EFFECT OF WAVY WALL LOCATION ON THE NATURAL CONVECTION IN AN ENCLOSURE CONTAINING A CONCENTRIC HEATED CIRCULAR CYLINDER

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ABSTRACT :-
An investigation has been made to study numerically the effect of wavy wall locations on natural convection flow in a two dimensional enclosure filled with air containing a concentric heated circular cylinder. The external walls of the considered enclosure are maintained at a constant cold temperature (T_c) while the surface of the circular cylinder is kept at a constant hot temperature (T_h). The numerical simulation, based on a finite element approach, is performed using program ANSYS version 11. The governing equations with boundary conditions are solved iteratively using the Tri-Diagonal Matrix Algorithm (TDMA). The current work is validated with previous numerical results available in the literature. The study aims to determine the effect of wavy wall location which is represented by varying the inclination angle (∅=0°, 90° and 180°), and undulation numbers (n=1 and 2) on flow field and heat transfer for four selected Rayleigh numbers (Ra=10^3, 10^4, 10^5 and 10^6). Streamline and isotherm contours are determined. The effect local Nusselt number along the hot surface of the circular cylinder is also examined in this study. The results indicated that the average Nusselt number at the hot surface of the cylinder increases with increasing the Rayleigh numbers and reduces when undulation number increases from one to two.

Keywords: Natural convection, inclination angle, undulation numbers, Streamline and isotherm contours, Heat transfer rate.

تأثير موقعِ الجدار المتموّج على الحمل الحراري الطبيعيِ في حيز يَحتوي على أسطوانة دائرية ساخنة مركزية

قصي رشيد عبد الأمير فاروق حسين العناوي حميد كاظم حمزة

الخلاصة :-

في هذه الدراسة العددية، تم أجراء دراسة تحقيقيه حول تأثيرِ تغير موقعِ الجدار المتموّج على الحمل الحراري الطبيعيِ في حيز مربع ثنائي الأبعاد مَمتلئ بالهواءِ يَحتوي على أسطوانة دائرية ساخنة مركزية. إنّ الجدرانَ الخارجيةَ للحيز المدروس ثبتت عند درجة حرارة باردة (T_c) بينما سطح الأسطوانة الدائريةِ ثبتت عند درجة حرارية ساخنة (T_h). تم إجراء المحاكاة العددية المستندة على نظرية العناصرِ المحدودةِ باستخدام برنامجُ (ANSYS) نسخة 11. إنّ المعادلاتَ

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الحَاكِمةَ لهذه الدراسة تم َحْلُها باستعمال خوارزميةَ مصفوفةِ ثلاثيةِ القطرَ بشكل تكراري (TDMA). كذالك تم تحقيق من النتائج العددية للعملَ الحاليَ ومقارنتها مع النَتائِجِ العدديةِ المتوفرةِ في الدراسات السابقة. إنّ هدفَ الدراسةِ هو دراسةَ تأثيرَ تغير موقعِ الجدار المتموّج المُمَثّلُ بتَغيير زاويةِ الميلَ الحيز (Ø=0°، 90° و180 °)، وعدد التموجات السطحية للجدار (n= 1و2) على مجال التدفقِ ونقلِ الحرارةِ لأربعة أعدادِ Rayleigh مختارة (Ra= 10^3, 10^4, 10^5 و10^6). رسمت النتائج العددية على شكل نماذج خطوط تساوي درجة الحرارة وخطوط الانسياب. تم دراسة أيضاً تأثير عددَ Nusselt ألموقعي والمعدل على طول السطحِ الأسطوانة الدائريةِ الساخنة. بينت النَتائِجَ بأنّ عددَ Nusselt المعدل في السطحِ الأسطوانة يَزِيدُ بزيَادَةَ عدد Rayleigh ويُقل عندما يَزِيدُ عددَ التموّجِ مِنْ واحد إلى اثنان .

NOMENCLATURE:

A	Distance (m)
g	Gravitational acceleration (m/s^2)
K	Thermal conductivity (W/m.K)
L	Height or Width of the enclosure (m)
p	Pressure (Pa)
R	Cylinder radius(m)
T	Temperature (K)
V	Velocity component (m/s)

GREEK SYMBOLS :-

β	Volumetric coefficient of thermal expansion (1/K)
α	Thermal diffusivity (m^2/s)
Ψ	Dimensional stream function (m^2/s)
Μ	Dynamic viscosity (kg.s/m)
Γ	Kinematic viscosity (μ /ρ) (m^2/s)
Ρ	Density (kg/m^3)
Τ	Coordinate

SUBSCRIPTS

c	Cold
f	Film
h	Hot
y	Cartesian coordinate in vertical direction (m)
x	Cartesian coordinate in horizontal direction (m)

ABBREVIATIONS :-

C_p	Specific heat(kg/kg.K)
Nu_s	Local Nusselt number on cylinder surface
$\overline{Nu}$	Average Nusselt number
Pr	Prandtl number (ϑ/α)
Ra	Rayleigh number $(g\beta L^3 \Delta T/\vartheta\alpha)$

INTRODUCTION :

Heat transfer and fluid flow for natural convection in a square enclosures containing of wavy walls are of great interest in many engineering and industrial applications especially in the mechanical engineering field such as nuclear reactor cooling, solar energy collectors, steam radiators, electronic devices, and energy transfer in rooms of the buildings. Several earlier studies have been demonstrated that the heat transfer is improved by surface modification or the utilize of baffles or fins mounted on the surface of the enclosure walls. For example, **Mahmud et al., 2002** conducted numerical predictions of heat transfer and fluid flow characteristics inside an enclosure surrounded by two isothermal wavy walls and two adiabatic straight walls. The influence of different aspects such as surface undulations, surface roughness, amplitude and phase of the wave on non-Darcian natural convection in a wavy vertical enclosure filled with a porous medium were analyzed numerically by **Kumar and Shalini, 2003**. They concluded that the heat transfer into the system decreases with increasing both of amplitude and number of waves per unit length. A numerical analysis was performed by **Abdallah et al., 2006** on the thermal laminar natural convection in boundary layer over a vertical wavy cylinder. The wavy wall temperature was kept at a constant value greater than the ambient temperature. They studied effects of the wavy geometry on the local and mean Nussselt number. **Saha et al., 2008** analyzed steady state and a 2-D natural convection in inclined enclosure with a sinusoidal wavy walls. In their study, two vertical sinusoidal wavy walls were kept at cold temperature, while an isoflux heat source was discretely embedded at the bottom wall. The length of isoflux heat source was varied from 20 to 80% of the total length of the enclosure. They showed that the Nusselt number increases with inclination angle for different sizes of isoflux heat source. **Khanafer et al., 2009** has investigated numerically the effect of natural convection in a wavy cavity filled with a porous medium. The sinusoidal vertical wavy walls were maintained isothermally whereas the bottom and top horizontal straight walls were taken as adiabatic. This analysis showed that amplitude of the wavy surface and the number of undulations affect characteristics of heat transfer inside cavity. In addition, the strength of natural convection was increased with increasing Rayleigh number. **Hussain et al., 2011** studied of 2-D steady natural convective in a square enclosure with vertical vee-corrugated sidewalls and horizontal top and bottom surfaces. The results indicated that the natural convection phenomenon is greatly affected by increasing inclination angle of enclosure. **Salek and Nayeem, 2013** conducted a 2-D study in a corrugated enclosure. Vee and sinusoidal corrugations on vertical walls of the enclosure were used in their study. They considered the effect of corrugation geometry on natural convection. **Zemani et al., 2014** have studied heat transfer and flow characteristics for natural convection in cubical enclosure with hot partial partitions. This investigation showed that the partition length affects the average Nusselt number. It was observed that the heat transfer inside the undulated cavity with partition reduces noticeably than the cavity without partition. Another paper in the same group, presented a numerical analysis of natural convection in cubical enclosure with hot surface geometry and partial partitions. It was found that there is an influence of the hot wall geometry with partitions on the heat transfer rate and the flow inside the cavity. **Sompong and Witayangkurn 2014** used finite element method to investigate the effects of wavy geometry on natural convection in heated enclosure containing saturated fluid porous media. They found that the increase in number of undulations yields small effect on natural convection inside the enclosure whereas the increase in wave amplitude reduces the strength of convection because higher wave volume plays a barricade role.

The studies about natural convection around a solid body located inside enclosure with a variety of configurations were investigated extensively. **Shu et al., 2001** conducted numerically the natural convection phenomena between heated circular inner cylinder and a square outer enclosure at a Rayleigh number of $3*10^5$. The simulation results indicated that the flow separation, global circulation and the top space between the circular inner cylinder and the square outer enclosure affect the plume inclination. **Kumar and Dalal, 2006** obtained a 2-D analysis for natural convection inside enclosure containing square, horizontal, heated cylinder. Effects of the enclosure geometry were evaluated by three different aspect ratios placing the square cylinder at different heights from the bottom. **Hussain and Hussein, 2010** made a numerical solution to investigate the effects of Rayleigh numbers and vertical cylinder locations on performance of heat transfer and fluid flow for circular cylinder inside square enclosure. They observed that the numerical solutions produce a two cellular flow field between the enclosure and the inner cylinder. The unsteady natural convection induced by a temperature difference between a hot inner circular cylinder and a cold outer square enclosure was considered numerically by **Roslan et al., 2014**. The cylinder temperature was varied sinusoidally with time. Their results showed that the heat transfer rate increases by oscillating the source temperature signal. More recently, **Yuan et al., 2015** analyzed the natural convection inside horizontal concentric cylindrical enclosure on the flow field and heat transfer characteristics. It was concluded that the surface radiation and presence of larger top space and corners improves the heat transfer rate. Also, effect of surface radiation at higher temperature levels has significant role in the overall heat transfer inside enclosure.

Based on the above literature review, there is no research work that deal with effect of various wavy wall locations in enclosure containing a concentric hot circular cylinder.

Present work aims to visualize of the streamlines and isotherms in the enclosure filled with air for various inclination angles and number of undulations of wavy wall at different Rayleigh numbers. The heat transfer and fluid flow characteristics also have investigated by calculating the local and average Nusselt number along the hot surface of circular cylinder placed inside center of enclosure.

PHYSICAL DESCRIPTION OF THE PROBLEM AND MATHEMATICAL EQUATIONS

A schematic diagrams of the two dimensional wavy enclosure with dimensions ($L \times L$) are displayed in **Fig. 1**. The wavy enclosure contains concentric circular cylinder with radius (R=0.2L). This enclosure consists of one wavy wall and three straight walls. The surface profile of the wavy wall can be determined by the following relation

$$f(\tau) = a + 0.003 \times (\sin 2n\pi\tau) \tag{1}$$

Where a is the distance between the origin and wavy wall, n is undulation number and τ is coordinate toward the x-axis and y-axis which depends on wavy wall location. The origin point of the cartesian coordinates (x, y) is placed at center of the enclosure. The enclosure is long enough in the z-direction (i.e. the direction of the fluid flow and heat transfer is two-dimensional). The external walls of the considered enclosure are cold isothermally while the surface of circular cylinder is heated at a constant temperature. The following ranges of the parameters are considered: the inclination angle (∅) varies from 0° to 180°, Rayleigh number is altered from 10^3 to 10^6, the air is filled the enclosure with Pr equal to 0.71. In the present problem, the flow of air is assumed to be steady, incompressible and laminar. The air properties are constant except for the density change according to Boussinesq approximation while viscous dissipation term in the energy equation is considered absent. Radiation effect is negligible.

The two dimensional incompressible Navier-Stokes equations from fluid dynamics couple together with a heat transfer equation. The equations governed the problem represented by mass, momentum and energy energies for 2-D steady natural convection in the enclosure can be written with the dimension forms:

Mass conservation equation

$$\frac{\partial(\rho V_x)}{\partial x}+\frac{\partial(\rho V_y)}{\partial y}=0 \tag{2}$$

Momentum conservation equations

$$\frac{\partial(\rho V_x V_x)}{\partial x}+\frac{\partial(\rho V_y V_x)}{\partial y}=-\frac{\partial P}{\partial x}+\frac{\partial}{\partial x}\left(\mu\frac{\partial V_x}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu\frac{\partial V_x}{\partial y}\right) \tag{3}$$

$$\frac{\partial(\rho V_x V_y)}{\partial x}+\frac{\partial(\rho V_y V_y)}{\partial y}=\rho g\ \beta(T-T_c)-\frac{\partial P}{\partial y}+\frac{\partial}{\partial x}\left(\mu\frac{\partial V_y}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu\frac{\partial V_y}{\partial y}\right) \tag{4}$$

Energy conservation equation

$$\frac{\partial}{\partial x}(\rho V_x C_p T)+\frac{\partial}{\partial y}(\rho V_y C_p T)=\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right) \tag{5}$$

The thermo-physical properties of air (ρ ,cp ,μ ,k and α) are obtained from tables at film temperature (T_f). The present study is conducted using the ANSYS version 11 software, a commercially available computational fluid dynamics package to solve the above equations. Rayleigh number (R_a) is defined as:

$$R_a=\frac{g.\beta(T_h-T_c).L^3}{\alpha.\nu} \tag{6}$$

Where $\beta=\frac{1}{T_f}, and \qquad T_f=\frac{T_h+T_c}{2}$

The above equations are subject to the problem boundary conditions :

At all the solid walls: $V_x=V_y=0$, $\psi=0$

The thermal boundary conditions are:

At enclosure walls: $T=T_c$

At circular cylinder surface: $T=T_h$

The stream function is computed for two dimensional structure and is defined by a way of its derivatives:

$$\frac{\partial\psi}{\partial x}=-\rho V_y \tag{7}$$

$$\frac{\partial \psi}{\partial y} = \rho V_x \tag{8}$$

The local Nusselt number can be calculated to estimate the heat transfer rate along the hot cylinder surface. It is a nondimensional term that defined as follows:

$$Nu_s = \frac{h.ds}{k_f} \tag{9}$$

Where: *ds* is the arc distance between the two nodes, *h* is the heat transfer coefficient which is determined from boundary condition as

$$-k\frac{\partial T}{\partial s} = h\left(T - T_f\right) \tag{10}$$

The average Nusselt number ($\overline{Nu}$) is determined by integration local Nusselt number around the hot circular cylinder as follows:

$$\overline{Nu} = \frac{1}{2\pi}\int_o^{2\pi} Nu_s dS \tag{11}$$

3- NUMERICAL SOLUTION AND RESULTS VALIDATION

In the present study, ANSYS version 11 is employed as the numerical solver to solve this problem. This program used the finite element approach to discretize the above the governing equations with boundary conditions of this problem on a grid where all the variables are collocated. The discretized differential equations were solved iteratively using the Tri-Diagonal Matrix Algorithm (TDMA). The differential equations have four unknown dependent variables which are the velocity field components (V_x and V_y), the pressure (P) and the temperature(T). The grid system over the computational domain of all enclosures is created using unstructured quadratic element. A grid convergence criterion test was done to ensure the accuracy of the simulation outputs and to find out an appropriate grid sizes. The convergence criterion in the solution domain is chosen as 10^{-6} for all dependent variables in all cases. Schematic of grid distribution is given in **Fig. 2** for configuration of wavy enclosure staggered and clustered towards the all enclosure walls and concentrated around the inner circular cylinder.

To validate the solution procedure, the numerical results obtained for both the streamlines and isotherms contours in an air-filled square enclosure containing heated circular cylinder were compared with results published by Roslan et al. [14]. The comparison was done using the following parameters: Ra = 10^5, Pr = 0.71, R=0.2L and Ø = 0°. As can be seen in the **fig. 3,** close agreements were observed between the present results and the presented in [14].

DISCUSSION OF RESULTS

Numerous results can be drawn from the output of program ANSYS, such as velocity vectors, pressure, flow trace, streamlines and isotherms contours, but because of the space limitations, only isotherms and streamlines contours results are shown in the **figs. 4** and **5**. Also, heat transfer rate along the hot surface of circular cylinder is expressed by local and average Nusselt number which is illustrated as in the figs. 6 and 7, respectively.

Effect of Rayleigh number on flow and thermal fields

As can be seen in the **figs. 4** and **5**, three cases are discussed: case(I) Ø=0°, case(II), Ø=90°, and case(III) Ø=180° for both n=1 and n=2 respectively, and for four selected Rayleigh numbers (10^3, 10^4, 10^5 and 10^6). At different Rayleigh numbers the fluid adjoining the concentric heated circular cylinder becomes hotter than that of enclosure walls and therefore becomes less density than the bulk fluid. Hence, the fluid then moves upward at the sides of the cylinder and falls downward when fluid contacts with the cold wall at top of the enclosure. This motion creates two rotating eddies developing in the right and left halves of the enclosure.

Fig. 4 shows the isotherms and streamlines contours within enclosure having wavy wall with one undulation for different Rayleigh numbers and inclination angles. When Ra = 10^3, the fluid motion is weak inside the enclosure due to its lower effect of the buoyancy force. The streamlines contours in this range of Rayleigh number show two rotating eddies near the right half and left half of the enclosure with one inner vortices inside each one of the eddies. These eddies expand in the longitudinal axis except in the case(II) at Ø=90°. At this case, the vertical wavy wall location pushes the eddy toward the top enclosure where a single large eddy appears. The flow circulation increases than the other cases at Ø=0, and Ø=180 and it is clear in magnitude of maximum stream function

(Ψ_{max}=+0.00289*10^{-3} for case (I) , Ψ_{max}=+0.00344*10^{-3} for case (II) and Ψ_{max}=+0.00254*10^{-3} for case (III)). When Ra = 10^4, the buoyancy force is still weak. Therefore, the patterns of streamlines are almost symmetric as compared to previous Rayleigh number for each case despite some differences in the values of maximum stream function between two Rayleigh numbers (Ψ_{max}=+0.0293*10^{-3} for case (I), Ψ_{max}=+0.037*10^{-3} for case (II) and Ψ_{max}=+0.026*10^{-3} for case (III)). The difference in the values of maximum stream function becomes more as Rayleigh number increases to 10^5 (Ψ_{max}=+0.18*10^{-3} for case (I), Ψ_{max}=+0.22*10^{-3} for case (II) and Ψ_{max}=+0.185*10^{-3} for case (III)). The flow circulation becomes more and a secondary flow is developed at upper part of the inner cylinder as shown in case (I). Therefore, the effect of buoyancy force becomes larger due to the effective fluid motion coming from the higher temperatures near the hot cylinder toward the enclosure walls. The convection heat transfer becomes more considerable. With further increase in Rayleigh number (Ra=10^6), the flow strengthens and the maximum stream function values increase more (Ψ_{max}=+0.284*10^{-3} for case (I), Ψ_{max}=+0.39*10^{-3} for case (II) and Ψ_{max}=+0.27*10^{-3} for case (III)). This is because the secondary flow is reinforced at top surface of the inner cylinder as illustrated in case (I) and case(III). The buoyancy force dominates over the entire domain.

The characteristics of the temperature distributions are represented by isotherms. When Rayleigh numbers varied from 10^3 to 10^4, the isotherms are also similar around the hot cylinder which displays as parallel rings around the surface of cylinder. This is because the difference in temperature between cylinder surface and enclosure walls is very small (about ~ 4°C). Therefore, the conduction or diffusion heat transfer is the dominant than the convection. This is evident from the pattern of isotherms. When Rayleigh number increase to Ra=10^5, the effect of conduction heat transfer decreases, so the temperature difference increases to about ~ 36°C. The isotherms start to curve around the cylinder due to the convection heat transfer. Thickness of the thermal boundary layer on the upper cylinder surface becomes thinner. A thermal plume appears on the top of the cylinder surface and therfore the thermal gradient increases in the upper part of the enclosure. The shape of thermal plume depends on the location of wavy wall represented by number of undulations of wavy wall and inclination angles of enclosure for each Rayleigh numbers. The shape of thermal plume is flat near the upper wall of enclosure with exception case(I) where the wavy wall is placed there. At this case (Ø=0), the thermal boundary layer on the top of cylinder surface separates into two parts displaying the formation of two thermal plumes with sloping about 45^{o} from the vertical mid-plane of the enclosure. When further increase in Rayleigh number (Ra=10^{6}), the thermal boundary layer thickness becomes more thinner when the temperature difference on the surfaces of the enclosure and cylinder increases (about ~ 367°C). Therefore, the isotherms distort from the hot cylinder surface towards the cold top wall of the enclosure. The thermal boundary layer separates from top of the cylinder because of presence of secondary vortices at upper cylinder surface as illustrated in case (I) and case(III). Therefore, a strong plume comes into view and rises from the top of cylinder towards the upper wall of enclosure where the plume impinges there.

Fig. 5 shows the streamlines and isotherms contours within the two undulation wall enclosure for different Rayleigh numbers and inclination angles. As shown in **fig. 5a** and **fig. 5b**, the streamlines and isotherms contours are somewhat similar at low Rayleigh numbers(10^{3} and 10^{4}). At this range of Rayleigh numbers, there is a little increasing in the maximum stream function (Ψ_{max}=+0.00263*10^{-3} to +0.0247*10^{-3} for case (I) , Ψ_{max}=+0.00319*10^{-3} to +0.0224*10^{-3} for case (II) and Ψ_{max}=+0.00244*10^{-3} to +0.0264*10^{-3} for case (III)). Therefore, the fluid motion is weak at the bottom region of the enclosure than that at the middle and top regions. Consequently, two main eddies are formed as before inside the enclosure and symmetric with respect to the vertical centerline. Due to surface waviness on right side wall (see case(II) i.e., Ø=90°, n=2), the shape of fluid circulation near that region is corrugated when fluid passes along that wall. It is found that the shape of fluid flow is squeezed due to the large wave volume in middle region of the enclosure reducing the space between the cylinder and vertical wall. For low Rayleigh numbers, the isotherms are packed as annular rings around the circular cylinder which guarantee that the mechanism of heat transfer by the conduction or diffusion mode is dominated while effect of the convection is very low. When Ra increases to 10^{5}, the flow circulation becomes more important due to the increased buoyancy as can be seen by the distorted isotherms starting from cylinder surface to enclosure walls. So that the thermal boundary layer separates from the top of the cylinder surface towards one direction or two directions as can be seen in case (I) and then impinges close to the horizontal wall of the enclosure. The dominating mechanism of heat transfer changes from conduction to convection. When Ra increases from 10^{5} to 10^{6}, the maximum stream function increases for all cases ((Ψ_{max}=+0.163*10^{-3} to + 0.255*10^{-3} for case (I), Ψ_{max}=+0.179*10^{-3} to +0.341*10^{-3} for case (II) and Ψ_{max}=+0.191*10^{-3} to +0.280*10^{-3} for case (III)). The strength of the flow becomes stronger and the circulation core of eddies changes from the upper portion to bottom portion of enclosure because of forming a secondary flow at the top of the cylinder surface as can be seen in case (I) and case(III). In these cases, the fluid then flows towards the bottom part of cylinder, showing the formation of a thermal plume. Therefore, the thermal boundary layer separates to two halves at the top of the cylinder surface and consequently two thermal plumes form there.

Effect of number of undulation on the streamline contour is depicted in Figs. 4 and 5 four selected Rayleigh numbers (10^3, 10^4, 10^5 and 10^6) and inclination angles (∅=0°, ∅=90°, and ∅=180°). The one undulation wall enclosure has maximum stream functions than that of two undulation wall enclosure. It is also observed from values of maximum stream functions that the Ψ_{max} for case (II) is the higher than the other cases for both of one and two undulation wall enclosure. These results remain for the same behavior with increasing the values of Ra. The effects of two undulations wall on the isotherms and streamlines contours for each cases become significant at high values of Rayleigh number. On the other hand, it can be observed from figures that the isotherms shape changes significantly from the uniform annular shape around the hot cylinder for the low values of Ra (10^3 and 10^4) to non-uniform annular shape around the hot cylinder for the high values of Ra (10^5 and 10^6).

Effect of Rayleigh number on characteristics of heat transfer

In this section, the characteristics of heat transfer for steady natural convection in wavy enclosure containing a concentric circular cylinder have been discussed. The values of local Nusselt number in **Figs. 6a** to **6c** are plotted along the surface of the cylinder for Rayleigh numbers in the range of $10^3 \leq Ra \leq 10^6$. Effects of number of undulations(n) and inclination angles(∅) of enclosure on the heat transfer rate along a concentric heated cylinder are clearly observable in these plots. At low Ra(10^3), the heat transfer rate is small for each case especially case (III): ∅=180, n=2 while it is high for case (II): ∅=90, n=2 due to nearing of vertical wavy wall. At Ra=10^4, the local Nusselt number values increase with an increase in Rayleigh number especially case (III): ∅=180, n=2. At lower values of Rayleigh numbers (10^3 and 10^4), the heat transfer by conduction occurs from the cylinder surface towards the side of enclosure walls. At Rayleigh numbers ranging from 10^4 to 10^5 and for both n=1 and n=2, the minimum values of local Nu mostly found at the upper point of the cylinder surface (i.e. θ= 90°) while the maximum values of local Nu around the hot cylinder observed at bottom point of the cylinder surface (i.e. θ= 270°)

where results steepest temperature gradient. In this range of Ra, there are significant increasing in values of local Nusselt number because of the increasing convection effect on heat transfer with an increase in Ra. It also found that the Nu_s raises and drops rapidly at the upper point of the cylinder surface (i.e. θ=90) especially in the case(I): Ø=0°, for both n=1 and n=2. This is because of the surface waviness of wall which is decreasing space between the inner cylinder and the top of wavy wall of the enclosure. With increase in the Ra from 10^5 to 10^6, the values of local Nu is less compared with Ra=10^5 because the dominant flow changes from the upper half to the bottom half of the enclosure which results decreasing in values of local Nusselt number there.

Fig. 7 shows the averaged Nusselt number versus Rayleigh numbers for different values of inclination angle and number of undulations calculated for heated circular cylinder. This figure illustrated that the values of average Nu increases with increasing Rayleigh numbers and decreases with increasing number of undulation. On the other hand, increasing the area under curves of averaged Nu results in an improved heat transfer through the enclosure. For this reason, the vertical wavy wall of enclosure (Ø=90°) gives enhancement of heat transfer followed by Ø=180° (bottom wavy wall) and Ø=0° (upper wavy wall). This behavior explains that the overall heat transfer process is sensitive to number of undulation and the inclination angle which is almost expected as seen from local Nusselt number in Fig. 6.

CONCLUDING REMARKS

Analysis of isotherms, streamlines, local and average Nusselt number were carried out to investigate the effect of the wavy wall locations and number of undulations on the heat transfer and fluid flow in the cooled enclosure containing a concentric heated circular cylinder for different Rayleigh numbers changing with the range of $10^3 \leq Ra \leq 10^6$. Calculations are performed by using program ANSYS version11. The results have shown that the average Nusselt number at the heated cylinder surface increases with increasing the Rayleigh numbers for all considered cases and reduces when undulation number increases from one to two. In addition, the results have also shown that the average Nusselt number can be optimized by placing the wavy wall at the left side as compared with other sides of enclosure.

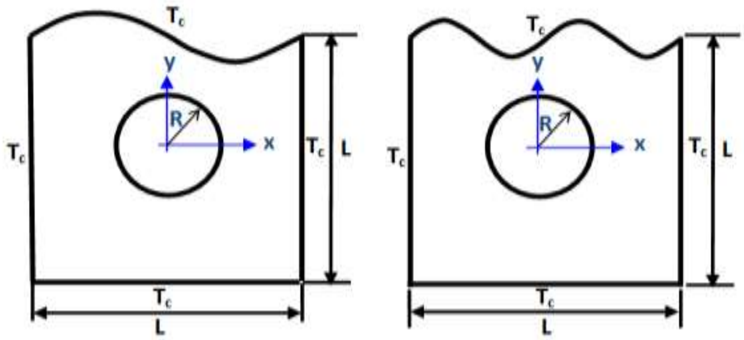


Fig. 1. Schematic diagrams of wavy wall enclosure at an inclination angle ∅=0° for one undulation (n=1) and two undulation (n=2)

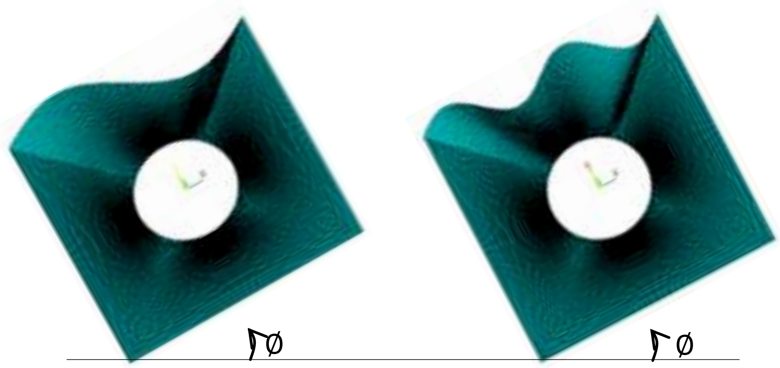


Fig. 2. A typical grid distribution with quadratic elements.

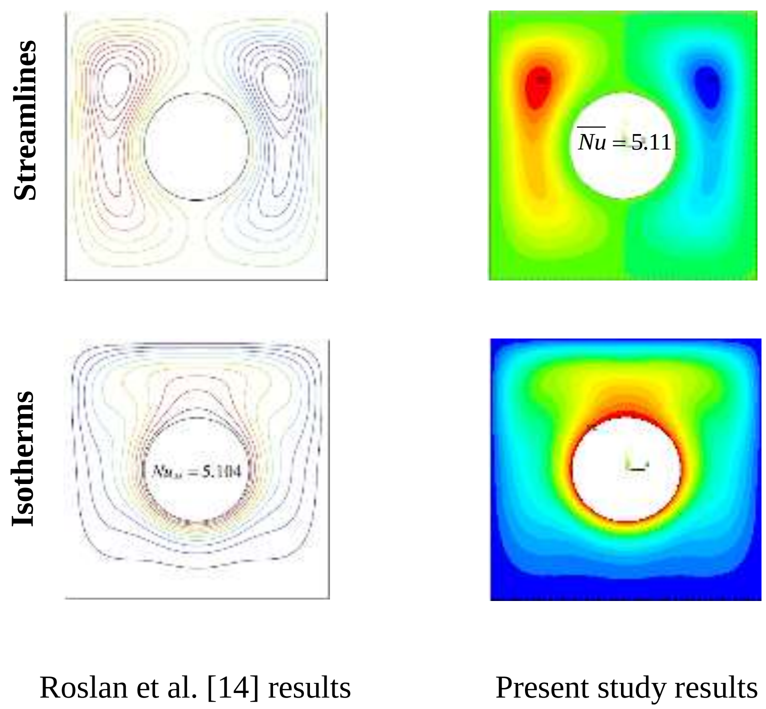
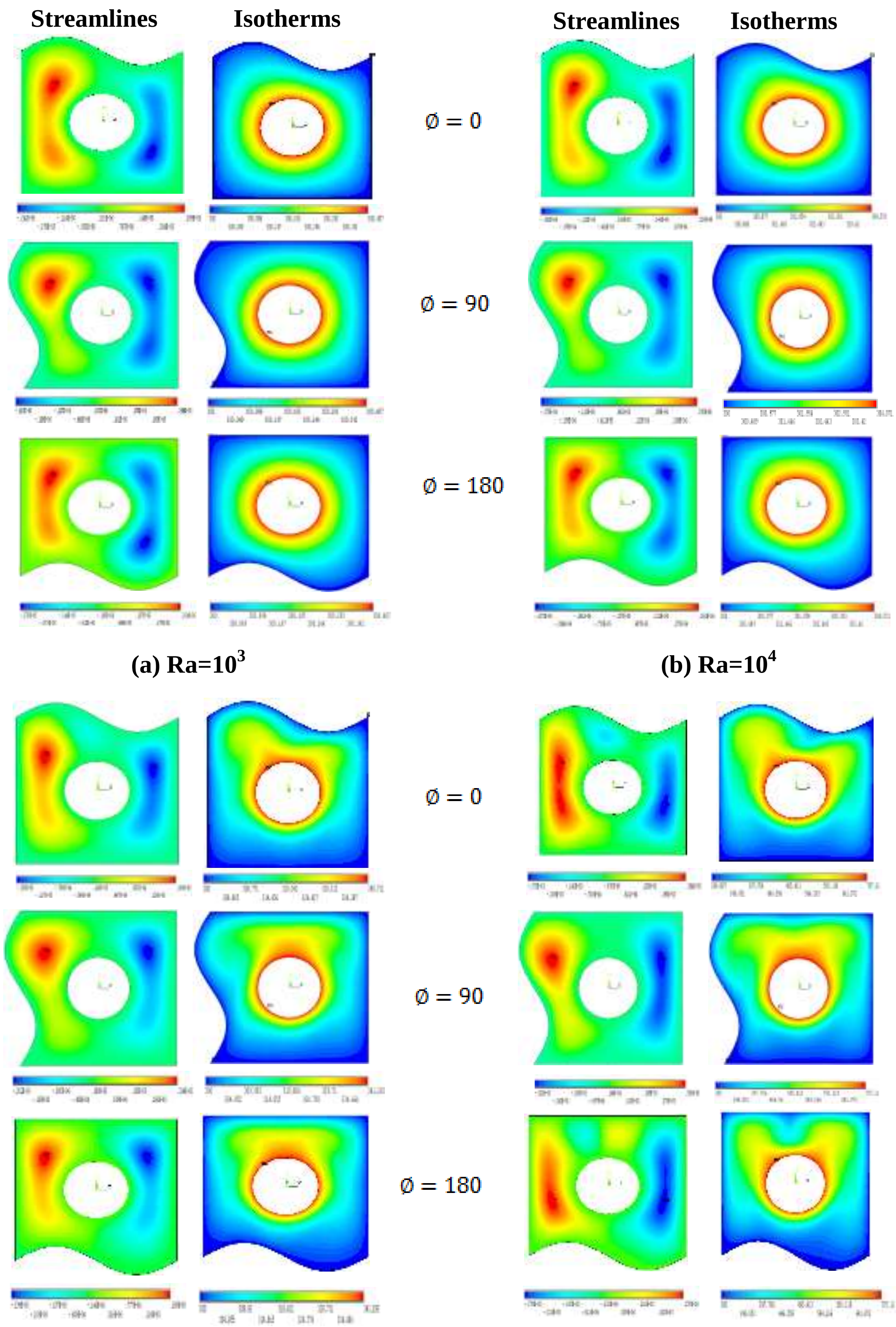


Fig. 3. Comparison of the streamlines and isotherms patterns between the percent study (right) and Roslan et al. [14] (left) at Ra=10^5 and R=0.2L.



(a) Ra=10^3 (b) Ra=10^4

(c) Ra=10^5 (d)) Ra=10^6

Fig. 4: Variation of streamlines and isotherms with inclination angle for one undulation

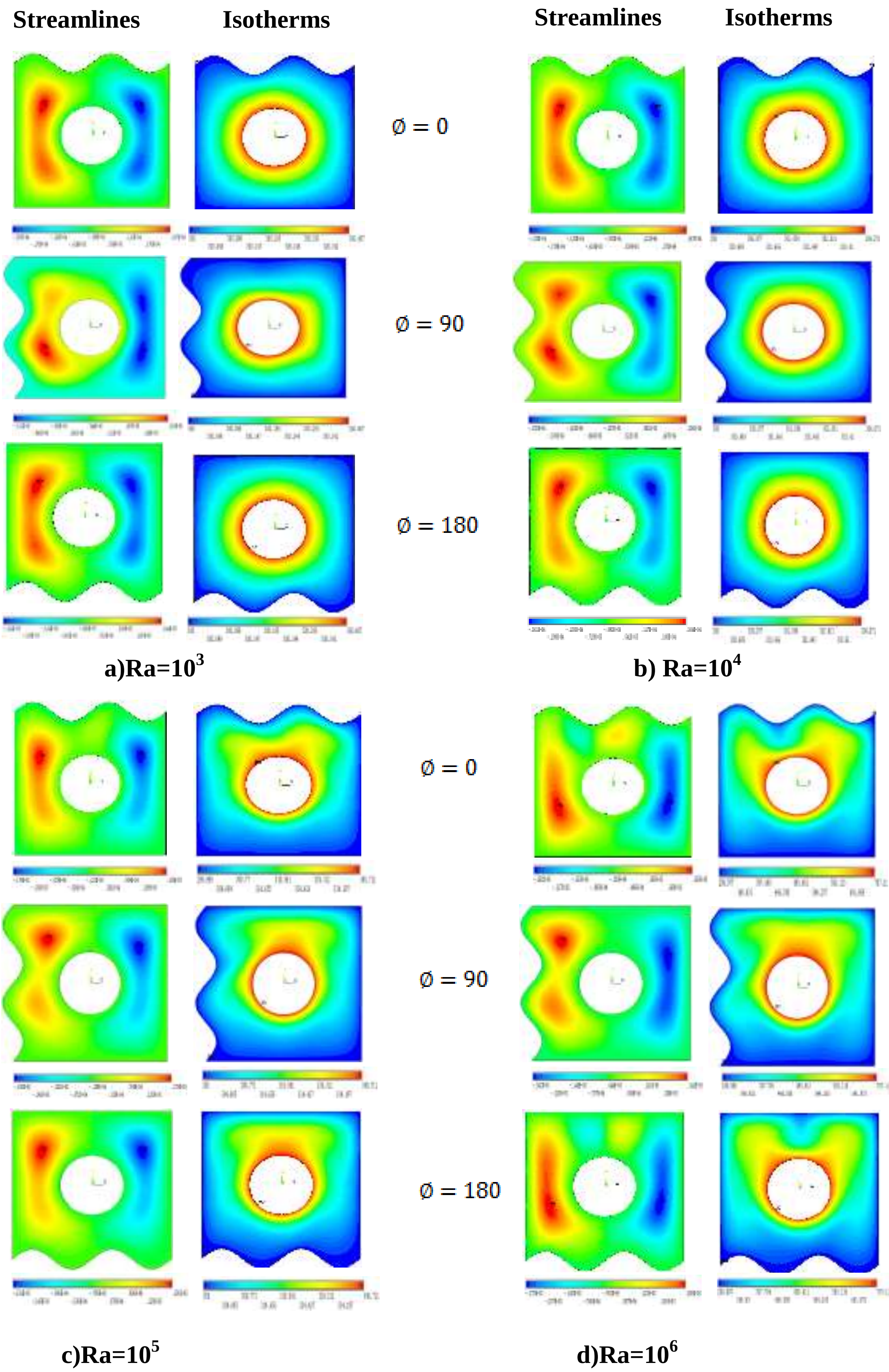


Fig. 5: Variation of streamlines and isotherms with inclination angle for two undulation

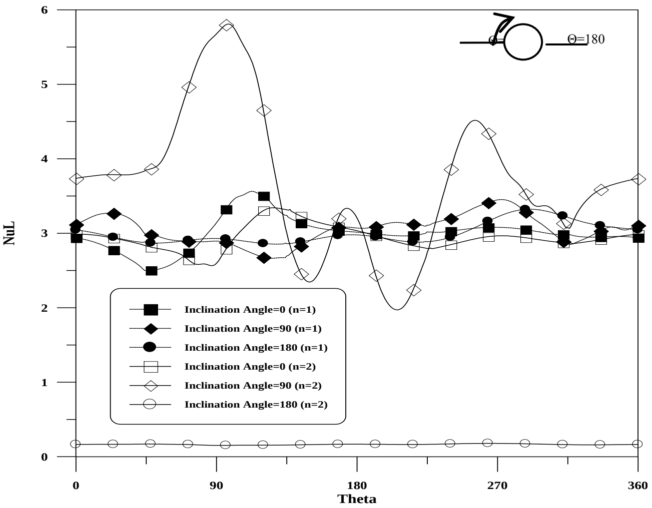


Fig.(6a) Variation of local Nusselt number along heated circular cylinder

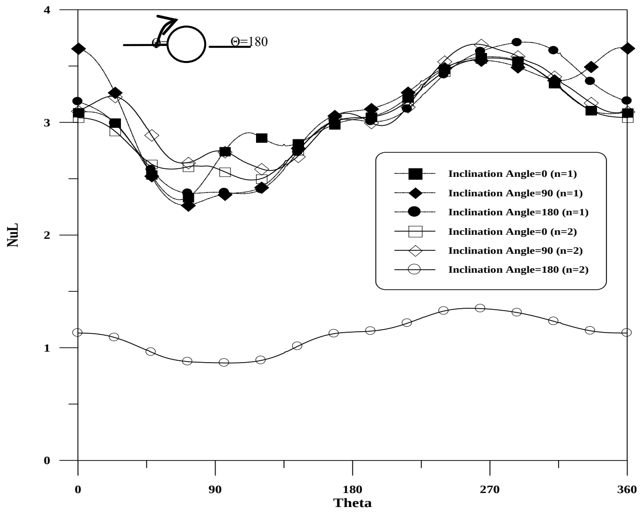


Fig.(6b) Variation of local Nusselt number along heated circular

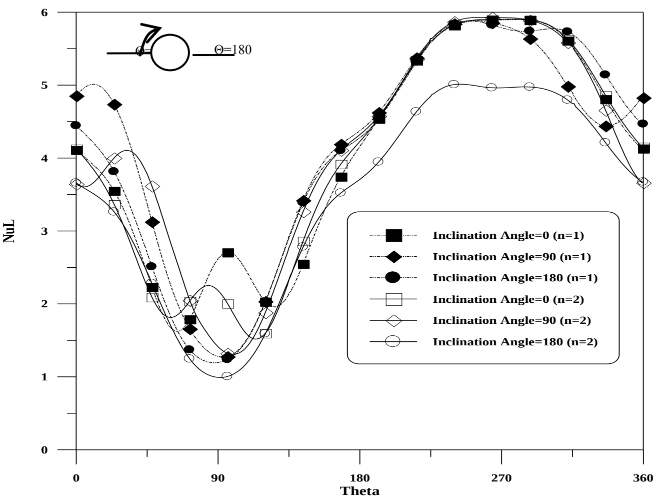


Fig.(6c) Variation of local Nusselt number along heated circular cylinder at $Ra=10^5$

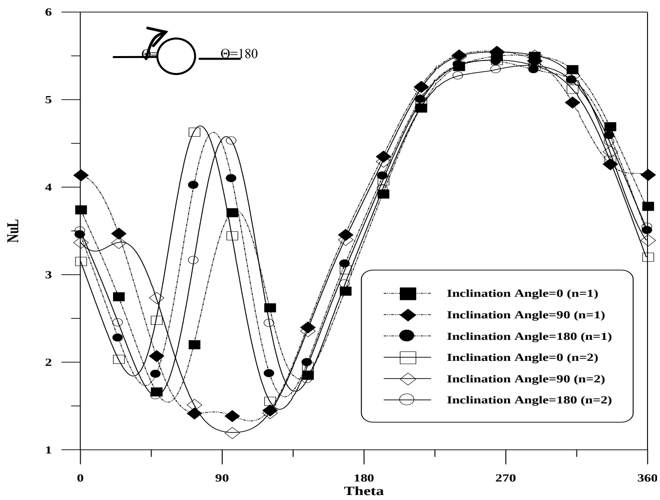


Fig.(6d) Variation of local Nusselt number along heated circular

Fig. 6. Variation of local Nusselt number with Rayleigh numbers

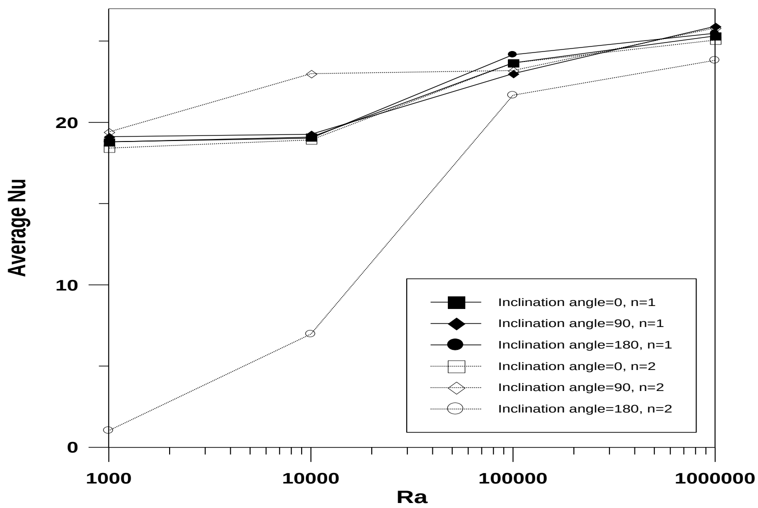


Fig. 7. Average Nu versus Ra with different parameters.

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