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Thermal Analysis of Natural Convection in an Annular Enclosure Partially Filled with Metal Foam: A Numerical Study Using ANSYS Fluent

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ABSTRACT

In this study presents a numerical investigation of natural convection heat transfer in an annular enclosure partially filled with metal foam. The inner cylinder is maintained at a constant temperature, while the outer cylinder is subjected to a uniform heat flux; both vertical sidewalls are thermally insulated. The fluid flow is modeled using the Navier–Stokes equations under the Boussinesq approximation, and the porous medium is described by the Brinkman–Forchheimer Darcy model, assuming local thermal equilibrium. Simulations are conducted using ANSYS FLUENT to examine the effects of pore density (10 PPI and 40 PPI), inclination angle (0°, 30°, 60°, 90°) and Rayleigh number ($10^4 \le Ra \le 10^6$) on thermal performance. Results indicate that the presence of metal foam significantly enhances heat transfer compared to the clear fluid case. The Nusselt number increases with both Rayleigh number and the inclination angle. Notably, the 10 PPI foam increases the average Nusselt number by approximately 50%, while the 40 PPI foam shows a 33% improvement compared to the clear fluid case. In all cases, temperature rises with increasing Rayleigh number, indicating stronger natural convection. Moreover, increasing the inclination angle from the horizontal configuration enhances heat transfer, with the most significant improvement between 60° to 90°.

Keywords: Natural convection, Metal foam, Horizontal annulus, Local thermal equilibrium, Numerical investigation.

1. INTRODUCTION

Natural convection in porous annuli has been previously studied mostly for vertical and horizontal annuli **(Bu-Xuan and Xing, 1990)** plays a crucial role in various engineering and scientific applications. These include geothermal systems, thermal insulation, chemical reactors with packed beds, porous heat exchangers, steam-assisted oil recovery, nuclear waste storage food preservation and the cooling of electronic devices **(Lu et al., 1998; Lu et al., 2006; Zhao et al., 2006)** such systems are widely studied to optimize heat transfer efficiency and understand the behavior of fluid flow and thermal characteristics in porous environments. **(Yasuyuki et al., 1984)** further explored the influence of inclination on

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natural convection in inclined annuli revealing that the average Nusselt number increased slightly with inclination angle. The maximum local Nusselt number, however, were strongly affected peaking at 75° for the inner cylinder and 60° for the outer cylinder due to viscous sharing forces near the ends at lower inclination angle. **(Prasad and Kulacki, 1985)** this research studies free convection heat transfer in short cylindrical annuli filled with saturated porous media. Using experimental and numerical methods, they demonstrated that Nusselt number predictions aligned with experimental data for Rayleigh number up to 4000. In subsequent studies **(Charrier-Mojtabi, 1997)** investigated two-dimensional free convection flows in porous horizontal annuli using the Dacry-Boussineqe equations. Their experimental and numerical work revealed bicellular flow structures that aligned with numerical predictions. **(El-Shazly, 2000)** free convection in annular enclosure filled with porous media and in systems divided horizontally between fluid and porous regions numerically investigation highlighting the influence of stratified structures on thermal behavior and offering insights into hybrid domain modeling strategies. **(Kiwan and Al-Nimr, 2000)** revealed that porous fins when designed with optimal porosity achieve similar heat transfer performance to traditional fins reducing material usage.

(Phanikumar and Mahajan, 2002) studied natural convection in high-porosity aluminum foams and highlighted enhanced heat transfer due to the porous structure. **(Aldoss et al., 2004)** followed the process of natural convection for double pipe heat exchanger in annular gap was partially filled with porous insulating material. The focus was to understand how heat transfer affected by the porous layer thickness, Darcy and Grasthof numbers with the efficiency evaluated using Nusselt number. Comparisons with other configurations established with fully and non-porous materials showed lesser efficiency for all conditions of testing indicating an important aspect of such system designs, the dominance of thermal insulation versus convective effectiveness. The researchers **(Braga and de Lemos, 2006)** studied turbulent natural convection heat transfer through an annular gap between two horizontal concentric cylinders filled with porous media using thermotical approach and two of the five turbulence models. They derived set of correlation equations governing heat transfer in the studied geometry and analyzed how the temperature distribution varies with the Rayleigh number. **(Mohammed, 2007)** performed experimental study on natural convection in vertical concentric cylinders deriving an empirical correlation between Nusselt and Rayleigh numbers.

Further, advancement was made by **(Khanafer et al., 2008)** studying the effect of porous sleeve on buoyancy-driven flow, concluding that the Rayleigh number significantly influenced heat transfer while the Prandtl number had a minimal effect. **(Hussein et al., 2009)** conducted numerical and experimental investigation of natural convection within vertical concentric annular cylinders filled with porous media. Their study explored how heat transfer is influenced by parameters such as Rayleigh number, radius ratio and porous structure properties. Their findings demonstrated a strong correlation between the Nusselt number and both geometric and thermal parameters with good agreement between experimental and theoretical results. **(Ali, 2012)** investigated theoretical and experimental studies on laminar and transient natural behavior were influenced by Rayleigh number and inclination angle. Theoretical analyses indicate that heat dissipation capacity is a function of the Rayleigh number with the Nusselt number showing a monotonic dependence on Ra under steady state conditions However, in transient conditions this relationship becomes inverse. Experimental findings further reveal that the nature of heat dissipation is influenced by the interaction between Nu and inclination angle. **(Saleh and Katea, 2013)** explored heat transfer in a three-dimensional annular enclosure filled with porous media, finding that the Nusselt number with Rayleigh number, fin dimensions and other geometric parameters. **(Qu et al., 2013)** studied natural convection in an annular region partially filled with metallic foam employing a two-equation model for non-equilibrium heat transfer. In their investigation model I placed the porous layer on the inner wall of the annulus while model II positioned it on the outer wall. Both models utilized the Forchheimer-Brinkman and local thermal non-equilibrium (LTNE) models to describe momentum and energy equations respectively. The study concluded that increasing the Rayleigh number and Prandtl number strengthened the convective performance resulting in an increase in the Nusselt number. Furthermore, by comparing the Nusselt number for the two models they observed that modeI consistently produced larger Nu values than model II, indicating the significance of the porous layers' positioning in enhancing heat transfer. **(Badruddin et al., 2014)** investigated heat transfer in a vertical annular cylinder filled with porous media, that the Nusselt number depended on radius ratio and interphase heat transfer coefficient. **(Sheremet, 2015)** provided a comprehensive analysis of transient conjugate natural convection in a vertical cylinder partially filled with porous media emphasizing the application of the Darcy and Brinkman extended Darcy models. The study focused on heat-conducting solid walls with finite thickness and convective heat exchange with the surrounding environment. The result served as valuable guidelines for utilizing these models in transient natural convection problems involving porous media. **(Massarotti et al., 2016)** validated numerical simulation for natural convection heat transfer in horizontal annuli. showing that empirical correlations effectively predicted thermal conductivity for small radii.

However, in turbulent regimes the correlations overestimated heat transfer emphasizing the need for turbulence models in transitional regions. **(Arpino et al., 2016)** study transient natural convection in partially porous vertical annuli, analyzing the effects of porous medium properties, layer positioning and cavity aspect ratio on heat transfer and flow behavior Using a generalized porous medium model solved by the artificial compressibility characteristic-Based split (AC-CBS) algorithm integrating the porous media and they integrated porous and fluid regions into a unified framework. The study examined key parameters such as the Darcy number, Rayleigh number, and radius ratios. Their findings demonstrated that the position and properties of the porous layer significantly influenced thermal behavior. For low Darcy numbers, conduction dominated, promoting stability while, high Darcy numbers led to chaotic oscillations, indicating complex flow behavior. The authors concluded that the design and placement of the porous layer were crucial for optimizing heat transfer and stability, with potential applications in geothermal systems and heat exchangers. Numerically examined unsteady conjugate natural convection and entropy generation in a semicircular porous enclosure bounded by a solid wall of finite thickness **(Chamkha et al., 2018)**. Their results highlighted the influence of Rayleigh and Darcy numbers, thermal conductivity ratio, and time on flow structure, heat transfer, and entropy generation. conclude that higher thermal conductivity ratios will increase the Bejan number and total entropy due to reduced heat loss through the solid wall. A Numerical study on heat transfer performance of vacuum tube solar collector integrated with metal foams studies by **(Lu and Chen, 2019)**. Results showed that these metal foam inserts helped distribute heat more evenly, reduced thermal resistance and speed up the heating of the working fluid. The effect was especially noticeable when using foams with lower porosity and higher (PPI) values pointing to a promising strategy for making solar thermal systems more energy efficient. **(Hamzah and Nima, 2020)**. examined the effectiveness of 40 PPI copper metal foam fins positioned at a 30° angle within the annular space of a double pipe heat exchanger experimentally. The fins were designed to disrupt airflow and enhance thermal performance. The results revealed that using foam fins improved heat transfer rates particularly under counterflow conditions while maintaining low pressure drop. The study highlights that open cell metal foams can effectively intensify convective heat transfer by promoting fluid mixing and disrupting thermal boundary layers. **(Mahmood et al., 2020)** focused on natural convection heat transfer in a concentric vertical cylindrical annulus embedded with porous media. The study utilized numerical simulations based on the finite volume method and the SIMPLE algorithm to solve the governing Navier-Stokes and energy equations. Key parameters explored included Rayleigh number. heat flux (300–450 W/m^2) and diameter ratios (r/R = 0.12, 0.20 and 0.26). The results revealed that the Nusselt number increased with higher Rayleigh numbers and larger diameter ratios highlighting the importance of these parameters in enhancing heat transfer. The study also demonstrated that the presence of a porous medium significantly improved natural convection with empirical correlations established between Nusselt number, Rayleigh number and diameter ratios. These findings offer valuable insights for optimizing heat transfer in systems that incorporate porous media. An experimental study of natural convection heat transfer from a vertical cylinder using porous fins was presented by **(Kiwan et al., 2020)**.

Two types of porous material and three diameters of aluminum cylinders with varying fin configurations, such as fin thickness and number were employed. The results revealed that the permeability (K) had a significant impact on the average Nusselt number. Using porous fins led to substantial improvements in heat transfer performance. Furthermore, a single fin with a thickness of 10 mm enhanced the heat transfer coefficient by 7.9% and up to 131% when a porous layer was applied. **(Poursharif et al., 2022)** carried out turbulent heat transfer of non-Newtonian fluid flow in an annular pipe embedded with porous discs in numerical method. Their study revealed that variations in power-law index, porous disc pitch and porous layer thickness significantly affect thermal performance. porous media significantly enhance heat transfer at the expense of increased flow resistance. By reviewing previous studies, it is noted that most investigations on natural convection in annular enclosures with metal foam have focused on fully porous domains, vertical or inclined configurations or conventional boundary conditions. Very few have addressed the case of a horizontal annulus with a cooled inner cylinder and a constant heat flux applied to the outer wall.

In this paper, a three-dimensional numerical study of natural convection in an annular space between two concentric horizontal cylinders partially filled with metal foam was introduced. Copper foam distributed in discrete annular rings along the axial direction has not been previously examined in natural convection. These foam discs are inserted in the inner surface of outer pipe at equal intervals along the axial direction allowing for localized thermal enhancement while minimizing flow resistance and material usage. To model buoyancy-driven flow the Boussinesq approximation is employed, wherein density variations are considered negligible except in the gravitational term. The flow within the porous medium is described using the Brinkman–Forchheimer Darcy model, which accounts for both viscous shear and inertial resistance making it suitable for high-porosity metal foam structures. All simulations were conducted using ANSYS Fluent 2022R2.

2. MATHEMATICAL MODELLING

2.1. Problem Description

Fig. 1 illustrates the physical configuration of the studied geometry, which consists of two concentric cylinders. The inner cylinder is maintained at a constant temperature, whereas the outer cylinder is exposed to a constant heat flux (q) with the side walls thermally insulated. Five copper foam rings are inserted along the outer cylinder in different sections. These foam discs are positioned at equal intervals along the axial direction, allowing for localized thermal enhancement while preserving flow continuity between segments.
The thickness of the metal foam (δ_p) is 1 cm, and the height (H_p) is 2 cm. The total length of the annular test section is 60 cm, with the inner and outer cylinders having diameters of 5 cm and 10 cm, respectively. The flow is assumed to be steady, 3D, laminar and incompressible. Thermophysical properties are constant except for fluid density, which is governed by the Boussinesq approximation. The metal foam is modeled as isotropic and homogeneous with a porosity of 0.9 and two different pore densities (10 and 40 PPI) are considered.

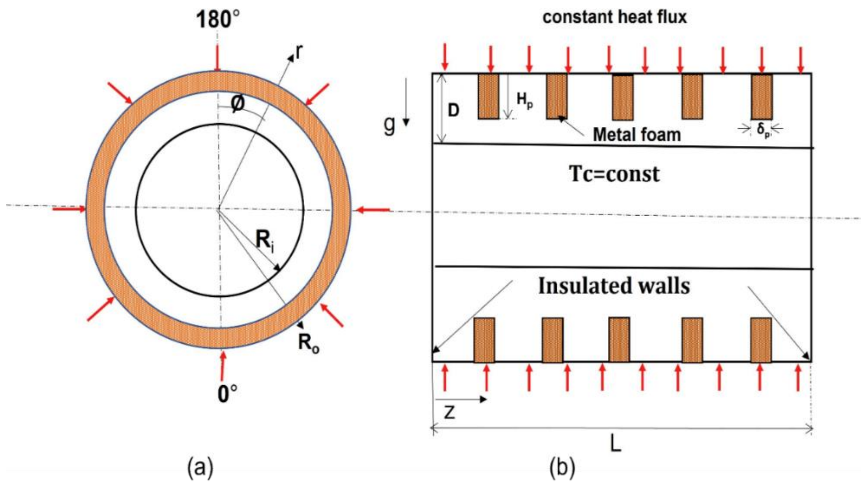

Figure1. Schematic of the studied geometry (a): cross section of the annular enclosure in the porous region (b): longitudinal section.

2.2 Governing Equations

Considering the above hypothesis, the dimensionless form of the governing equations for both clear and copper foam regions can be written as:

2.2.1 Fluid Region

The standard Navier-Stokes and energy equations apply **(Wei and Tao, 1996)**

- **Continuity Equation**: (for both clear and porous regions)

$$\frac{1}{R}\frac{\partial(RV_R)}{\partial R}+\frac{1}{R}\frac{\partial V_\phi}{\partial \phi}+\frac{\partial V_Z}{\partial Z}=0 \tag{1}$$

- **Momentum Equations:**

-Radial Momentum:

$$V_R\frac{\partial V_R}{\partial R}+\frac{V_\phi}{R}\frac{\partial V_R}{\partial \phi}+V_Z\frac{\partial V_R}{\partial Z}=-\frac{\partial \mathrm{p}}{\partial R}+\mathrm{Pr}\left(\nabla^2 V_R-\frac{V_R}{R^2}-\frac{2}{R^2}\frac{\partial V_\phi}{\partial \phi}\right)+\mathrm{Ra}_q\Theta\cos(\phi) \tag{2}$$

-Azimuthal Momentum:

$$V_R \frac{\partial V_\phi}{\partial R} + \frac{V_\phi}{R}\frac{\partial V_\phi}{\partial \phi} + V_Z \frac{\partial V_\phi}{\partial Z} = -\frac{1}{R}\frac{\partial p}{\partial \phi} + \Pr\left(\nabla^2 V_\phi - \frac{V_\phi}{R^2} + \frac{2}{R^2}\frac{\partial V_R}{\partial \phi}\right) - \mathrm{Ra}_q \Theta \sin(\phi) \quad (3)$$

-Axial Momentum:

$$V_R \frac{\partial V_Z}{\partial R} + \frac{V_\phi}{R}\frac{\partial V_Z}{\partial \phi} + V_Z \frac{\partial V_Z}{\partial Z} = -\frac{\partial p}{\partial Z} + \Pr \nabla^2 V_Z \quad (4)$$

- **Energy Equation:**

$$\frac{\partial \Theta}{\partial R} + \frac{V_\phi}{R}\frac{\partial \Theta}{\partial \phi} + V_Z \frac{\partial \Theta}{\partial Z} = \frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial \Theta}{\partial R}\right) + \frac{1}{R^2}\frac{\partial^2 \Theta}{\partial \phi^2} + \frac{\partial^2 \Theta}{\partial Z^2} \quad (5)$$

2.2.2 Metal Foam Region

For the porous metal foam, the Darcy-Brinkman-Forchheimer model and the Local Thermal Equilibrium (LTE) **(Nield and Bejan, 2017)**.

- **Momentum Equations:**

-Radial Momentum:

$$V_R \frac{\partial V_R}{\partial R} + \frac{V_\phi}{R}\frac{\partial V_R}{\partial \phi} + V_Z \frac{\partial V_R}{\partial Z} = -\frac{\partial p}{\partial R} + Pr\left(\nabla^2 V_R - \frac{V_R}{R^2} - \frac{2}{R^2}\frac{\partial V_\phi}{\partial \phi}\right) - \frac{Da^{-1} V_R}{\epsilon^2} - C_f \frac{V_R^2}{\sqrt{Da}} + Ra_q \Theta \cos(\phi) \quad (6)$$

-Azimuthal Momentum:

$$V_R \frac{\partial V_\phi}{\partial R} + \frac{V_\phi}{R}\frac{\partial V_\phi}{\partial \phi} + V_Z \frac{\partial V_\phi}{\partial Z} = -\frac{1}{R}\frac{\partial p}{\partial \phi} + \Pr\left(\nabla^2 V_\phi - \frac{V_\phi}{R^2} + \frac{2}{R^2}\frac{\partial V_R}{\partial \phi}\right) - \frac{\mathrm{Da}^{-1} V_\phi}{\epsilon^2} - C_f \frac{V_\phi^2}{\sqrt{\mathrm{Da}}} - \mathrm{Ra}_q \Theta \sin(\phi) \quad (7)$$

-Axial Momentum:

$$\mathrm{V_R} \frac{\partial \mathrm{V_Z}}{\partial \mathrm{R}} + \frac{\mathrm{V_\phi}}{\mathrm{R}}\frac{\partial \mathrm{V_Z}}{\partial \phi} + \mathrm{V_Z} \frac{\partial \mathrm{V_Z}}{\partial \mathrm{Z}} = -\frac{\partial \mathrm{p}}{\partial \mathrm{Z}} + \Pr \nabla^2 \mathrm{V_Z} - \frac{\mathrm{Da}^{-1}\mathrm{V_Z}}{\epsilon^2} - \mathrm{C_f}\frac{\mathrm{V_Z^2}}{\sqrt{\mathrm{Da}}} \quad (8)$$

- **Energy Equation**

$$V_R \frac{\partial \Theta}{\partial R} + R V_\phi \frac{\partial \Theta}{\partial \phi} + V_Z \frac{\partial \Theta}{\partial Z} = \frac{\mathrm{k_{eff}}}{\mathrm{k_f}}\left[\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial \Theta}{\partial R}\right) + \frac{1}{R^2}\frac{\partial^2 \Theta}{\partial \phi^2} + \frac{\partial^2 \Theta}{\partial Z^2}\right] \quad (9)$$

while k_{eff} the effective thermal conductivity can be found from **(Nield and Bejan, 2013)**

$$k_{\mathrm{eff}} = \varepsilon k_f + (1-\varepsilon) k_s \quad (10)$$

- ε*: Porosity of the metal foam.*
- Da: *Darcy number* ($\mathrm{Da} = K/r_2^2$, *with K as permeability*).
- Cf:*Forchheimer coefficient (accounts for inertial resistance in the foam).*
- kf:Thermal conductivity of the fluid.
- ks: Thermal conductivity of the solid metal foam.

Where K is Permeability for metal foams, it is often expressed as: **(Ergun, 1952)**

$$K = \frac{\varepsilon^3}{(1-\varepsilon)^2} \cdot \frac{D_p^2}{150} \quad (11)$$

D_p is the pore diameter in the case of the metal foam and can be calculate as follows **(Iranmanesh and Moshizi, 2024).**

$D_p = 0.0254/PPI$

The properties of copper foam such as permeability, porosity, fiber diameter(d_f) and pore diameter (D_p) are shown in **Table1**.

Table 1. Properties of metallic foam

Pores per inch (PPI)	Permeability (k)	Fiber size df (m)	Pore size Dp (m)	Porosity (ε)%
10	3.1×10^{-6}	3.3×10^{-4}	2.54×10^{-3}	90
40	1.95×10^{-7}	8.05×10^{-5}	6.35×10^{-4}	90

2.2.3 Dimensionless Variables

To non-dimensional equations, the following transformations are used:

$$R = \frac{r}{D} \quad , Z = \frac{z}{L} \quad , \Theta = \frac{T - T_{ref}}{qD_2/k} \quad , V_R = \frac{V_R\, D}{\alpha}, V\Theta = \frac{V\Theta\, D}{\alpha}, Vz = \frac{Vz\, D}{\alpha}\ P = \frac{P}{\rho \nu^2/D^2}\ ,\ D = r_o - r_i$$

And the Ra number and the Nusselt number can be expressed as follows **(Kiwan and Alzahrany, 2008)**

$$Ra_q = \frac{g\beta q D^4}{\nu \alpha kf} \quad , \quad Pr = \frac{\nu}{\alpha} \quad , Nu = \frac{hD}{kf} \tag{12}$$

2.2.4. Boundary Conditions

1. $\Theta = 0 \quad at \quad (R = R_1)$ (13)

2. $\frac{\partial \Theta}{\partial R} = -1 \quad at \quad (R = R2)$ (14)

3. $\frac{\partial \Theta}{\partial \phi} = 0 \quad at \quad (\phi = 0\,, \pi)$ (15)

4. $\frac{\partial \Theta}{\partial Z} = 0 \quad at \quad (Z=0, L)$ (16)

5. Interface Between Fluid and Metal Foam Regions:

Continuity of Temperature $\Theta_{\text{fluid}} = \Theta_{\text{foam}}$ (17)

Continuity of Heat flux $k_{\text{fluid}} \frac{\partial \Theta}{\partial R}\Big|_{\text{fluid}} = k_{\text{foam}} \frac{\partial \Theta}{\partial R}\Big|_{\text{foam}}$ (18)

Continuity of Velosity Components $V_{\text{fluid}} = V_{\text{foam}}$ (19)

Shear Stress Continuity $\mu_{\text{fluid}} \frac{\partial V}{\partial R}\Big|_{\text{fluid}} = \mu_{\text{foam}} \frac{\partial V}{\partial R}\Big|_{\text{foam}}$ (20)

And thermophysical properties for both air and copper referred to Temperature at T=300K are taken from **(Holman, 2010)** in **Table 2.**

Table 2. Thermophysical properties

Properties	Air	Copper(cu)
density (kg/m^3)	1.22	8978
Thermal conductivity (W/(m.K))	0.026	387.6
Kinematic viscosity (m^2/s)	$1.59x10^{-6}$	------------
Specific heat (J/(kg .K))	1006	385

3. NUMERICAL PROCEDURE

3.1 Numerical Details

The numerical approach is employed to analyze natural convection within the annular system, utilizing ANSYS Fluent for computational simulations. The finite volume method (FVM) implemented in Fluent is adopted to discretize the governing equations. A pressure-based algorithm with the steady-state option is used to model the laminar, incompressible flow behavior. The geometric model is created in SolidWorks for both configurations with and without metal foam, where only half of the annulus is simulated to reduce computational cost while preserving accuracy through symmetric considerations. A mesh independence study was conducted to ensure the reliability of the computational results, leading to the selection of a refined mesh containing 460,000 elements, shown in **Fig. 2**. Convergence criteria were set to less than (1×10^{-4}) for continuity and momentum residuals, while set to less than (1×10^{-8}) for energy. The annular with porous discs is considered to examine the grid independence of the research geometry and the average Nusselt number. is studied for the mentioned geometry and the values of $Ra=10^4$, 10ppi, Ø=0.The results of the grid independence evaluation are demonstrated in **Fig. 3**. The local thermal equilibrium model is applied to the metal foam, and air was used as the working fluid.

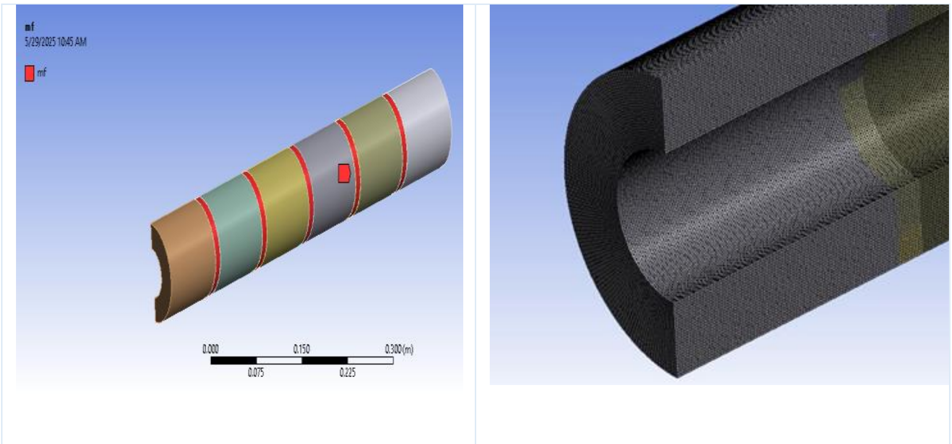

Figure 2. Mesh grid in Ansys fluent.

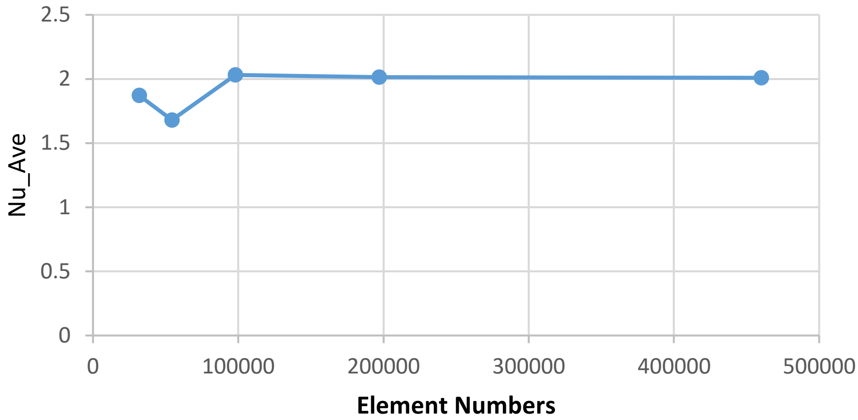

Figure 3. The Nuav Over Various Mesh Sizes.

3.2 Validation

A numerical analysis was conducted to solve the conservation equations of mass and energy, simulating free convection within an annular enclosure. The finite volume method (FVM) was employed to evaluate the effects of modified Rayleigh number, inclination angle and metal foam pore density on heat transfer and flow behavior. To validate the accuracy of the simulation for the under-study case a comparison was made with the results presented by **(Yasuyuki et al., 1984)** for a horizontal annular enclosure at Ra = 10^5 and an inclination angle of Ø=0 and Z=90mm. **Fig. 4** presents the comparison of isotherms contours., in convective cell formation and temperature gradients at Ra = 10^5 and Ø= 15 the model reliably reproduced thermal structures along the axial direction and the local Nusselt number distributions followed similar trends as shown in **Figs. 5 and 6.** Another comparison was performed using the result of **(Khanafer et al., 2008)** at Ra = 7.5× 10^5 and Ra = 10^5, r_{porous}/r_i=1.5, **Fig. 7** show the results of comparison for isotherms contours **. Table 3** shows the result of this comparison for Nusselt number. These comparisons showed good agreements between the present result and those reported in previous studies.

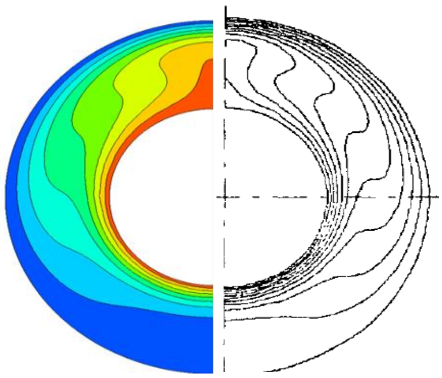

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Figure 4. Comparison of the current study with **(Yasuyuki et al., 1984)** for isotherms contours.

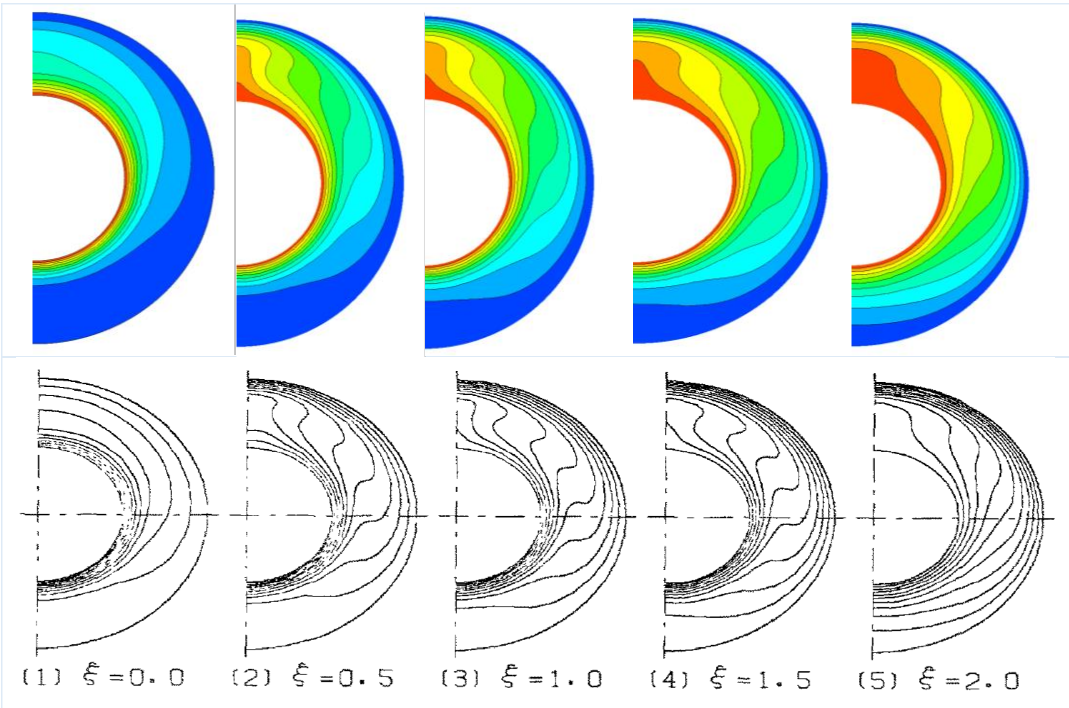


Figure 5. Comparison of the current study with **(Yasuyuki et al., 1984)** Isothermal lines.

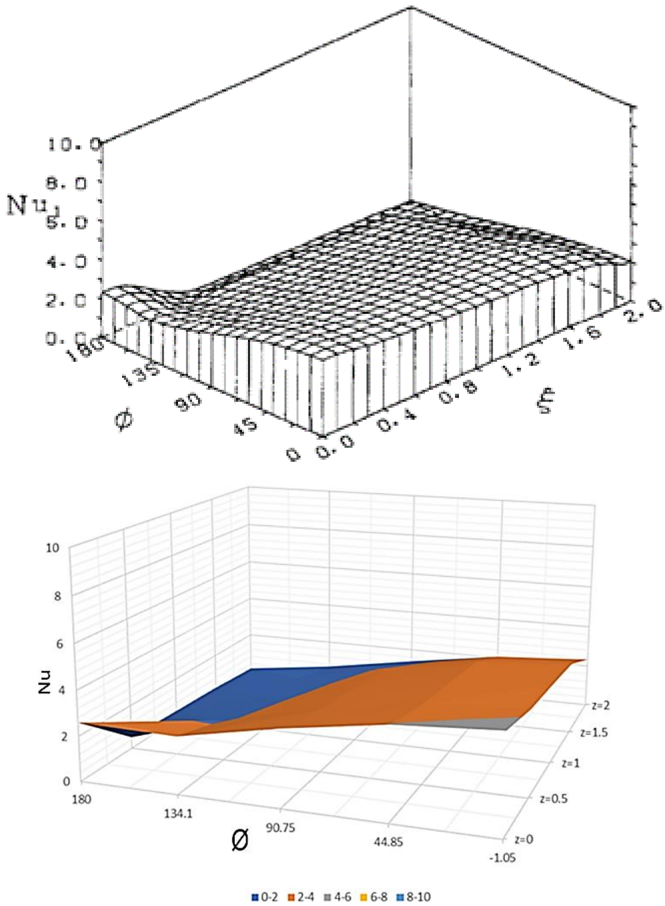


Figure 6. Comparison with **(Yasuyuki et al., 1984)** for Local Nusselt numbers at Ra 10^5,

Ø=15°.

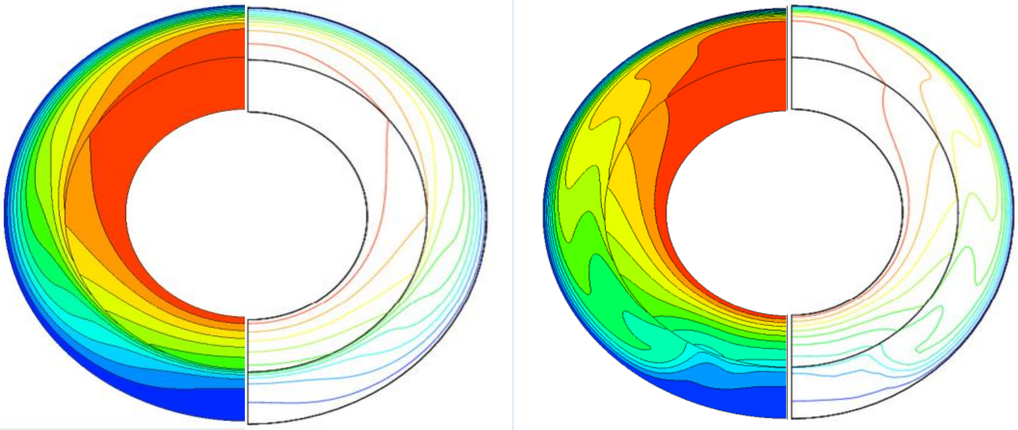


Figure 7. Comparison with **(Khanafer et al., 2008)** for isotherms contours.

Table 3. Nusselt number of the present work and the corresponding researchers work.

Yasuyuki	Present study	Khanafer	Present study
2.1	2.25	8	8.20

4. RESULTS AND DISCUSSION

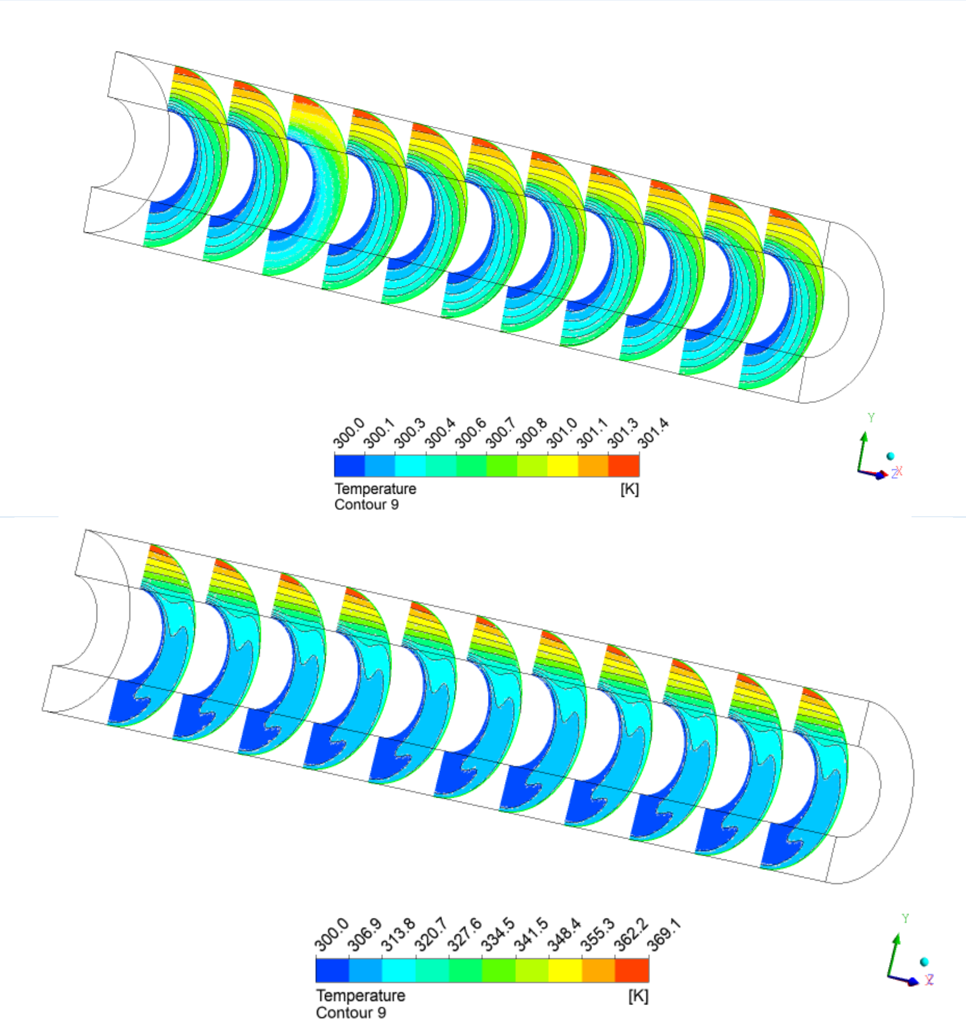
The study investigates the effect of metal foam on natural convection heat transfer within an annular under different Rayleigh numbers (Ra) and inclination angles (Ø). The results are analyzed through temperature contours and Nusselt number distributions for clear and porous cases (10 PPI and 40 PPI).

4.1 Effect of Metal Foam on Temperature Distribution

The analysis of isothermal contours in clear annular case at different Rayleigh numbers provides insight into the transition from conduction dominated to convection enhanced heat transfer. At Ra 10^4, heat transfer is primarily governed by conduction and natural convection effects weak. The temperature contours exhibit smooth transition and closely spaced, considering a relatively uniform thermal distribution with limited fluid motion. heat transfer mechanism in this regime is characterized by steady diffusion of heat from the inner to the outer surface, with minimal disruption by buoyancy forces, shown in **Fig. 8a**

As the Rayleigh number increases to Ra 10^6, the thermal and fluid behavior within the annulus undergoes a significant transformation. The contours in this case become more irregular and widely spaced, because of the effect of buoyancy-driven convection currents. Enhanced convective mixing leads to intensified heat transfer and more pronounced temperature variations across the domain. higher Ra numbers will increase the instability of flow inside the system and cause a vortex that will enhance the thermal energy exchange. With improving the overall heat transfer efficiency of the system, shown in **Fig. 8b**

Utilized metal foam with different pore densities significantly influences the thermal and fluid behavior within an annular system, particularly under varying Rayleigh numbers. In all cases the temperature decreases inside foam fins and increases in the gaps after it, the effective thermal conductivity is much higher in foam, so heat transfer is enhanced, and the temperature gradient decreases in variation. The 10 PPI copper foams have larger pores with higher permeability, enabling better fluid diffusion and reducing thermal resistance. They enhance convection and conduction by increasing the contact surface area between the fluid and the foam structure. while 40PPI has a smaller pore with lower permeability, with increasing flow resistance, although promoting stronger turbulence. This has resulted in an improvement in heat transfer efficiency but with a higher pressure drop. for Ra 10^4, the isotherms in both 10 and 40PPI conduction remain the dominant mode of heat transfer. By increasing Ra, it can be noted that convection becomes more effective and increases, but in 10PPI maintains smoother isotherms, whereas 40PPI exhibits more irregular thermal patterns. These results confirm that the metal foam fins significantly impact local temperature distribution, creating distinct thermal gradients. This demonstrates their effectiveness in enhancing heat transfer between the inner and outer pipes, making them beneficial for thermal management applications, as shown in **Fig. 9(a) and (b).**



(b)

Figure 8. Isothermal contours for the horizontal annular without metal foam (a) at Ra=10^4, (b) Ra = 10^6.

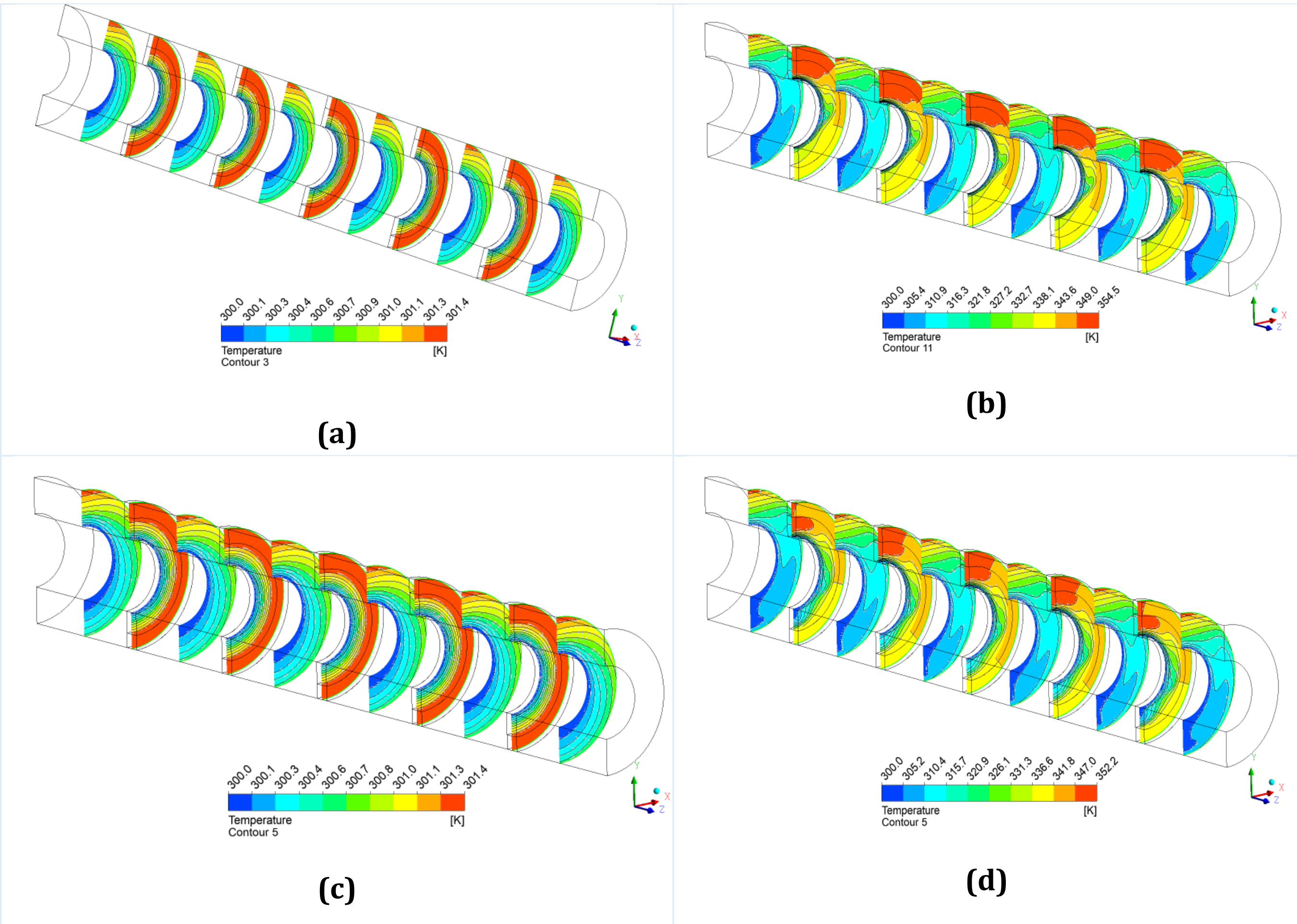


Figure 9. Isothermal contours for the horizontal annular with metal foam 10ppi (a) at Ra=10^4, (b) Ra = 10^6, 40ppi, (c) at Ra=10^4, (d) Ra = 10^6.

4.2 Influence of Metal Foam on Nusselt Number

4.2.1 Local Nusselt Number Along Annular Length

The local Nusselt number varies along the annulus for different inclination angles (Ø=0°,30°,60°,90°) to show the effect of orientation on convection.

- When the system is horizontal (Ø=0°) the development of symmetrical convective cells shows a periodic peak in the local Nu number specially in clear fluid and 10 PPI metal foam case attributed to the formation of recirculating cells. The metal foam dampens these oscillations, and 10 PPI foam has higher local Nu than 40 PPI as shown in **Fig. 10(a)**
- As the inclination increases to 30° and 60° the periodic nature of Nu remains observable in the clear case though its intensity reduces. The porous case is relatively steady with moderate enhancement as shown in **Fig. 10 (b) and (c)**
- At Ø=90°(vertical), the heat transfer mechanism shifts towards dominance by buoyancy-driven convection. The clear case shows a reduction in peak fluctuations, while the metal foam cases exhibit more uniform behavior with enhanced heat transfer as shown in **Fig. 10(d)**

4.2.2 Overall Nusselt Number

Fig. 11 illustrates the variation of the average Nusselt number with Ra, demonstrating that heat transfer is enhanced with increasing Rayleigh number as expected. For a given Ra the presence of metal foam (both 10 PPI and 40 PPI) results in a higher value compared to the clear case. The 10 PPI metal foam exhibits the highest Nusselt number when compared to other metal foams.

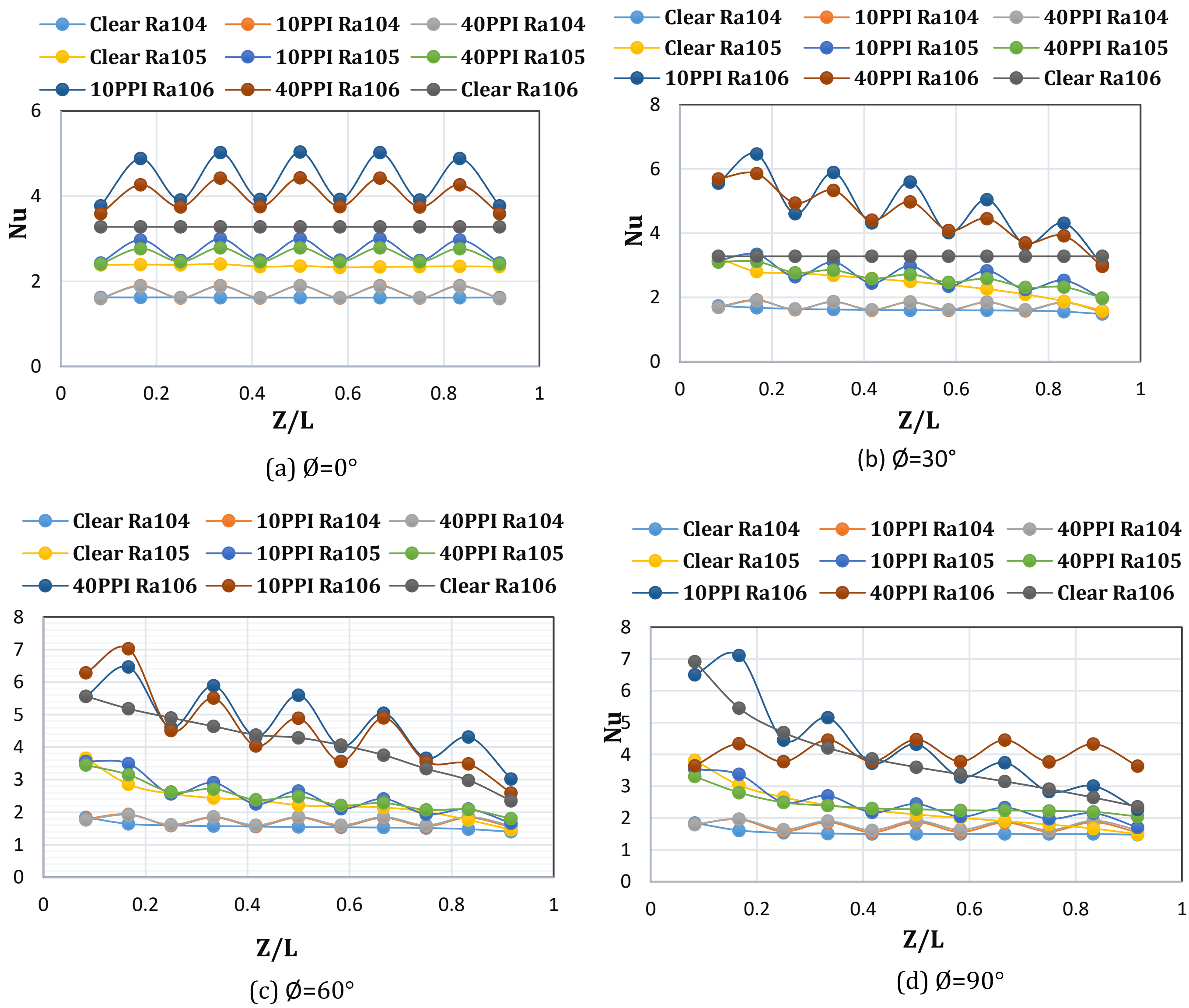


Figure 10. The local Nusselt number varies along the annulus for different inclination angles (Ø=0°,30°,60°,90°).

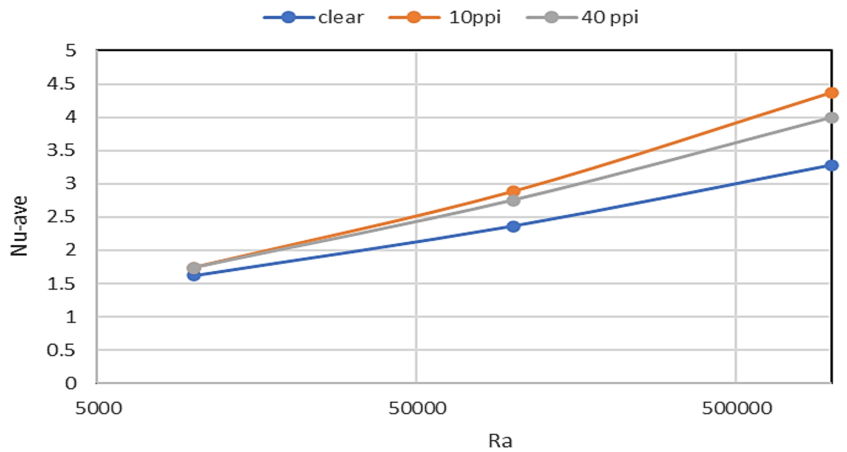


Figure 11. Average Nusselt Number for annular with and without metal foam.

5. CONCLUSIONS

Natural convection heat transfer in a two-concentric cylinder system with and without metal foam was numerically investigated under a three-dimensional steady-state regime. The inner cylinder was maintained at a constant temperature, while the outer cylinder was subjected to a uniform heat flux with insulated end plates. Discrete annular rings of copper metal foam were embedded at equal axial intervals to enhance thermal performance while minimizing material usage and cost. The convective behavior within the porous medium was modeled using the Brinkman–Forchheimer-extended Darcy equation under the local thermal equilibrium assumption. Comparative analysis of the annulus with and without metal foam revealed the following:

1. Metal foam can considerably enhance heat transfer performance and the temperatures increase in both cases with and without metal foam when Ra number increases.
2. The 10 PPI foam outperforms 40 PPI due to its lower pressure drop at constant porosity.
3. Temperature gradients along the heated wall are reduced in foam-filled sections due to higher thermal diffusivity.
4. Average Nusselt number increases with Rayleigh number in all cases.
5. Increasing the inclination angle from 60° to 90° yields the greatest improvement in convective heat transfer for both clear and foam-filled setups.

As a part of future research work will focus on

- Applying discrete heat sources directly to segmented foam rings to improve thermal efficiency and reduce energy and material costs.
- Extending porous fin placement to both inner and outer cylindrical surfaces for enhanced performance.
- Investigating fin positioning while keeping the total number constant, to assess its impact on local convection and overall heat transfer.

NOMENCLATURE

Symbol	Description	Symbol	Description
df	Fiber diameter, m	PPI	Pore per inch
dp	Pore diameter, m	Z	Dimensionless axial coordinate z/L
g	Gravitational acceleration	α	Thermal diffusivity m^2/s
h	Local heat transfer coefficient, $W/m^2.^{\circ}C$	β	Volume expansion coefficient (K−1)
H_p	Height of metal foam, cm	Ø	Angle of inclination, deg
K	Permeability of the metal m^2	δ_p	Thickness of metal foam, cm
D	Gap between inner and outer cylinder, cm	ρ	Density, kg/m^3
Nu	Nusselt number	υ	Kinematic viscosity $m^2.\ s^{-1}$
Pr	Prandtl number	ε	Porosity
q''	Heat flux, W/m^2	Θ	Dimensionless temperature
R	Dimensionless radial coordinate, r/D, R1=Ri/D, R2=Ro/D	eff	Effective
Ra	Rayleigh number	f	fluid
r	Radial coordinate	i	inner
T	Temperature, °C	o	outer
V_R	Nondimensional radial velocity	p	Porous
V_Z	Nondimensional axial velocity	s	solid
V_Θ	Nondimensional tangential velocity		

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Credit Authorship Contribution Statement

Shahad Mahdi Saleh: Writing – review & editing, Writing – original draft, Validation, Software, Methodology. Luma Fadil Ali: Supervising and following up the research.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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دراسة عدديه لأنتقال الحرارة بالحمل الحراري في حيز حلقي مملوء جزئيا برغوه معدنيه

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الخلاصة

في هذه الدراسة، تم التحقيق عدديًا في انتقال الحرارة بالحمل الطبيعي داخل حيز حلقي ممبوء جزئيا بالرغوة معدنية، حيث يتم الحفاظ على درجة حرارة ثابتة للأسطوانة الداخلية، بينما تتعرض الأسطوانة الخارجية لتدفق حراري منتظم، وتُعزل الجدران الجانبية. تم تمثيل ديناميكا الموائع باستخدام معادلات نافير–ستوكس مع افتراض **(Boussinesq)** بينما وُصف سلوك الرغوة المعدنية باستخدام نموذج Brinkman– Forchheimer Darcy بالإضافة إلى ذلك، تم اعتماد فرضية التوازن الحراري المحلي في معادلة الطاقة الخاصة بالوسط المسامي. أُجريت المحاكاة باستخدام برنامج ANSYS FLUENT. لدراسة تأثير كثافة المسام (10 و40 وحدة لكل بوصة – PPI)، وزوايا الميل (0°، 30°، 60°، 90°)، وأرقام رايلي ($10^4 \leq 10^6 \geq$ Ra) على الأداء الحراري. أظهرت النتائج أن وجود الرغوة المعدنية يعزز انتقال الحرارة بشكل ملحوظ مقارنةً بالحالة الخالية من الوسط المسامي. كما تبين أن عدد نوسلت يزداد مع زيادة كل من رقم رايلي وزاوية الميل. وقد تبين أن رغوة 10 PPI تُحقق زيادة في متوسط عدد نسلت بنسبة تقارب 50%، بينما تُظهر رغوة 40 PPI تحسنًا بنسبة 33% مقارنةً بالحالة الصافية.

وفي جميع الحالات، ترتفع درجة الحرارة مع زيادة عدد رايلي، مما يدل على قوة الحمل الحراري الطبيعي. علاوةً على ذلك، فإن زيادة زاوية الميل تؤدي إلى أعلى تحسين في انتقال الحرارة خاصه عند زاوية 60° إلى 90°.

الكلمات المفتاحية: الحمل الحراري الطبيعي، الرغوة المعدنية، الحلقة الأفقية، الاتزان الحراري المحلي، الدراسة العددية.