.21	1.09
H.W. Wu, Wen, C.Tsai [8]	3.25	1.10

Table 4 shows the comparison of present work with the previous work and the result shows good agreement. It has found that the percentage error lies in between 0.9-1.118 for average nusselt number and for maximum stream function it lies in between 0.62-7.38.

From the values of Chu and Lee [6] the comparison has been made for different Rayleigh number. Before evaluating the present data for comparison, a correlation has been developed between Rayleigh number used by Chu and Lee present Rayleigh number. From the developed correlation it has been observed that the results are very accurate as tabulated in table 4. The developed correlation are mentioned below

$$Ra_P = (rr-1)^3 \times Ra$$

Where, $Ra = \dfrac{g\beta\Delta T(r_i)^3}{\vartheta a}$, Chu and Lee adopted this expression of Rayleigh number.

$Ra_p = \frac{g\beta\Delta T(r_o - r_i)^3}{\vartheta a}$, developed expression of Rayleigh number utilized in present work

Table-4 Comparison of FEV scheme result with Chu and .Lee for different Ra value and radius ratio at Pr= 0.7

rr	Ra	Ra_p	Nusselt number (Chu and Lee) [6]	Present Average Nusselt number
1.2	$1x10^3$	8	1.00	1.00
1.2	$1x10^4$	$8x10^1$	1.00	1.00
1.2	$1x10^5$	$8x10^2$	1.01	1.01
1.5	$1x10^2$	$1.25x10^1$	1.00	1.00
1.5	$1x10^3$	$1.25x10^2$	1.00	1.00
1.5	$1x10^4$	$1.25x10^3$	1.07	1.07
1.5	$1x10^5$	$1.25x10^4$	1.92	1.95
1.5	$1x10^6$	$1.25x10^5$	3.71	3.47
2.0	$1x10^2$	$1x10^2$	1.00	1.00
2.0	$1x10^3$	$1x10^3$	1.10	1.10
2.0	$1x10^4$	$1x10^4$	1.97	1.96
2.0	$1x10^5$	$1x10^5$	3.49	3.43

Flow and Temperature fields

Figure 4 and 5 shows the isotherms and stream lines for Pr=0.7, rr = 2.0 at Ra= 10^4 and Ra= 10^5 for velocity distribution. The influence of Rayleigh number has seen in the isotherms and flow field. At the inner annuls the fluid near closer to the inner radius has lower density that the region near the outer radius this is because of inner radius region has higher temperature than the outer region. Therefore the fluid near the inner radius region moves downward. As the fluid migrates towards downward, it dissipates energy and ultimately leads the separation of the thermal boundary layer along the outer radius region. The heavier fluid enters the thermal boundary layer of the inner radius region and completes the recirculation trend. As the Rayleigh number increases the crescent shaped eddy move towards mid gap at the upper portion. This trend has been obvious from figure 4 as compared with radius ratio 2.2. In figure 5 where Rayleigh number is 10^5, in this case laminar convection is more influential for heat transfer rather than Rayleigh number is 10^4. At the upper region due to circulation of fluid are more influential leads to warming to outer region. The movement of hot fluid to outer region is more prominent from the Figure. 4 and 5. The stream function increases as Ra increases and it has been found that for Ra=l0^4 ψ_{max}= 14.138164 obtains at angle =63°, similarly for Ra=l0^4 ψ_{max}= 28.772739 at angle =58.500004°.

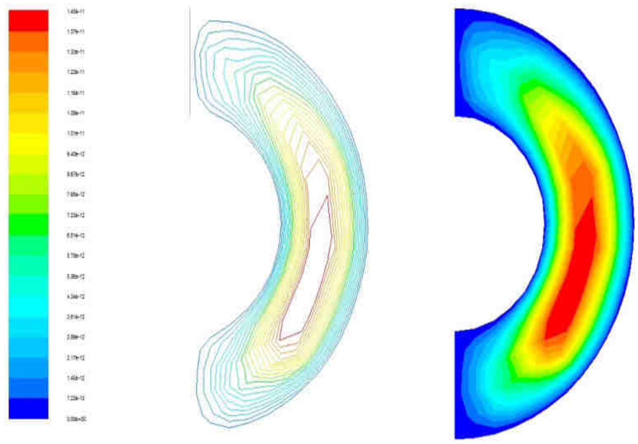

Figure 4 Isotherms and stream lines for Pr=0.7, rr = 2.0 at Ra= 104 for Velocity distribution

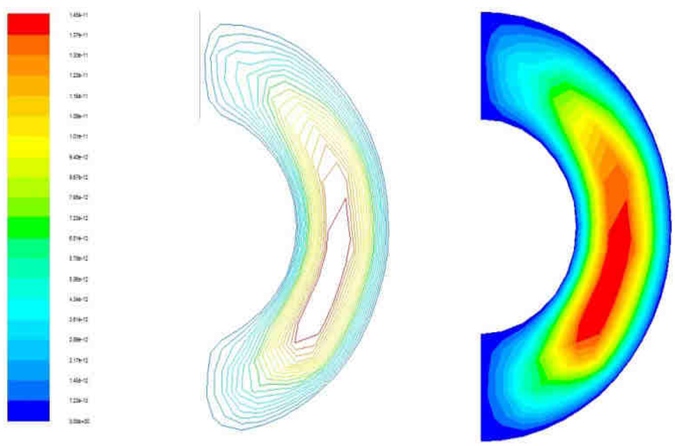

Figure 5 Isotherms and stream lines for Pr=0.7, rr = 2.2 at Ra= 105 for Velocity distribution

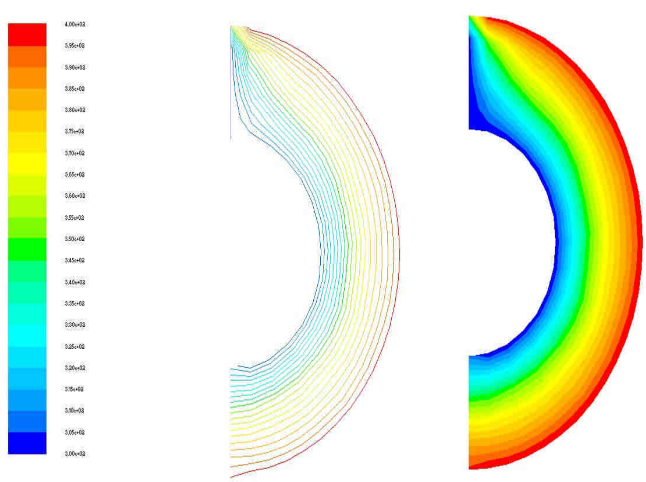

Figure 6 Isotherms and stream lines for Pr=0.7, rr = 2.2 at Ra= 104 for Static temperature

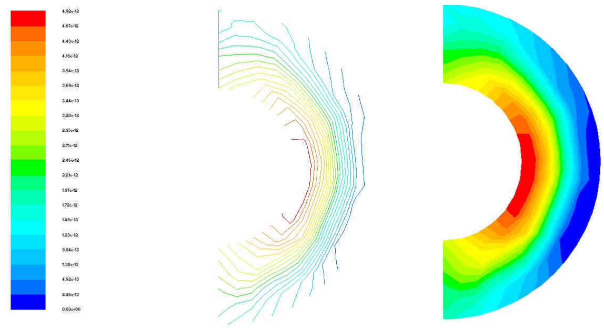

Figure 7 Counter of Stream Function of Spherical Annulus

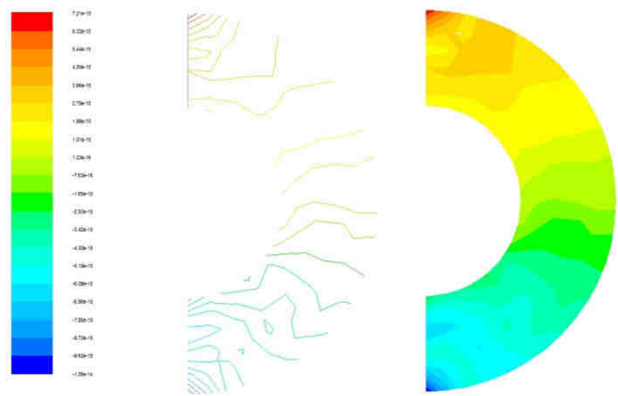

Figure 8 Isotherm and stream line contour of Static Pressure

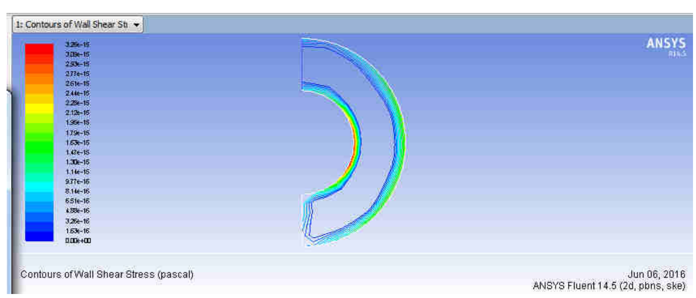

Figure 9 Isotherm and stream line contour of Wall shear stress

VARIATION OF AVERAGE NUSSELT NUMBER WITH RAYLEIGH NUMBER

Figure 10 and 11 shows the effect of Rayleigh number at radius ratio 2 on Nusselt number for different Prandlt number.Fig.11 shows the Effect of Ra= (10 - 5x10^4) at radius ratio 2 on Nusselt number for different Prandlt number .and in Fig 10 Ra. No. varies from 5xl0^3-10^5. From the both the graphs two characteristics are being noticed;

As the Rayleigh number increases the effect in Prandlt number is insignificant. As the Rayleigh number increases the Nusselt number increases linearly.

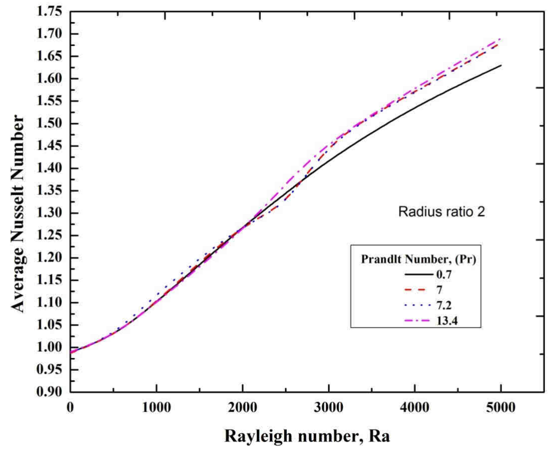

Figure 10 Effect of Ra= (10 - 5x104) at radius ratio 2 on nusselt number for varying Prandlt number.

Figure 10 illustrates the Variation of Nusselt number at different Pr values. It has been observed that as the Rayleigh number (Ra) increases the average Nusselt number linearly increases. This is because of Rayleigh number is strong function of Prandtl number and Grashof number. Rayleigh number associates with buoyancy-driven flow, therefore Rayleigh number below the critical value, the heat transfer is primarily through conduction and after exceeding the critical value this heat transfer takes place through convection.

Figure 11 illustrates the effect of Rayleigh number on average Nusselt number for different Prandlt number at radius ratio 2. It has been observe that as the Rayleigh number increases the Nusselt number parabolic-ally increases. This is due to the average Nusselt number is function of Pr and Ra number. This can also be explained by buoyancy effect and viscous force are dominant therefore nusselt number increases.

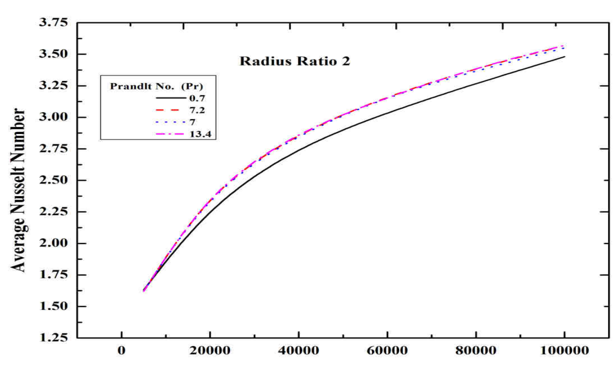

Figure 11 Effect of Rayleigh number (5x103 -105) on Nusselt number for varying Prandlt number at radius ratio 2

EFFECT OF NUSSELT NUMBER WITH RADIUS RATIO

In the figure 11 graph has been plotted between the avg. Nusselt No and the Radius Ratio at different Rayleigh No. Here we have taken Pr= 0.7 (for air) fixed as from the Figs.11 & 12 we have seen that there is no influence of Prandlt values up on Nusselt number variation. From the figure 6.7 we can see that:-

For every Ra. value there is a critical radius ratio. Up to which the value of Nusselt No increases and after that it starts decreasing.

As the Rayleigh number increases the critical radius ratio decreases eventually.

Both the value of critical rr increases when Rayleigh No. varies up to 103.

When Rayleigh number increases upto 103 the critical radius ratio increases.

From the computational analysis (FEV) the critical radius ratio has been evaluated for different values of Rayleigh number and it has been tabulated in table 5

Table 5 Critical radius ratio for different Rayleigh number

Ra value	Radius Ratio (Critical)
1000	3
5000	2.4
10000	2.2
50000	2
100000	2

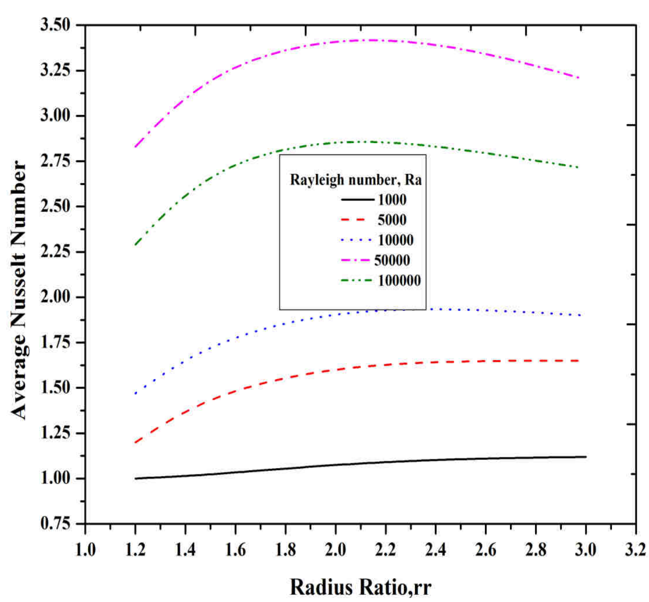

Figure 12 Effect of radius ratio on Nusselt number for different Rayleigh number.

Effect of Nusselt number

Figure 13 illustrates the Variation Nusselt Number at different angular position at radius ratio 2 and Pr=0.7 at different Ra number. It has been observed that the Nusselt number (Nuc, Nusselt number on cold sphere) significantly goes on increasing in various annular positions i.e. from 0 to 180.

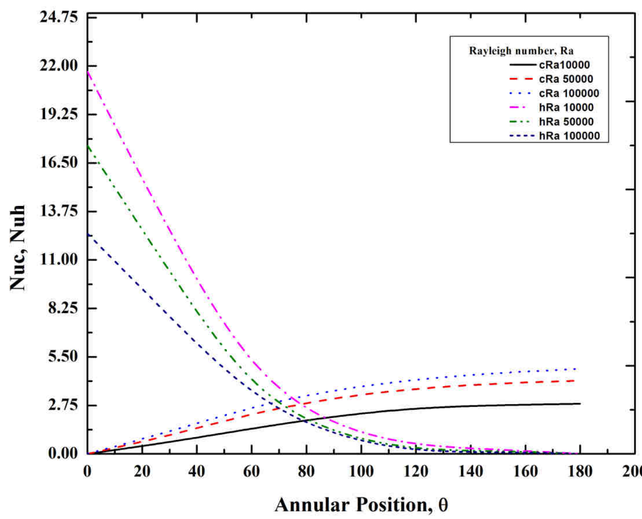

Figure 13 Steady local Nusselt number versus angular position at radius ratio 2 and Pr=0.7 at different Ra number.

Similarly for (Nuh, Nusselt number on hot sphere) the opposite trend has been observed. It has been observed that at inner sphere 00 annular position the conductive heat transfer increases due to conduction phenomenon while at outer sphere under goes convective heat transfer and significantly increases as angular position varies. At θ=180° Nuh increases and attain maximum at this point whereas Nusselt number of outer sphere (Nuc) decreases steadily as angular position increases. it attains minimum heat transfer at θ=180°.

The variation of Nuc and Nuh is quite declines because of heat boundary layer becomes wider as it shifts downward and generates stagnation area at the bottom of outer sphere.

CONCLUSION

The natural convection heat transfer of an two concentric isothermal spheres with Newtonian fluid has been investigated and thermal performance characteristics has been evaluated in terms of temperature and flow field which has been graphically illustrated. The obtained are compared with available literature and it has found that the result shows good agreement. Some other results are enlisted below

Increasing Rayleigh number at constant radius ratio, the average Nusselt number increases significantly

Increasing radius ratio Nusselt number increases first, till it reaches critical value and then starts decreasing at constant Rayleigh number.

The centre of main vortex, when stream function is maximum, moves upward with increase in both Rayleigh number and radius ratio.

Te effect of Prandlt number has very insignificant upon Nusselt number at constant radius ratio.

REFERENCES

[1] L. R. Mack and H. C. Hardee, "Natural convection between concentric spheres at Low Rayleigh numbers", Int. J. Heat Mass Transfer 11, 387-396 (1968).

[2] S.H. Yin, R.E. Powe, J.A. Scanlan, E.H. Bishop, "Natural convection flow patterns in spherical annuli", International Journal of Heat and Mass Transfer, Volume 16, Issue 9, September 1973, Pages 1785-1786,IN1-IN4,1787-1795

[3] R.W. Douglass, B.R. Munson, E.J. Shaughnessy,"Thermal convection in rotating spherical annuli—1. Forced convection", International Journal of Heat and Mass Transfer, Volume 21, Issue 12, December 1978, Pages 1543-1553

[4] Ralph E. Powe, Jack A. Scanlan, Thomas A. Larson, "Flow studies for natural convection in liquids between a sphere and its cubical enclosure", International Journal of Heat and Mass Transfer, Volume 20, Issue 2, February 1977, Pages 159-169

[5] Sanjay K. Roy, Subrata Sengupta, "A numerical study of natural convection heat transfer in a vertically eccentric spherical annulus", International Communications in Heat and Mass Transfer, Volume 15, Issue 5, September–October 1988, Pages 615-626

[6] Hsin-Sen Chu and Tzong-Shing Lee, "Transient natural convection heats transfer between Concentric spheres", Int.J. Heat and Mass transfer vol.36 N o. 13, pp.3159-3170(1993)

[7] H Asan ,"Natural convection in an annulus between two isothermal concentric square ducts", International Communications in Heat and Mass Transfer, Volume 27, Issue 3, April 2000, Pages 367-376

[8] Horng Wen Wu, Wen Ching Tsai, Huann-Ming Chou, "Transient natural convection heat transfer of fluids with variable viscosity between concentric and vertically eccentric spheres" International Journal of Heat and Mass Transfer, Volume 47, Issues 8–9, April 2004, Pages 1685-1700

[9] Wen Ruey Chen, "A numerical study of laminar free convection heat transfer between inner sphere and outer vertical cylinder", International Journal of Heat and Mass Transfer, Volume 50, Issues 13–14, July 2007, Pages 2656-2666

[10] Chung-Hyo Jung, and Takahiko Tanahashi, "Natural convection between concentric spheres in electromagnetic fields", Journal of Mechanical Science and Technology 22 (2008) 1202~1212

[11] B.V.K. Reddy, Arunn Narasimhan, "Heat generation effects in natural convection inside a porous annulus", International Communications in Heat and Mass Transfer, Volume 37, Issue 6, July 2010, Pages 607-610

[12] M. Jahanshahi, S.F. Hosseinizadeh, M. Alipanah, A. Dehghani, G.R. Vakilinejad , "Numerical simulation of free convection based on experimental measured conductivity in a square cavity using Water/SiO2nanofluid", International Communications in Heat and Mass Transfer, Volume 37, Issue 6, July 2010, Pages 687-694

[13] [13] A. Abouei Mehrizi, M. Farhadi, S. Shayamehr, "Natural convection flow of Cu–Water nanofluid in horizontal cylindrical annuli with inner triangular cylinder using lattice Boltzmann method", International Communications in Heat and Mass Transfer, Volume 44, May 2013, Pages 147-156

[14] M. Sheikholeslami, M. Gorji-Bandpy, D.D. Ganji, Soheil Soleimani, "Effect of a magnetic field on natural convection in an inclined half-annulus enclosure filled with Cu–water nanofluid using CVFEM", Advanced Powder Technology, Volume 24, Issue 6, November 2013, Pages 980-991

[15] M. Benmerkhi, M. Afrid, D. Groulx, Thermally developing forced convection in a metal foam-filled elliptic annulus International Journal of Heat and Mass Transfer, Volume 97, June 2016, Pages 253-269.