

Parametric studies of mixed convective fluid flow around cylinders of different cross-sections

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Abstract

A numerical study of mixed convective heat transfer in a lid-driven square enclosure containing a hot elliptic cylinder is conducted. The impacts of the Grashof number ($10^3 \leq Gr \leq 10^6$), Reynolds number ($1.0 \leq Re \leq 100$), cylinder tilt angle ($0^\circ \leq \phi \leq 90^\circ$), and aspect ratio ($1.0 \leq AR \leq 3.0$) have been examined for a fluid of Pr of 0.71. The horizontal enclosure walls are insulated, while its vertical walls are restricted to a nonvarying temperature T_c , whereas a sinusoidal temperature of $T_h + \Delta T \sin(\pi x/L)$ is imposed on the wall of the elliptical cylinder. The governing equations are solved using COMSOL Multiphysics 5.6 software. The fluid dynamic and the heat transport profiles between the enclosure and the elliptical cylinder walls are represented by the stream function, isothermal contours, and average Nusselt number. Results established that for all the considered aspect ratios, the thermal heating range of $10^3 \leq Gr \leq 10^4$ is predominantly a conduction mechanism. The critical position of the ellipse where the inclination effect becomes insignificant is determined by the Grashof number and aspect ratio when the $Re = 100$. The strength of vortices

and cell numbers are significantly influenced by the aspect ratio, particularly when the $Gr = 10^4$. When $AR = 1.0$, the average heat transfer from the cylinder remains the same regardless of the cylinder's orientation. The impact of cylinder orientation on heat transfer from the cylinder wall is minimal for $1.5 \leq AR \leq 2.0$. For AR values of $2.5 \leq AR \leq 3.0$, increasing the inclination angle does not result in improved heat transfer. The influence of the increasing inclination angle on the right wall diminishes as the angle increases, except when the Grashof number is greater than 10^5 , where the rate of heat transfer is enhanced for inclination angles beyond 45° .

KEYWORDS

aspect ratio, geometric parameters, lid-driven, mixed convective heat transfer

1 | INTRODUCTION

Over the years, understanding the dynamics of flows in a lid-driven enclosure for different scenarios of flow regimes has been extensively discussed by researchers. This is because of the ability of flows in enclosures to effectively provide cooling solutions to systems.^{1–6} The popularity enjoyed by the bounded cavity problem among researchers is not only due to the simplicity of the geometry but also due to the practical significance of the subject. Furthermore, the associated flow field in this problem is used as a tool for checking the accuracy of new numerical codes being advanced, due to the ability to enact basic behavioral tendencies in incompressible flows, such as chaotic particle motion, transition, turbulence, three-dimensional patterns, eddies, and so forth.^{7–9}

Consequently, as a result of the challenge of effectively increasing heat transfer, reducing thermal load, and boosting system effectiveness in several industrial and technical applications, including solar energy storage, mixing operations, food processing technology, drying chambers, coating processes, and microelectronic device cooling, there have been enormous research contributions on the numerical study of mixed convection flow in enclosures subjected to differential heating conditions.^{10–14}

Using two magnetic field configurations, thermomagnetic convection and entropy analysis in a ferrofluid-filled cubical cavity with a cylindrical heat source, as well as magnetohydrodynamic (MHD) mixed convection flow and heat transfer of a Cu–water nanofluid in a square cavity filled with a porous media, have been studied.^{15,16} The results demonstrate that both designs significantly improve cooling, with the double-Halbach array configuration performing somewhat better.¹⁵ It is discovered that the ideal cooling is achieved with a middle magnetic Nusselt number. Average heat transmission initially falls as the slip parameter is raised but eventually stabilizes for all Richardson number values.¹⁶

Using an adiabatic block, Biswas et al.¹⁷ looked at the thermal energy transfer in a differentially heated cavity. The findings demonstrated that thermal energy transmission improves up to a certain block size, with opposite flow scenarios showing the most excellent augmentation. Additionally, employing a Cu-Al₂O₃-H₂O hybrid nanofluid, Chatterjee et al.¹⁸ discovered that heater-cooler positions have a major impact on heat transfer and entropy formation. The cross layout is the most efficient for increased heat transmission.

Al-Mudhaf et al.¹⁹ explored the Soret and Dufour effects on unsteady double-diffusive natural convection in trapezoidal enclosures filled with an isotropic porous material.

Inspired by the dynamism of convective flow problems in enclosures, Bhopalam et al.²⁰ employed the lattice Boltzmann simulations to explain the reliance of the flow field in both antiparallel and parallel wall motions. The activity of dominant parameters, such as Reynolds number and oscillation frequency, in influencing the intensity of fluid penetration inside the cavity and employing speed ratio to direct the course of corner vortices produced by the motion of the bottom lid was also considered. The work provided a magnificent pathway for achieving clearly defined flow modes using salient parameters to attain specific industrial needs.

Similarly, Nahak et al.²¹ used a circular cylinder firmly placed centrally in a triangular domain by varying the diameter vis-à-vis the cavity height and relating it to the aspect ratio and other relevant parameters. The findings from the investigation were demonstrated with the aid of the Artificial Neural Network. Kareem and Gao²² conducted a comparative numerical analysis in a differentially heated domain involving two self-induced rib-roughness and conventional smooth walls subjected to varying magnitudes of speeds and Reynolds numbers while ensuring the condition of other surfaces of the domain remained unchanged. It was determined that artificial roughness significantly impacted fluid and heat transfer properties. Finally, incorporating the introduced artificial ribs dramatically improved the heat transfer rate.

The outcomes demonstrate the effects of variables on flow patterns, temperature distribution, concentration distribution, and heat and mass transfer rates. These variables include the buoyancy ratio, Dufour number, and Soret number.

Olayemi et al.²³ analyzed the influence of internal heat generation/absorption in MHD flow in the space between the enclosure and the elliptical cylinder. It was indicated that internal heat generation promoted heat transfer, while heat absorption discouraged heat transfer enhancement. Olayemi et al.²⁴ reported the effects of Prandtl and Rayleigh numbers in a square domain subjected to different heating modes and driven by buoyancy forces. They reported the relationship between Nusselt and Rayleigh numbers, and that low Prandtl number fluids (like, air and water) are preferred in achieving a very high heat transfer rate than fluids of high Prandtl numbers, which are expensive but result in an insignificant increase in heat transfer enhancement when compared with fluids of low Prandtl number. Razera et al.²⁵ analyzed numerically mixed convective flow and heat transfer optimization following the constructive design of an elliptical fin embedded inside a lid-driven square enclosure. According to their findings, optimal configuration produced better-augmented transfer of heat than other geometries. The Rayleigh and Reynolds numbers played a substantial role in controlling and dictating the optimal shapes of the flow field.

Selimefendigil²⁶ explored carbon nanotube-water nanofluid in a lid-driven cavity, revealing that the average Nusselt number increases significantly with the addition of carbon nanotubes and an optimal obstacle size maximizes heat transfer. The study on MHD mixed convection boundary layer flow, which discusses velocity, temperature, and concentration

profiles, revealed that the effect of the magnetic field parameter reduces the velocity profile.²⁷ Al-Farhany et al.²⁸ studied MHD heat transfer in a ferrofluid-filled inclined enclosure; it was shown that the average Nusselt number increased with modified Rayleigh number, Darcy number, and fins length, while it decreased with Hartmann number. Adegun and Bello-Ochende²⁹ analyzed heat transfer in an inclined rotating rectangular duct. It was found that radiation and rotation enhanced heat transfer. A stabilizing influence of Pr over the flow characteristics was studied by Iwatsu et al.³⁰ When Pr was low, the temperature field in the interior tends to a vertically linear profile throughout the entire cavity. An investigation of the mixed convective heat transfer in shallow rectangular lid-driven cavities showed that the average Nusselt number increased with cavity inclination.³¹ Cheng³² examined the flow and heat transfer in a square cavity; it was revealed that the average Nusselt numbers increased with Grashof and Reynolds numbers.

Alsabery et al.³³ reported the impacts of a solid square, which was situated in a square enclosure, and nanofluid on heat transfer enhancement under mixed convective heat transfer mode. It was concluded that heat transfer augmentation was realized for large sizes of the solid body at high values of Richardson and Reynolds numbers and that nanoparticle volume fraction aids heat transfer only at high Reynolds numbers. Izadi et al.³⁴ reviewed heat transfer in various configurations using various nanofluids. It was concluded that an increase in nanofluid volume fraction, Reynolds number, and Richardson number favored heat transfer improvement. Chamkha et al.³⁵ investigated mixed convection in a vented square cavity containing a square cylinder; it was submitted that the position of the inner square and its size play a dominant role in heat transfer augmentation. Tayebi et al.³⁶ considered the influence of entropy generation, internal heat generation, and nanofluid volume fraction on heat transfer and flow profiles in an elliptical domain. It was noted that these parameters significantly impacted the enhancement of heat transport. The merits of nanofluid in powering ships and solar aircraft were reported by Obalalu et al.^{37,38}

Earlier works on configurations having obstructions within them inspire the current analysis. The present study examines the effect of the Grashof number, Reynolds number, ellipse axis ratio, and orientation angle on the profiles of combined forced-free convection flow in a square domain with its top horizontal wall moving with a fixed velocity in the positive x -direction. The vertical boundaries are subjected to cooling conditions, while the obstruction is heated sinusoidally. Findings from this investigation have varied applications, some of which include the electronic industry, nuclear technology for cooling purposes, and the food industry for packaging. Furthermore, in the academic sphere, results from this finding could serve as a guide for the experimentalist.

2 | PHYSICAL MODEL

Figure 1A depicts the physical model being considered, while Figure 1B represents the mesh profile of the current study. The geometry consists of a sinusoidally heated elliptic cylinder of temperature $T_h + \Delta T \sin\left(\frac{\pi x}{L}\right)$, which is placed centrally in a square enclosure. The enclosure is filled with a fluid having Pr equal to 0.71. The horizontal walls are thermally insulated, while the vertical walls are subjected to a cooling condition (T_c).

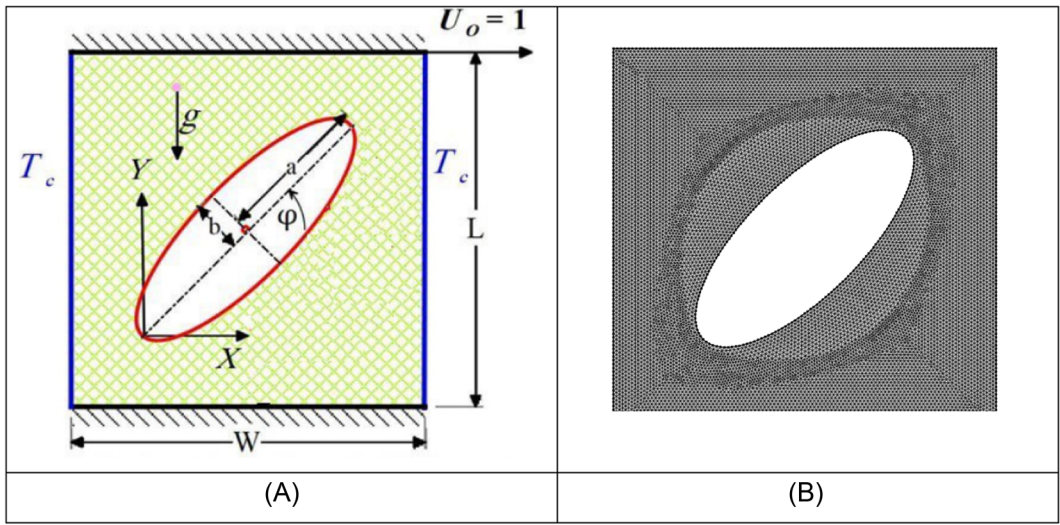


FIGURE 1 (A) Physical geometry and (B) mesh profile of the geometry. [Color figure can be viewed at wileyonlinelibrary.com]

3 | GOVERNING EQUATION

The model's equations are continuity, momentum, and energy transport equations. The following assumptions prevail: (i) the fluid in the domain obeys the Newtonian law; (ii) the flow in the domain is laminar with negligible variation in the density except for the buoyancy term; (iii) the flow and thermal characteristics are time-independent; (iv) the top horizontal wall moves to the right side at a constant velocity, $U_o = +1$, while the other walls of the configuration are stationary; (v) the density variation in the buoyancy component is computed by applying the Boussinesq approximation.

The continuity, momentum, and energy equations in the dimensionless form are given by Equations (1)–(4)³¹:

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0. \quad (1)$$

The dimensionless X and Y momentum equations are reckoned by Equations (2) and (3):

$$U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \frac{1}{Re} \left[\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} \right], \quad (2)$$

$$U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \frac{1}{Re} \left[\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} \right] + \frac{Gr}{Re^2} \Theta. \quad (3)$$

The dimensionless energy transport equation is as in Equation (4):

$$U \frac{\partial \Theta}{\partial X} + V \frac{\partial \Theta}{\partial Y} = \frac{1}{Re \cdot Pr} \left[\frac{\partial^2 \Theta}{\partial X^2} + \frac{\partial^2 \Theta}{\partial Y^2} \right]. \quad (4)$$

3.1 | Normalization parameters

Equation (5) was used to convert the governing equations to their equivalent dimensionless forms²⁶:

$$X = \frac{x}{L}, \ Y = \frac{y}{L}, \ U = \frac{u}{U_0}, \ V = \frac{v}{U_0}, \ \Theta = \frac{T - T_c}{T_h - T_c}, \ P = \frac{p}{\rho U_0^2}, \ Pr = \frac{\nu}{\alpha}.$$
 (5)

3.2 | Nusselt number evaluation

The local and mean Nusselt numbers on the boundaries of the configuration of interest are resolved, respectively, by Equations (6) and (7)³¹:

$$Nu = -\frac{\partial \Theta}{\partial X} \Big|_{x=L},$$
 (6)

$$\overline{Nu} = \frac{1}{L} \int_0^L \frac{\partial \Theta}{\partial X} dY,$$
 (7)

where L is the length/perimeter of the wall of interest.

4 | NUMERICAL SOLUTION

The boundary value problem in the current investigation, which is represented by Equations (1)–(4), is reckoned subject to the prevailing boundary conditions by FEM, as detailed by Madhu and Kishan.²⁷ Stability was assured by employing the P2–P1 Lagrange elements and the Galerkin least-square model. Grid tests were conducted as enunciated in Table 1. Table 1 shows that the normal grid size has the least percentage error. Still, as a compromise between convergence time and enhanced field resolution, an extremely fine grid size was chosen and used in all performed simulations.

TABLE 1 Mesh sensitivity analysis of $\overline{Nu}$ on the cylinder wall for $Gr = 10^6$, $Re = 100$, $AR = 3$, and $\phi = 90^\circ$.

Grid size	Number of elements	$\overline{Nu}$	% Error
Coarse	356	33.86	–
Normal	734	34.38	1.54
Fine	1048	35.36	2.85
Finer	1990	38.37	8.51
Extra fine	6482	40.34	5.13
Extremely fine	25,196	41.21	2.16

TABLE 2 $\overline{Nu}$ of the hot top wall in comparison with $\overline{Nu}$ from previous studies.^{30–32}

<i>Re</i>	Authors	<i>Gr</i>		
		10^2	10^4	10^6
400	Iwatsu et al. ³⁰	3.84	3.62	1.22
	Sharif ³¹	4.05	3.82	1.17
	Cheng ³²	4.14	3.90	1.21
	Present work	3.99	3.76	1.15
1000	Iwatsu et al. ³⁰	6.33	6.29	1.77
	Sharif ³¹	6.55	6.50	1.81
	Cheng ³²	6.73	6.68	1.75
	Present work	6.21	6.16	1.66

4.1 | Validation of results

Independence examinations of various mesh sizes on the $\overline{Nu}$ of the hot elliptical cylinder are presented in Table 1. The extremely fine grid size was chosen for enhanced field resolution over the normal grid size. Also, the reliability of the numerical outcomes acquired from the current investigation was found by juxtaposing the results with the $\overline{Nu}$ achieved by analyzing a square enclosure, which experiences different thermal conditions at its boundary in the absence of the heated obstruction with the works of Iwatsu et al.,³⁰ Sharif,³¹ and Cheng.³² Table 2 shows that the results from the current study and those of Iwatsu et al.,³⁰ Sharif,³¹ and Cheng³² are a perfect fit.

5 | STREAM FUNCTION

The motion of the fluid in the regions between the enclosure and the cylinder is enunciated with the stream function (Ψ)-velocity relation given by Biswas et al.,⁸ Al-Farhany et al.,²⁸ and Adegun and Bello-Ochende²⁹:

$$U = \frac{\partial \Psi}{\partial Y}, \quad V = -\frac{\partial \Psi}{\partial X}. \quad (8)$$

And Ψ is reckoned as follows (9):

$$\frac{\partial^2 \Psi}{\partial X^2} + \frac{\partial^2 \Psi}{\partial Y^2} = \frac{\partial U}{\partial Y} - \frac{\partial V}{\partial X}. \quad (9)$$

6 | RESULTS AND DISCUSSION

This section discusses the impacts of the variation of AR , Re , Gr , and cylinder orientation (ϕ) on lid-driven mixed convective heat transfer and fluid flow around the annulus between a square enclosure and an elliptical cylinder, which is situated concentrically inside the square enclosure.

The impacts of dimensionless parameters have been investigated in this study, including:

1. Grashof number ($10^3 \leq Gr \leq 10^6$),
2. Reynolds number ($1.0 \leq Re \leq 100$),
3. cylinder tilt angle ($0^\circ \leq \phi \leq 90^\circ$),
4. aspect ratio ($1.0 \leq AR \leq 3.0$).

6.1 | Impacts of Re on isotherm and stream function

Figure 2 displays the charts of isotherms and stream functions for various varying values of Grashof and Reynolds numbers while maintaining $\phi = 0^\circ$ and $AR = 3$. Figure 2 shows that as the Grashof number increases, the gradients of the isothermal contours become steeper, which is due to increased heating, and the isothermal lines begin to deviate from being parallel to the vertical walls; the formation of plumes above the elliptical cylinder becomes more vigorous. This indicates a change in the mode of heat transfer from the conduction- to the convection-dominated regime. Moreover, the contours surrounding the bottom wall of the cylinder and along the enclosure's right and left side walls became more densely packed; this shows that the intensity of convection rises with the increase of the Grashof number (Gr). The isothermal contour line distribution when Re was increased from 1.0 to 10 is comparable; however, at $Re = 100$, and specifically for Grashof numbers of $10^3 \leq Gr \leq 10^5$, the isothermal lines above the elliptical cylinder became biased toward the left wall of the enclosure, which could have been due to weakness in convection strength of the fluid, but at $Gr = 10^6$, the isothermal lines above the cylinder re-stabilized, possibly due to much more improvement in the buoyancy force compared with the shear force. Figure 3 displays the stream function plots. For the stream functions, at $Re = 1$, four vortices are formed in the cavity. As the Grashof number increases, the strength of the circulation improves with those to the right of the cylinder rotating clockwise, while those to the left are rotating anticlockwise; this rotation pattern is mainly due to the direction of lid-motion. Furthermore, the strength of the circulations above the cylinder is much more vigorous than those below. However, as the Re increases in value, the strength of the circulation diminishes, which is due to an increase in the strength of the conduction mode of heat transfer. At the lowest Gr value and Re values of 10 and 100, a single vortex was formed above the cylinder, which is a result of improvement in the strength of the conduction mode of heat transfer and resulted in the weakening of the impact of convection; this observation also played out when $Gr = 10^4$ and $Re = 100$.

6.2 | Influence of AR on isotherm and stream function

The isothermal contours and stream functions for the range of Gr and AR values considered when $\phi = 0^\circ$ are presented in Figure 4. As the value of Gr improves, the influence of conduction on heat transfer weakens. From $Gr = 10^5$, the impact of convection becomes established with the formation of plumes above the elliptical cylinder. Also, the activities of plumes spread closer and closer to the cold vertical walls, which indicates a higher value of temperature change, and hence improvement in the transfer of heat. For the stream functions in Figure 5, as the Gr values increase, similar to Figure 2B, the convective strength predominates over the conductive strength. This is evident in the increasing strength of the circulations as Gr

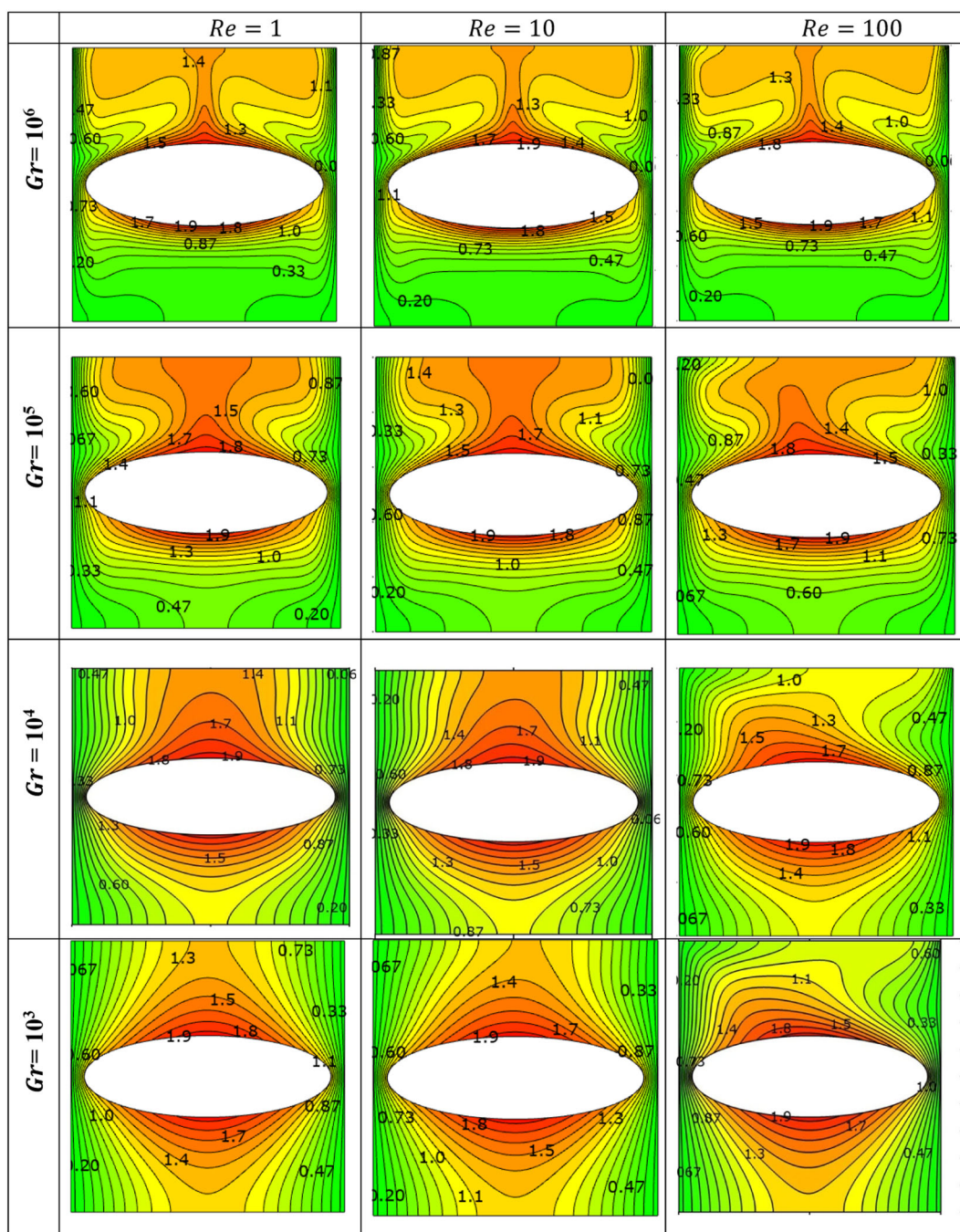


FIGURE 2 Effect of Gr and Re on isothermal contours when $\phi = 0^\circ$ and $AR = 3.0$. [Color figure can be viewed at wileyonlinelibrary.com]

improves. As the AR value increases, the circulation strength reduces, and the formation of four vortices within the flow domain becomes increasingly established. This is mainly due to the phenomenon of flow separation that led to an increase in the number of vortices as the clearance between the ellipse and side walls reduces, friction along half vertical wall retards the

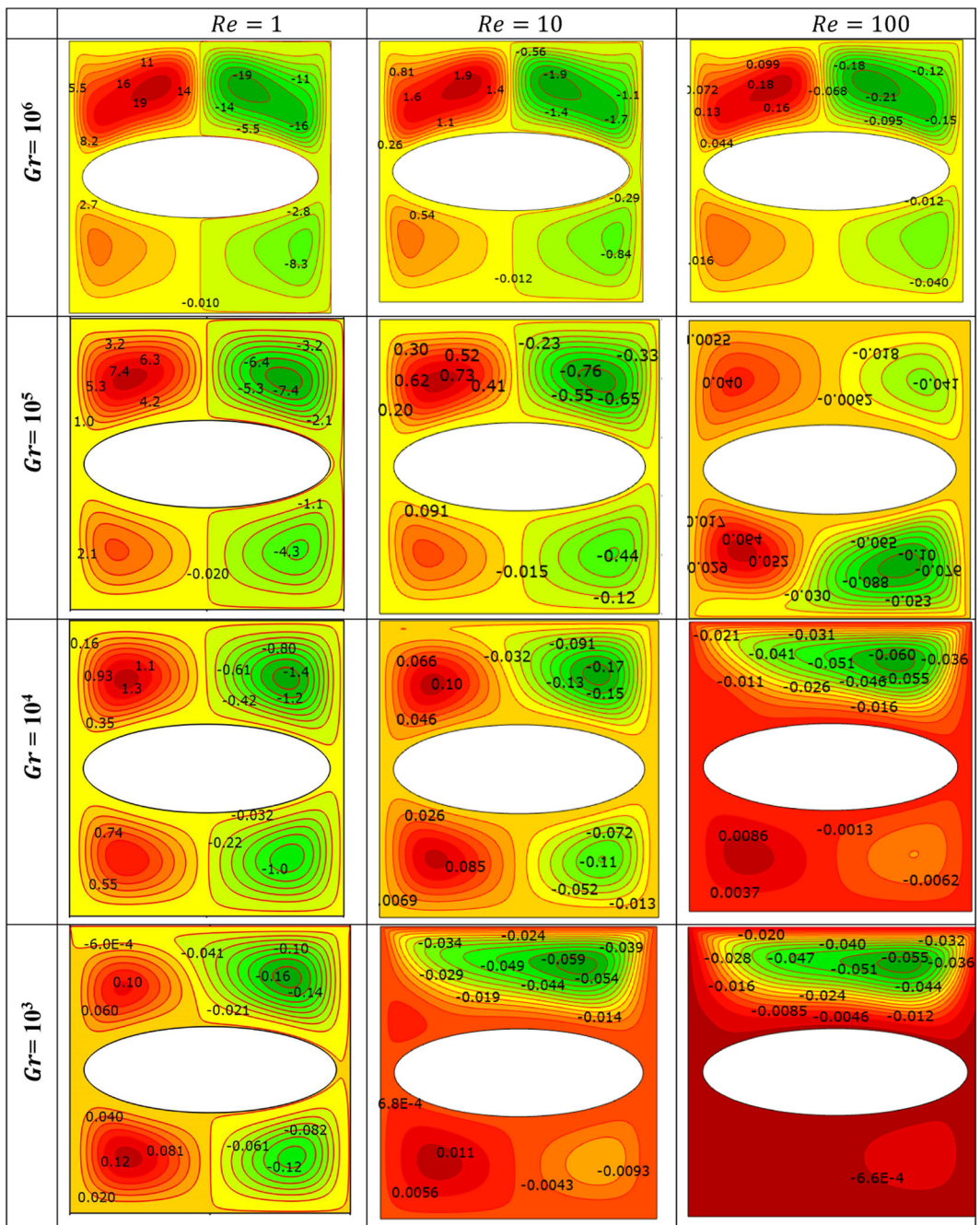


FIGURE 3 Effect of Gr and Re on stream functions when $\phi = 0^\circ$ and $AR = 3.0$. [Color figure can be viewed at wileyonlinelibrary.com]

flow and gradually separated the flow domain; this restriction of flow at $AR = 3.0$ created the four vortices. Also, as AR increases, more heat is being dissipated into the fluid domain because of the increase in the total heat transfer surface area; that was why an increase in temperature as the aspect ratio increases were experienced.

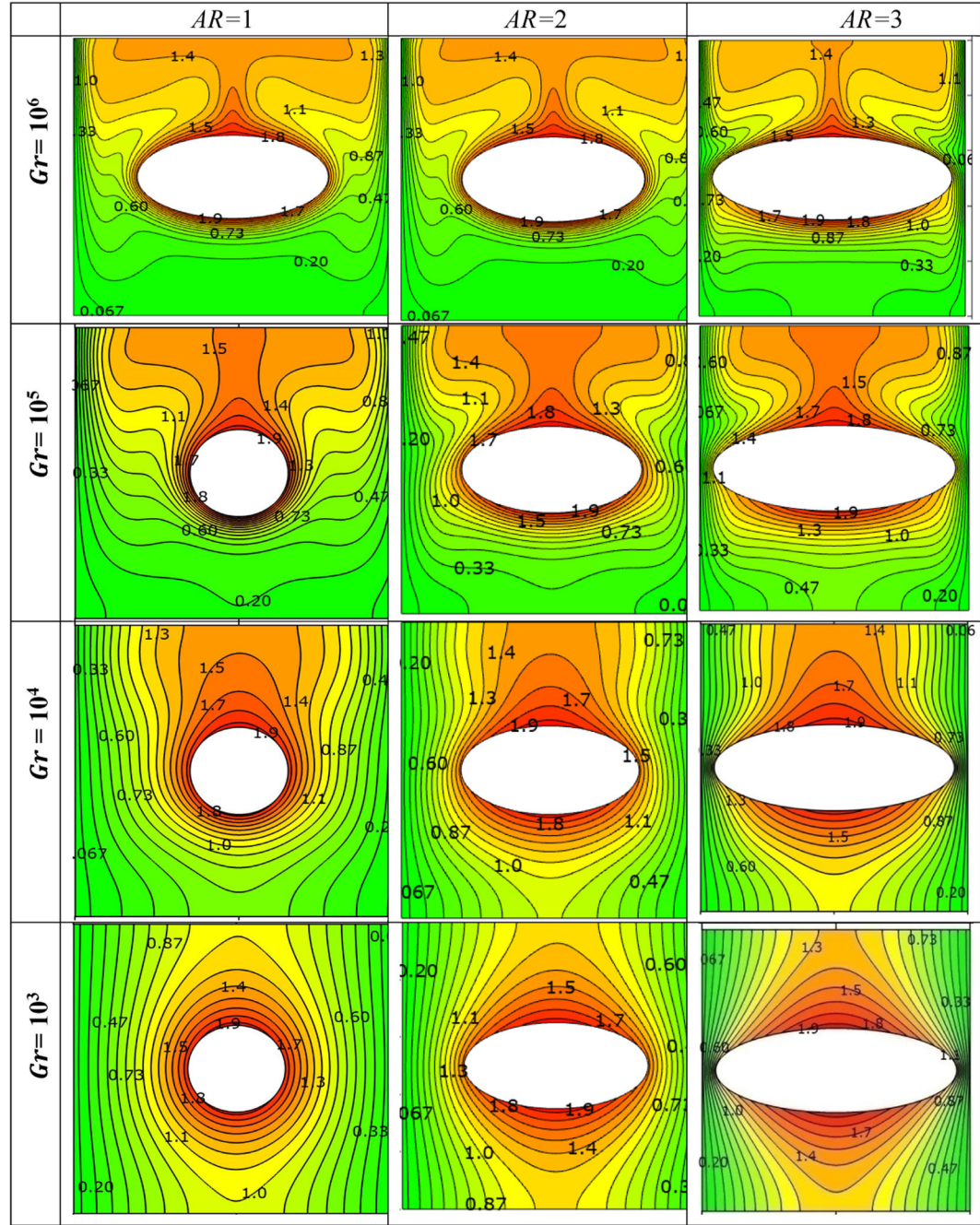


FIGURE 4 Isothermal contour plots for various values of Gr and AR when $\phi = 0^\circ$ and $Re = 1$. [Color figure can be viewed at wileyonlinelibrary.com]

6.3 | Influence of ϕ on stream function and isotherm

Figures 6 and 7 present the isothermal contours and stream function plots for different tilt angles of the cylinder when $AR = 3.0$ and $Re = 1.0$. With the increasing tilt angle of the

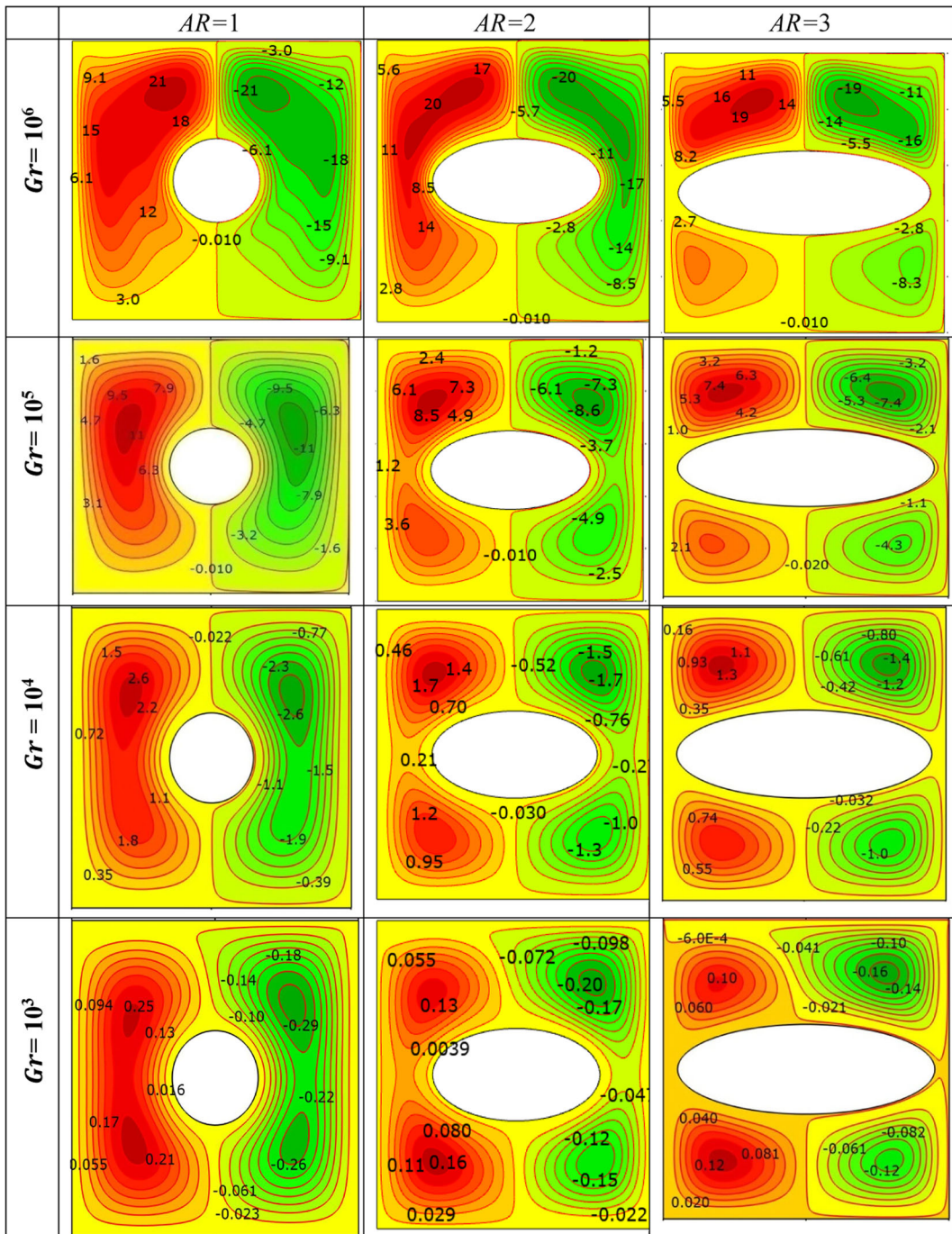


FIGURE 5 Influence of AR and Gr on stream function plots for $\phi = 0^\circ$ and $Re = 1.0$. [Color figure can be viewed at wileyonlinelibrary.com]

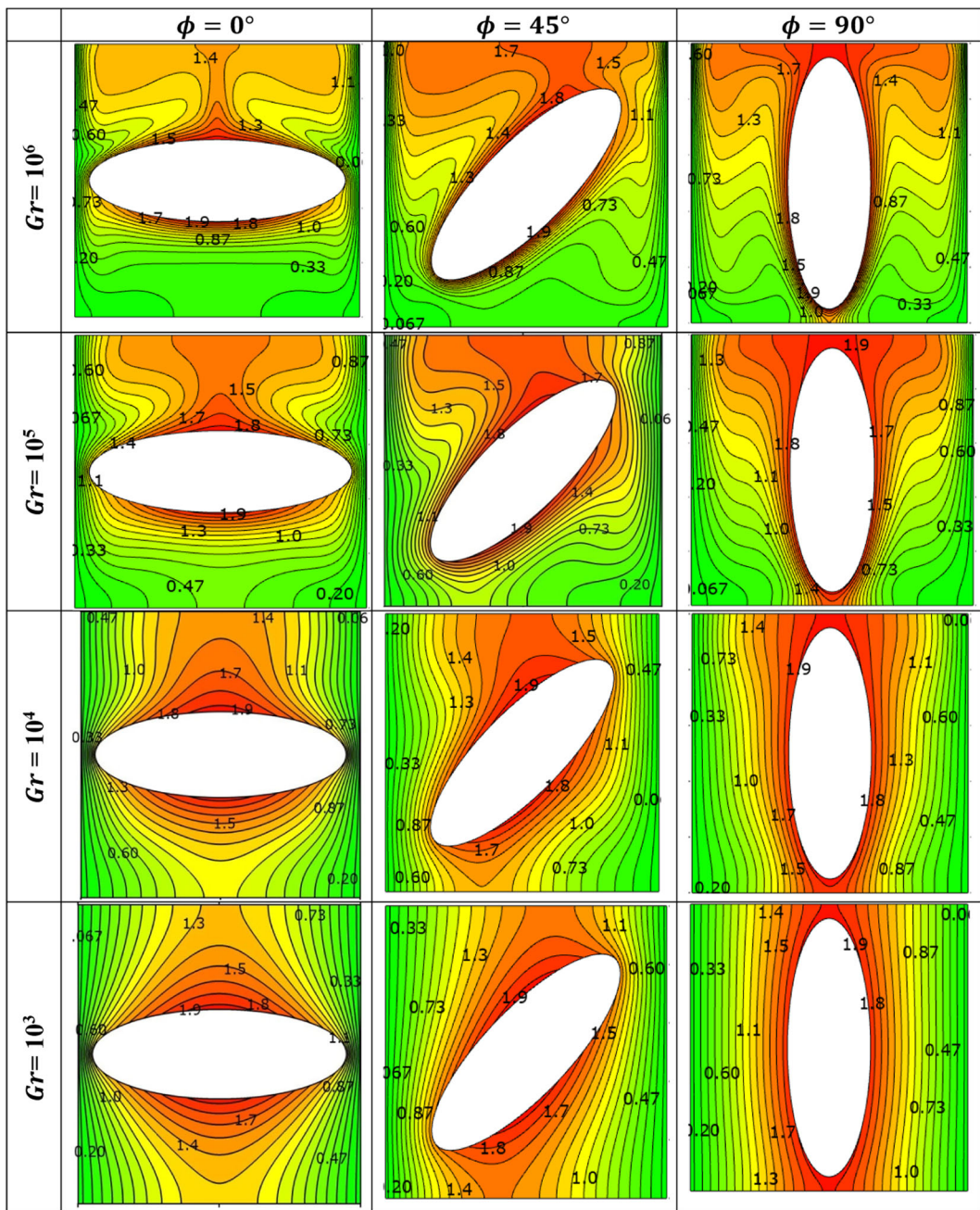


FIGURE 6 Effects of the inclination angle (ϕ) on isotherm plots for $Gr = 10^3$ – 10^5 , $Re = 1$, and $AR = 3.0$. [Color figure can be viewed at wileyonlinelibrary.com]

cylinder, more heat is transferred from the upper surface of the ellipse into the fluid. In addition, the response of the isotherms to improvement in Gr is consistent with the previous patterns shown in Figures 2 and 4. As per the stream function plots in Figure 7, for an orientation angle of $\phi = 0^\circ$, four vortices are present for all the Gr values

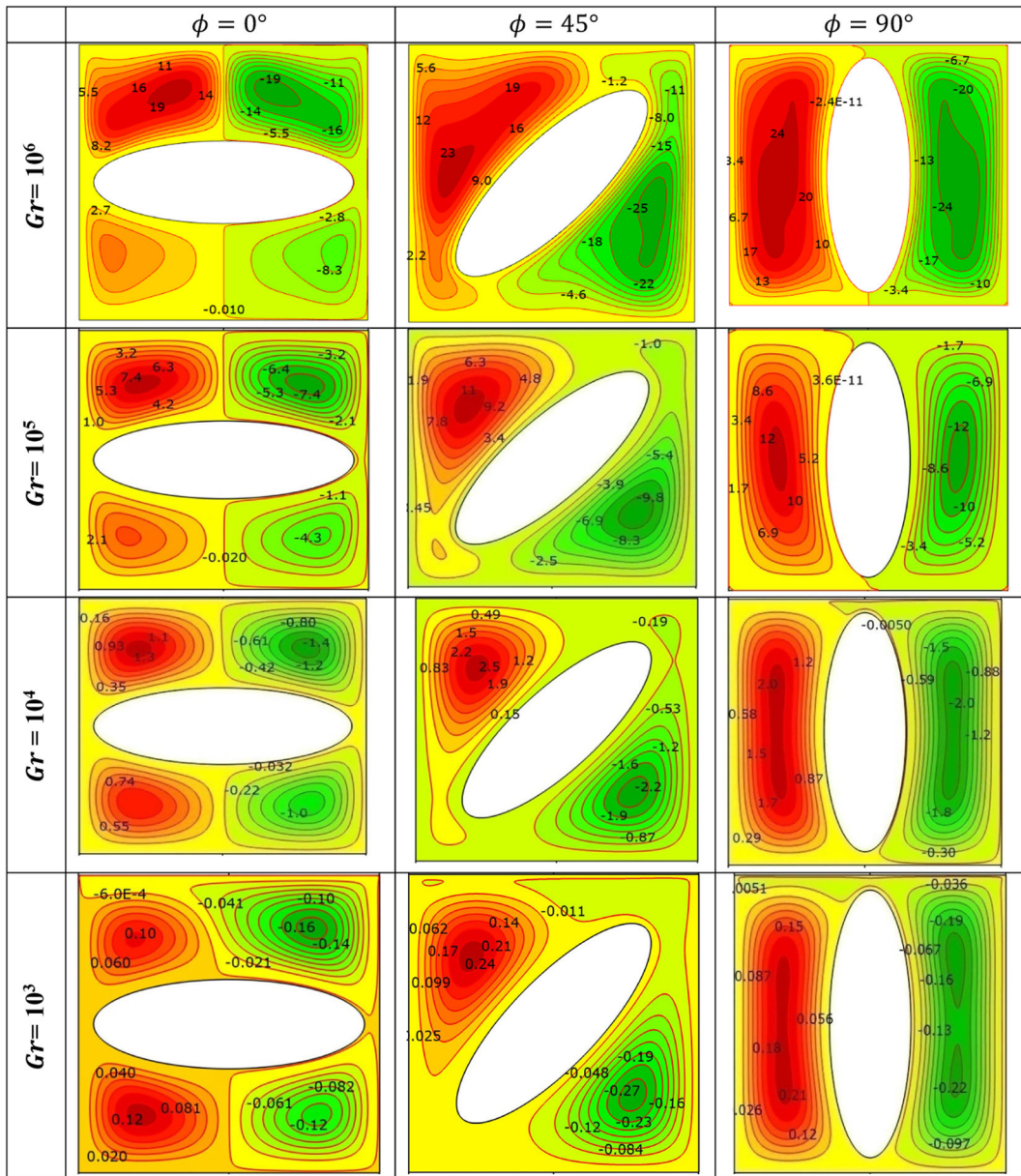


FIGURE 7 Impact of the inclination angle (ϕ) and Grashof number on stream function for $Re = 1.0$ and $AR = 3.0$. [Color figure can be viewed at wileyonlinelibrary.com]

investigated, with those to the right of the cylinder rotating clockwise, while those to the left in an anticlockwise fashion. As the orientation angle of the cylinder increases, the circulation strength increases, and the vortices on each side of the cylinder merge with $\phi = 90^\circ$ yielding circulation with the highest strength.

6.4 | Effects of AR and ϕ on $\overline{Nu}$ of the elliptical cylinder for various Gr values

The plots of $\overline{Nu}$ along the cylinder wall versus inclination angle are displayed in Figures 8–11 for varying AR and for Gr of $10^3 \leq Gr \leq 10^6$. For all the cylinder inclination angles considered, the values of $\overline{Nu}$ when $AR = 1.0$ do not respond to changes in the inclination angle, which is

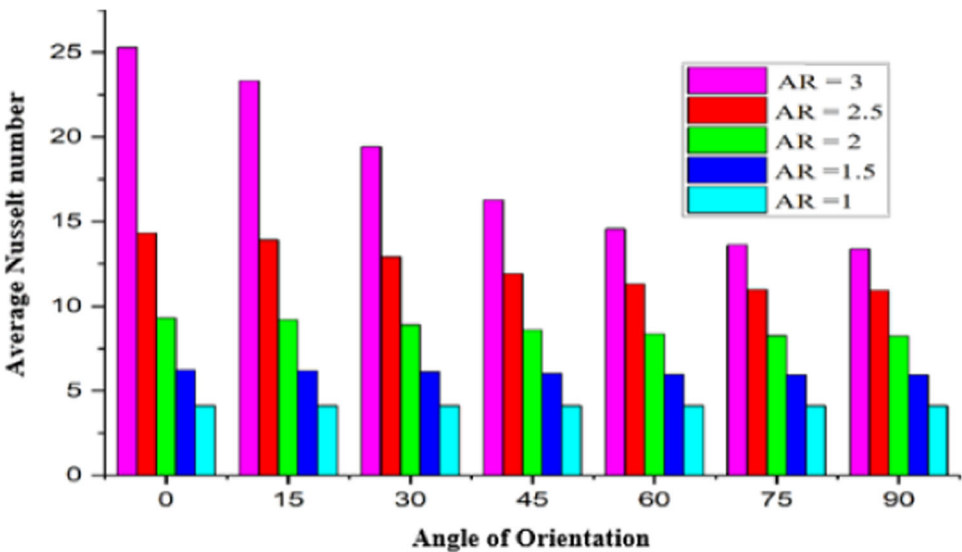


FIGURE 8 Plot of $\overline{Nu}$ of cylinder versus ϕ for $Gr = 10^3$ and $Re = 100$. [Color figure can be viewed at [wileyonlinelibrary.com](#)]

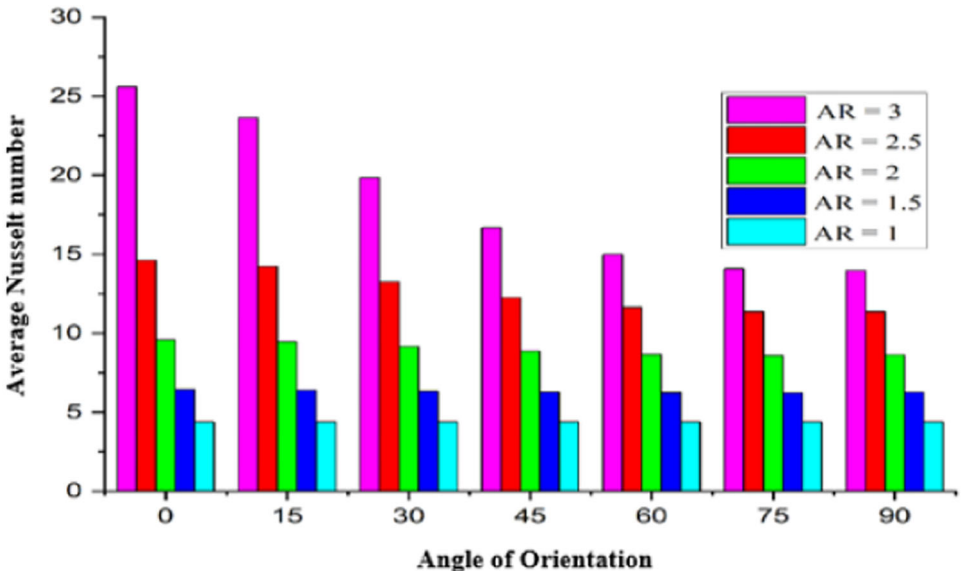


FIGURE 9 Plot of $\overline{Nu}$ versus ϕ for $Gr = 10^4$ and $Re = 100$. [Color figure can be viewed at [wileyonlinelibrary.com](#)]

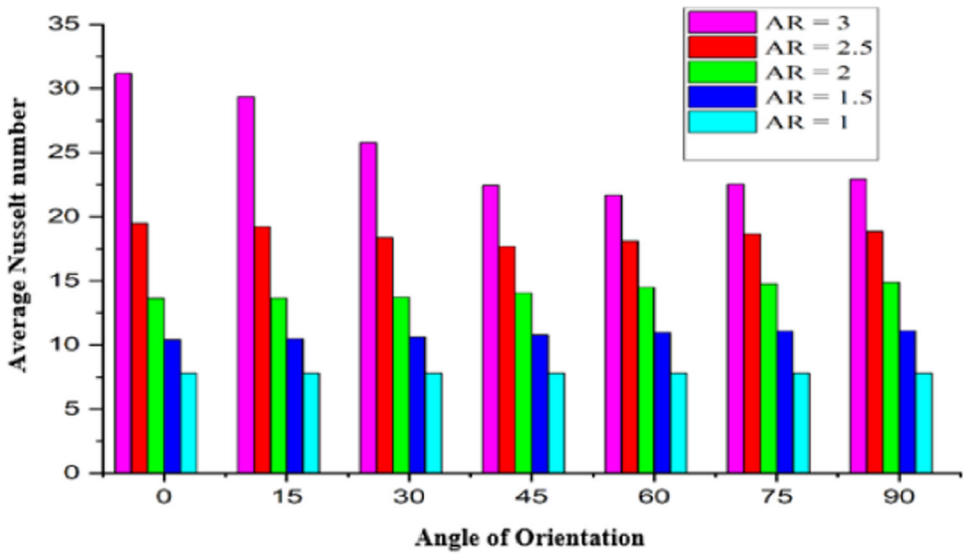


FIGURE 10 Plot of $\overline{Nu}$ versus ϕ for $Gr = 10^5$ and $Re = 100$. [Color figure can be viewed at wileyonlinelibrary.com]

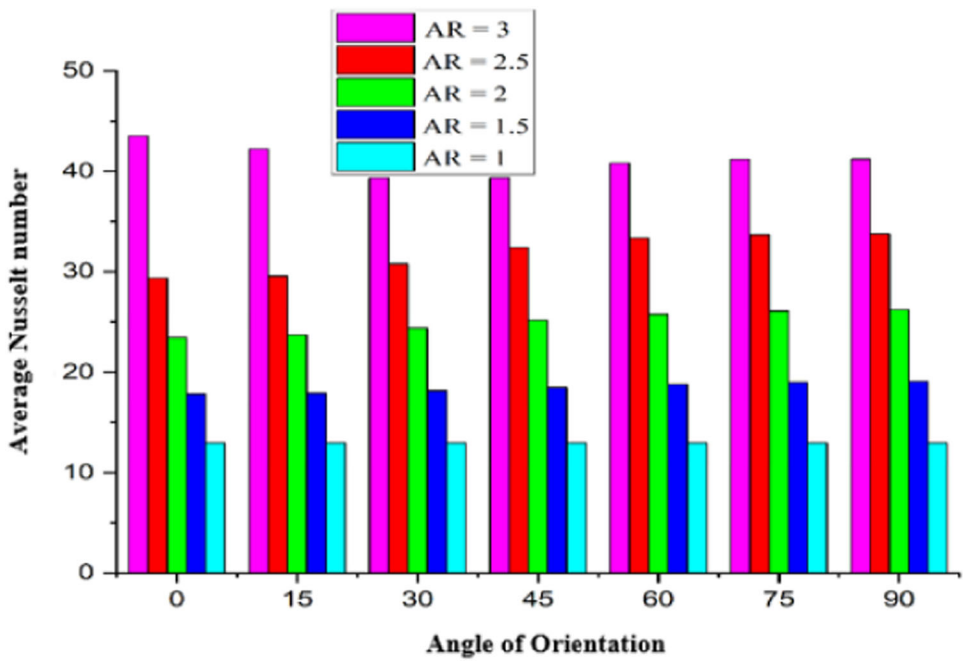


FIGURE 11 Plot of $\overline{Nu}$ versus ϕ for $Gr = 10^6$ and $Re = 100$. [Color figure can be viewed at wileyonlinelibrary.com]

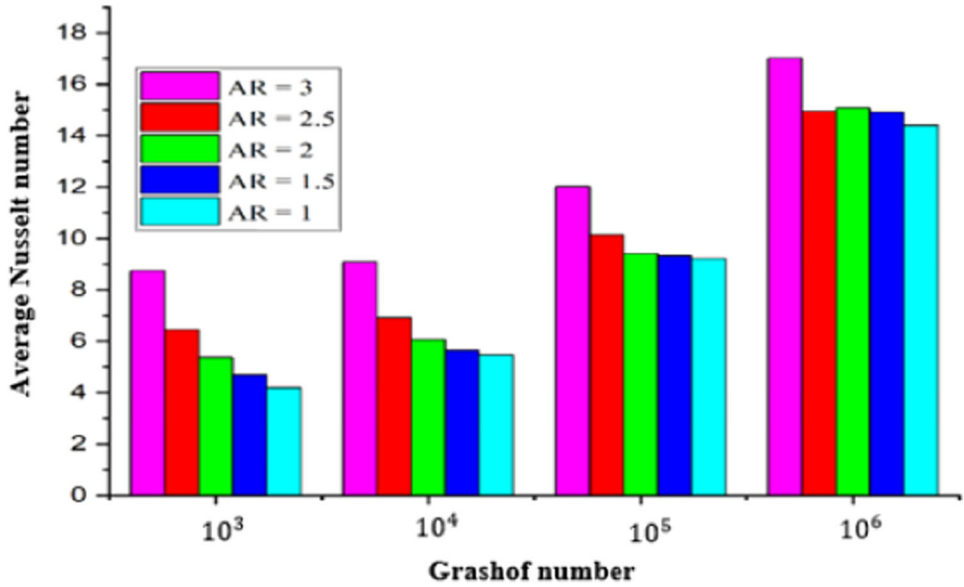


FIGURE 12 Graph of $\overline{Nu}$ on the right wall against Gr for various AR when $\phi = 0^\circ$ and $Re = 100$. [Color figure can be viewed at wileyonlinelibrary.com]

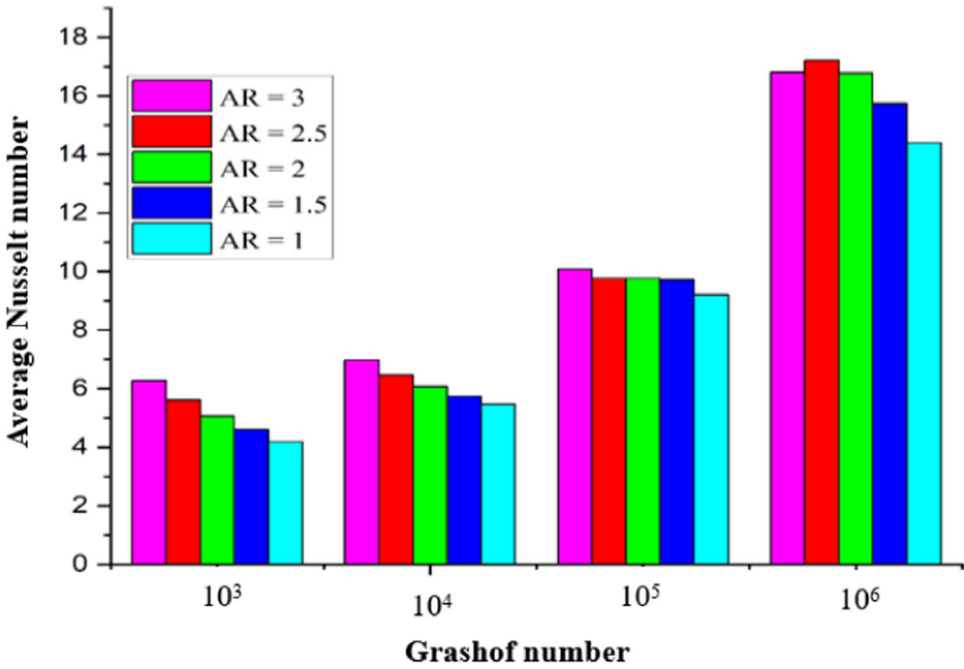


FIGURE 13 Graph of $\overline{Nu}$ on the right wall against Gr for varying AR values when $\phi = 45^\circ$ and $Re = 100$. [Color figure can be viewed at wileyonlinelibrary.com]

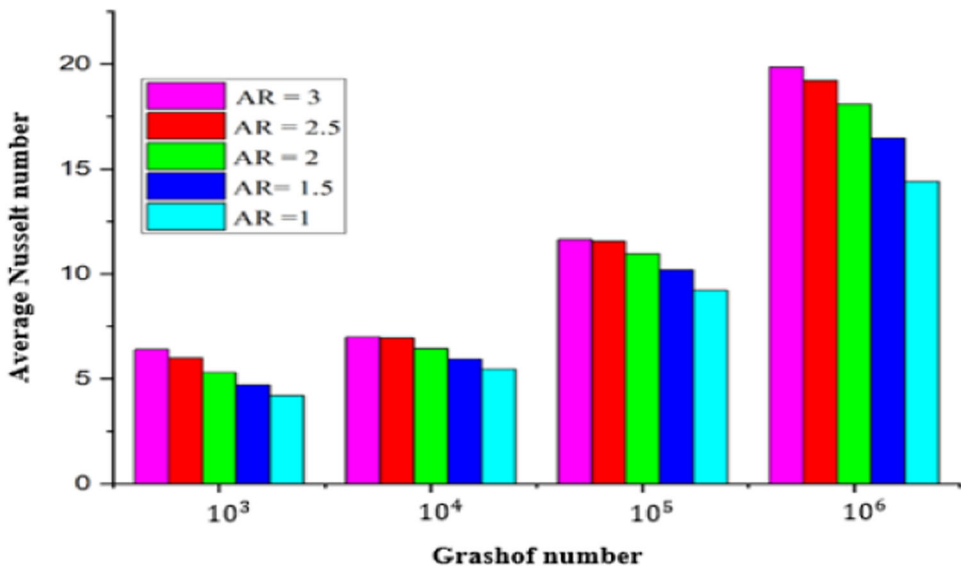


FIGURE 14 $\overline{Nu}$ on the right wall against Gr for distinct AR values when $\phi = 90^\circ$ and $Re = 100$. [Color figure can be viewed at wileyonlinelibrary.com]

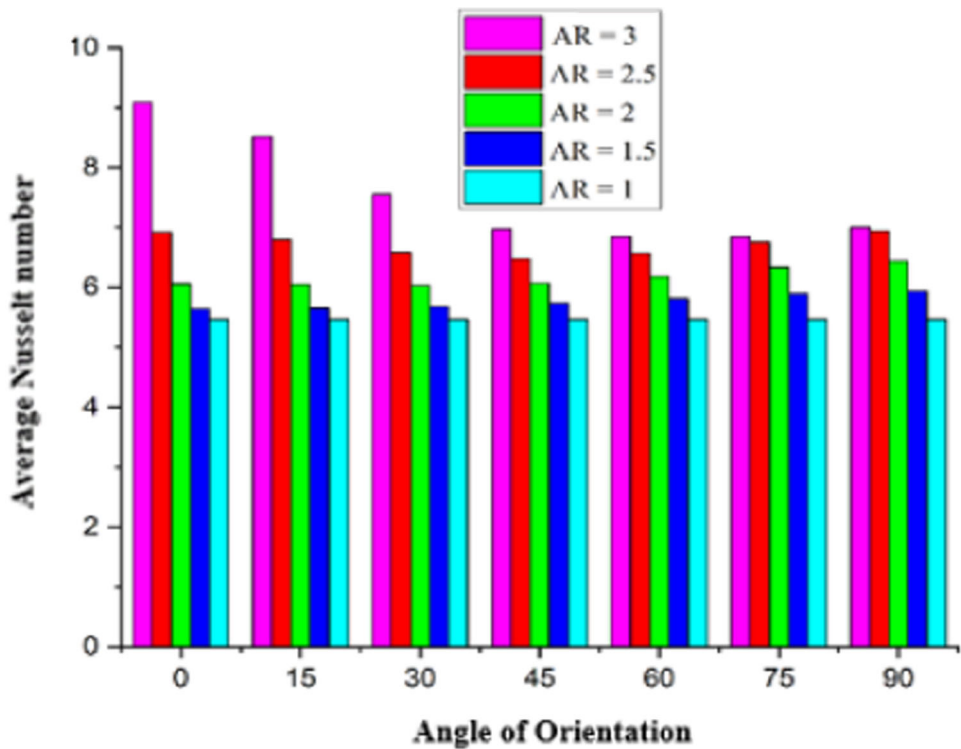


FIGURE 15 Graph of $\overline{Nu}$ on the right wall against ϕ for different AR values when $Re = 100$ and $Gr = 10^3$. [Color figure can be viewed at wileyonlinelibrary.com]

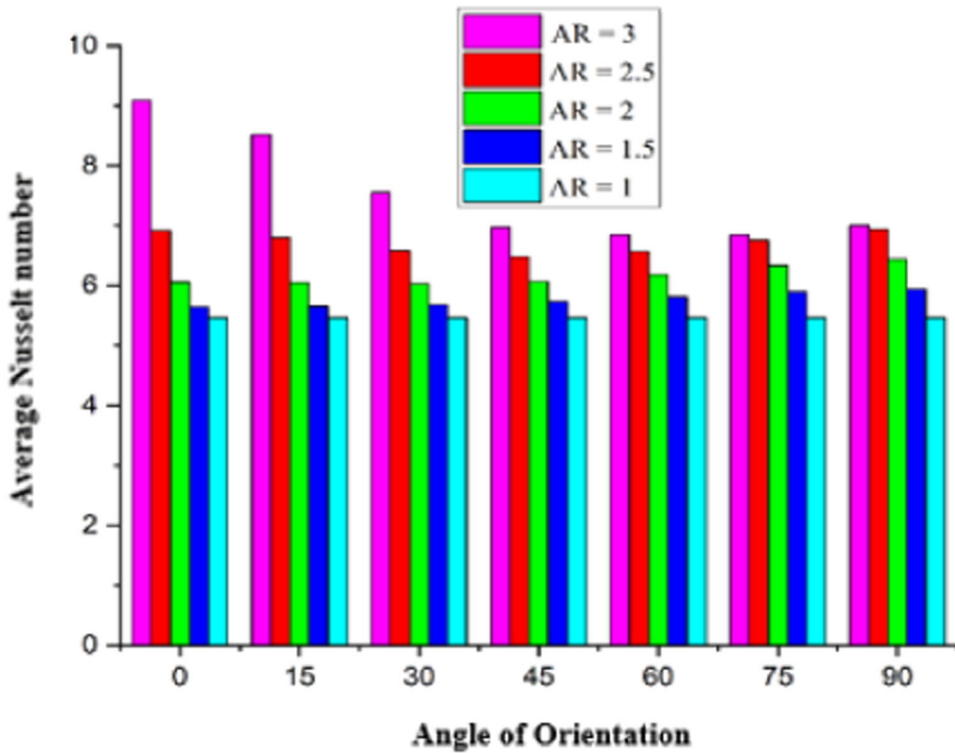


FIGURE 16 Graph of $\overline{Nu}$ on the right wall ϕ for different AR values $Re = 100$ and $Gr = 10^4$. [Color figure can be viewed at wileyonlinelibrary.com]

due to its circular nature. Also, the Gr increase resulted in heat transfer augmentation. For $Gr = 10^3$, the increase in cylinder inclination angles as presented in Figure 8 shows a reduction in $\overline{Nu}$ with inclination angle increment, particularly for $2.5 \leq AR \leq 3.0$, and this trend diminishes as the aspect ratio decreases, while for AR values of $1.0 \leq AR \leq 2.5$, inclination angle increment has a very marginal impact on the value of $\overline{Nu}$ of the cylinder wall. The trends observed in Figure 8 are featured in Figures 9–11.

6.5 | Effects of AR and Gr on Nu of the enclosure right wall for different ϕ values

Figures 12–14 present the plots of the $\overline{Nu}$ on the right wall of the enclosure versus the Gr for different cylinder sizes for $\phi = 0^\circ, 45^\circ, 90^\circ$, and $Re = 100$. $\overline{Nu}$ gets enhanced for all the angles investigated by increasing the Gr . Additionally, the inclination angle variation of the cylinder has no effect on the augmentation of heat transfer on the right enclosure wall when $AR = 1.0$, which is due to its circular shape. When the inclination angle was increased from $\phi = 0^\circ$ to 45° , $\overline{Nu}$ decreased for low heating ($Gr = 10^3$). However, for higher heating ($Gr = 10^4$), the pattern was sustained except for low aspect ratios (2.0 and 1.5) where $\overline{Nu}$ remained almost constant. The same trend as $Gr = 10^4$ is sustained for AR of 3.0 and 2.5 when Gr was increased to 10^5 . However, for $Gr = 10^6$, heat transfer is augmented except for $AR = 3.0$. When ϕ was increased from $\phi = 45^\circ$ to 90° , $\overline{Nu}$ improved, and the improvement became conspicuous as the Grashof number increased. Comparing the heat transfer on the right wall

when the cylinder is in a horizontal position to that of the vertical position, the heat transfer on the right wall for the vertical cylinder position is more than that when the cylinder is in the horizontal position except for a very low Gr value of 10^3 and high-to-medium aspect ratios ($AR = 3.0, 2.5$, and 2.0). At the same time, both positions transferred heat at the same rate for low aspect ratios (1.5 and 1.0).

6.6 | Effects of AR and ϕ on $\overline{Nu}$ of the enclosure right wall for various Gr values

Figures 15–18 display the plots of $\overline{Nu}$ on the right wall against the orientation angle of the elliptical cylinder for the Grashof number in the range of $10^3 \leq Gr \leq 10^6$ when $Re = 100$. The plots show that for a particular Gr value, the average Nusselt number when $AR = 1.0$ is independent of the orientation angle (ϕ). For $10^3 \leq Gr \leq 10^4$, the $\overline{Nu}$ of the right wall increases with increasing AR values, but the impact of AR (particularly for $AR = 2.5$ and 3.0) on the right wall $\overline{Nu}$ reduces as the cylinder orientation angle increases. For $Gr = 10^5$ and $0^\circ \leq \phi \leq 45^\circ$ when $Re = 100$, the impact of AR on heat transfer weakens. When 60° , the exact amount of heat was transferred for $AR = 3.0$ and 2.5 ; for $60^\circ < \phi \leq 90^\circ$, AR improvement resulted in heat transfer enhancement. For $Gr = 10^6$ and $0^\circ \leq \phi \leq 30^\circ$, heat transfer on the right wall improves with AR increment, except that the heat transferred when $AR = 2.0$ is higher than that transferred when $AR = 2.5$. For $45^\circ \leq \phi \leq 90^\circ$, the

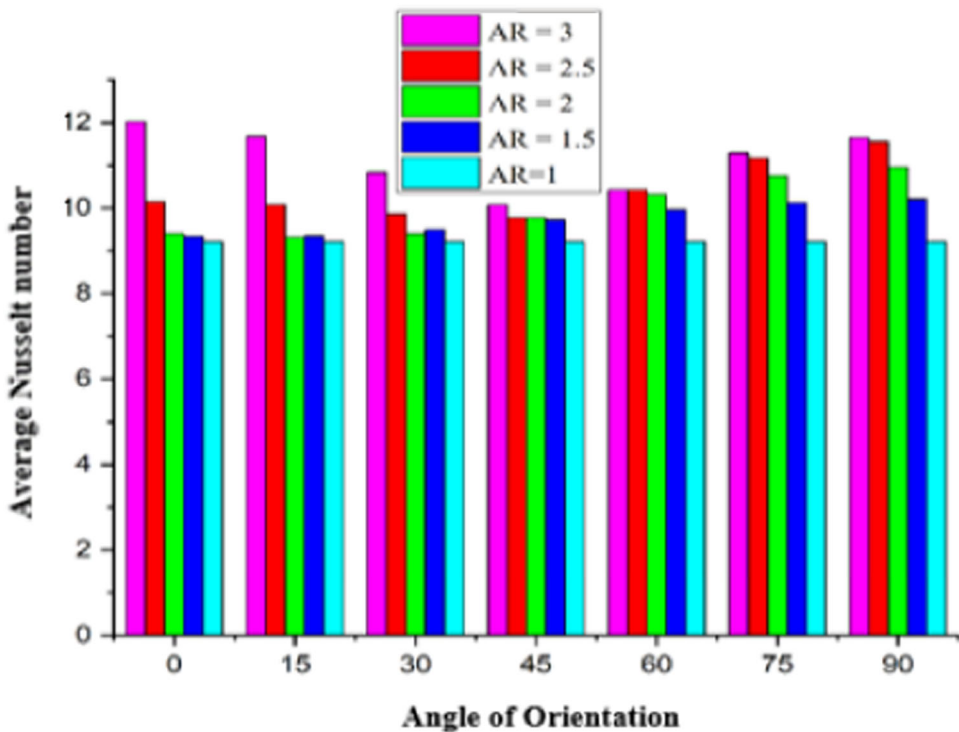


FIGURE 17 Plot of $\overline{Nu}$ on the right wall against ϕ for different AR values when $Re = 100$ and $Gr = 10^5$. [Color figure can be viewed at wileyonlinelibrary.com]

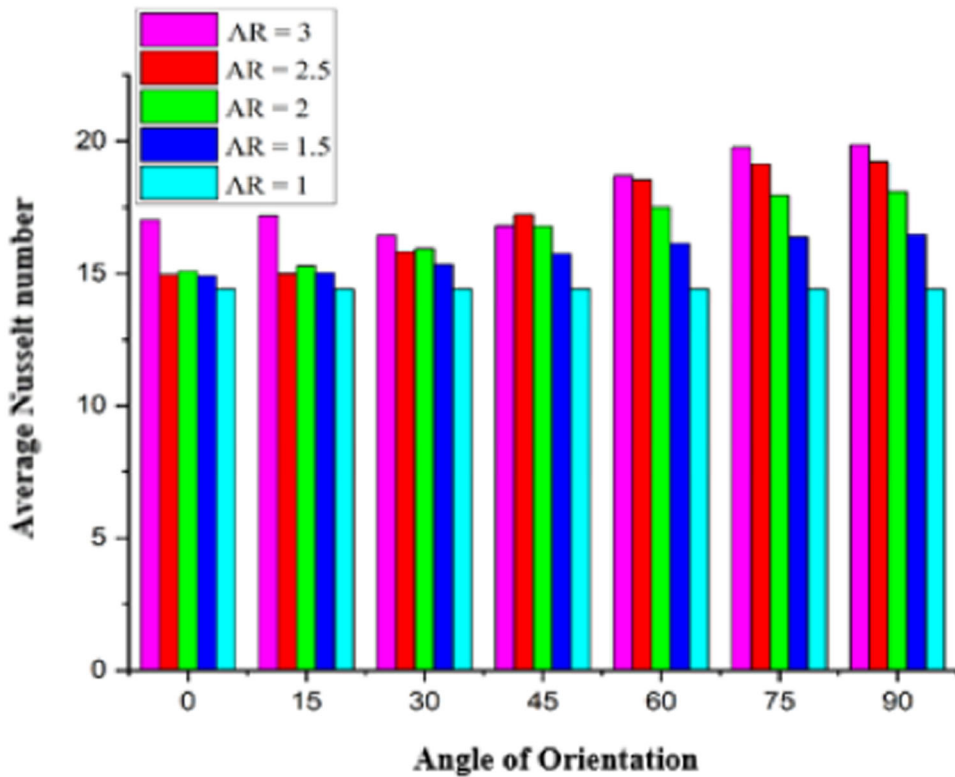


FIGURE 18 Plot of $\overline{Nu}$ on the right wall against wall ϕ for different AR values when $Re = 100$ and $Gr = 10^6$. [Color figure can be viewed at wileyonlinelibrary.com]

heat transferred on the right wall improves with AR except for $\phi = 45^\circ$. The right wall transferred less heat when $AR = 3.0$ compared with AR of 2.5.

7 | CONCLUSIONS

The present research investigated the impact of the Grashof number, Reynolds number, angular orientation, and elliptical cylinder aspect ratio on average Nusselt number, rate of heat transfer, and the general fluid flow characteristics inside a square enclosure containing a centrally fitted elliptical cylinder whose wall was heated sinusoidally. The horizontal walls of the square enclosure are insulated thermally, while its vertical walls are conditioned to a cold fixed temperature. The major conclusions of our investigation are as follows:

- For all the cylinder positions and aspect ratios considered, the fluid flow and heat transfer in the geometry for Grashof number values of $10^3 \leq Gr \leq 10^4$ are conduction-dominant while convection-dominant in the range of Grashof number values of $10^4 \leq Gr \leq 10^6$.
- For Reynolds number, $Re = 100$, the critical orientation of the ellipse beyond which the effect of inclination becomes insignificant is a function of the Grashof number and aspect ratio.

- Vortex strength and cell numbers significantly depend on the aspect ratio, particularly for thermal heating of $Gr = 10^4$.
- The heat transfer from the cylinder wall is marginally affected by cylinder orientation for aspect ratios of $1.5 \leq AR \leq 2.0$, and for aspect ratio values of $2.5 \leq AR \leq 3.0$, inclination angle increment does not result in heat transfer augmentation.
- For all the cylinder orientations considered, the average heat transfer from the cylinder is independent of the cylinder orientation when the aspect ratio is 1.0.
- For $10^3 \leq Gr \leq 10^5$, the aspect ratio increment enhanced the heat transfer rate of the right wall; for $Gr > 10^5$, the aspect ratio impact on the right wall is a function of the Grashof number and cylinder orientation.
- The impact of the inclination angle increment on the right wall diminishes as the inclination angle increases except for $Gr > 10^5$ where the rate of heat transfer is enhanced beyond $\phi > 45^\circ$.

NOMENCLATURE

a	semimajor axis
AR	aspect ratio
b	semiminor axis (m)
g	acceleration due to gravity (m/s^2)
Gr	Grashof number
K	thermal conductivity (W/m K)
L	dimensionless height of the enclosure
Nu	local Nusselt number
$\overline{Nu}$	average Nusselt number
p	dimensional pressure (N/m^2)
P	dimensionless pressure
Pr	Prandtl number
Re	Reynolds number
T	dimensional temperature (K)
T_c	dimensional cold wall temperature (K)
T_h	dimensional hot wall temperature (K)
U_0	lid velocity in the x -direction
u, v	dimensional velocity components (m/s)
U, V	dimensionless velocity components
x, y	dimensional Cartesian coordinates (m)
X, Y	dimensionless Cartesian coordinates

GREEK SYMBOLS

α	thermal diffusivity (m^2/s)
ν	kinematic viscosity (m^2/s)
ρ	density (kg/m^3)
Θ	dimensionless temperature

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