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Numerical Study on Natural Convection and Fluid Flow Inside a Tilted Wavy Enclosure

دراسة نظرية للحمل الحر لمائع داخل حيز متعرج

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الملخص:

دراسة رقمية لانتقال الحرارة بالحمل الطبيعي داخل حيز مغلق مائل. الجدار العلوي ذو مسطح مستو عند درجة حرارة منخفضة بينما الجدار السفلي ذو مسطح متعرج عند درجة حرارة مرتفعة. كلا من الجدارين الايمن والايسر معزولان حراريا. تم استخدام المعادلات الواصفة للسريان وانتقال الحرارة في صورة ثنائية الابعاد ذات المحاور الكارتيزية. تم وضع هذة المعادلات في صورة لابعدية مناسبة. تم حل هذة المعادلات اللابعدية بطريقة الفروق النهائية بالحاسب الالي بلغة الفورتران. تم دراسة العوامل المؤثرة علي السريان وهي رقم رايلى (5×10^3 - 10^6)، نسبة الارتفاع المميز للحيز (1-5)، النسبة البعدية المميزة للحيز (1-5) و زاوية الميل (10- 60) للحصول علي توزيع السريان وكذلك رقم نوسلت المحلي والمتوسط وذلك باعتبار الهواء كمائع مستخدم للدراسة. وجد ان دوامة دورانية قوية في اتجاة عقارب الساعة تتكون معتمدة علي العوامل المؤثرة علي السريان. وجد ان معدل انتقال الحرارة يزداد بزيادة رقم رايلى و النسبة البعدية المميزة للحيز. وجد ان الفقد في انتقال الحرارة يقل عند استخدام نسبة الارتفاع المميز للحيز تساوي 2ونسبة البعدية المميزة اكبر من 1عند زاوية ميل 20-40 مع المحور الافقي.

Abstract

Natural convection heat transfer and fluid flow is investigated, numerically, inside a tilted wavy walled enclosure. Simulating solar air heater, flat ceiling is considered as cover (cold temperature) of solar air heater and wavy wall as absorber (hot temperature). The flow describing equations are presented in, two dimensional in Cartesian coordinates, dimensionless form by introducing appropriate dimensionless independent and dependent variables. The dimensionless form of the governing equations is solved by using the finite divided difference technique. Solving these equations, the temperature distribution and the hydrodynamic flow field in the enclosure are predicted and accordingly, local and average Nusselt number can be calculated. The effects of system parameters on the natural convection heat transfer inside enclosure are simulated. These parameters are Rayleigh number ($5\times10^3 \le Ra \le 10^6$), the characteristic height ratio of the channel ($1 \le A \le 5$), characteristic geometric ratio ($1 \le L \le 5$) and the inclination angle ($10° \le \phi \le 60°$) on heat transfer performance are investigated considering air as working fluid. The numerical results show that, the character of clockwise rotating vortex is found to be, strongly, depending on problem parameters. Heat transfer is increased with increasing Rayleigh number and characteristic geometric ratio. To suppress the natural convection heat loss effectively, A should be equal to 2, L larger than 1 and ϕ between 20° and 40°.

Nomenclature

a	Amplitude height of the wavy wall	m
A	Characteristic height ratio	--
g	Gravity of acceleration	m/s^2
h	Heat transfer coefficient	$W/m^2\ K$
H	Height of the channel	m
$\overline{H}$	Average height of the channel	m
L	Characteristic geometric ratio of wavy wall, $L = \lambda/4a$	--
Nu_x	Local Nusselt number	--
Nu	Average Nusselt number	--
p	Dimensionless pressure	--
Pr	Prandtl number, $Pr = \upsilon/\alpha$	--
Ra	Rayleigh number, $Ra = \left(g\,\beta\,\Delta T\,\overline{H}^3/\upsilon^2\right) Pr$	--
T	Dimensionless temperature	--
u, v	Dimensionless velocity components in x and y directions	--
x, y	Dimensionless Cartesian coordinates	--
Greek symbols		
α	Thermal diffusivity	m^2/s
β	Thermal expansion coefficient	$1/K$
ρ	Density	kg/m^3
υ	Kinematics viscosity	m^2/s
ϕ	Inclination angle	*degree*

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λ	Wavelength	m
Subscripts		
C, H	Cold wall and hot wall	--
Superscripts		
*	Dimensional form	--

1. Introduction

Natural convection heat transfer and fluid flow in enclosure with flat walls are important in engineering applications such as, cooling of electronic devices, building design, furnaces, solar collectors, and double pane windows. In addition to enclosures with flat walls, inclined or wavy walled enclosures have great importance for natural convection in roofs or solar collectors to control heat transfer. However, flow and temperature fields are very complex in these kinds of complex geometries.

Yao [1] investigated the flow and temperature field near a vertical wavy wall. The local heat transfer rate is found to be smaller than that of the flat plate case and decreases with increase of the wave amplitude. Mahmud et al. [2] performed a numerical solution to investigate free convection inside an enclosure bounded by two isothermal wavy walls and two adiabatic straight walls. The height ratio was found to be the most important parameter for heat transfer and fluid flow. Kumar and Shalini [3] investigated the effects of surface undulations on natural convection in a porous enclosure with global cumulative heat flux boundary conditions for different undulation numbers and thermal stratification levels. The local Nusselt number was very sensitive to thermal stratification. Rees and Pop [4] performed an analytical study by solving boundary layer equations for wavy surface in a porous medium. Adjlout et al. [5] investigated the inclined square enclosure with right vertical wall having a sinusoidal function with different undulation number. The governing equations were solved in stream function–vorticity form using curvilinear coordinates. Das and Mahmud [6] investigated the natural convection inside an enclosure that contains two wavy isothermal walls (bottom and top) and two adiabatic straight (left and right) walls using the finite volume method. The temperature and flow field was affected by the amplitude-wavelength ratio. Gao et al. [7] performed a numerical study to investigate natural convection inside a wavy collector. The results indicated that, in order to suppress the natural convection heat loss effectively, characteristic height ratios should be larger than 2, amplitude lengths should be larger than 1 and inclination angles should be smaller than 40^{o}. Dalal and Das [8] performed a numerical study to obtain temperature and flow fields in an inclined square cavity whose right wall has a sinusoidal function. A spatially variable wall temperature to the upper wall was applied. The results indicated that, with the increase in amplitude, the average Nusselt number on the wavy wall was high at low Rayleigh numbers. Lun-Shin Yao [9] investigated, theoretically, the natural-convection boundary layer along a complex vertical surface created from two sinusoidal functions. The total heat-transfer rates for a complex surface were found to be greater than that of a flat plate. The enhanced total heat-transfer rate was seemed to be depending on the ratio of amplitude and wavelength of a surface. Aounallah et al. [10] investigated, numerically, the turbulent natural convection of air flow in a confined cavity with two differentially heated side walls. The strong influence of the undulation of the cavity and its orientation was indicated. The trend of the local heat transfer rate was varied with different frequencies for each undulation. An increase in the convective heat transfer on the wavy wall surface compared to the square cavity for high Rayleigh numbers was caused by turbulence. Abdalla Al-Amiri [11] studied numerical the mixed convection heat transfer in lid-driven cavity with a sinusoidal wavy bottom surface. The implications of Richardson number, number of wavy surface undulation and amplitude of the wavy surface on the flow field and heat transfer characteristics were investigated. The trend of the local Nusselt number was found to follow a wavy pattern. The results illustrated that the average Nusselt number increases with an increase in both the amplitude of the wavy surface and Reynolds number. Recently, the boundary heat flux profiles in an enclosure containing solid conducting block was investigated by Fu-Yun Zhao et al. [12]. Numerical solutions were obtained for the case of a square enclosure centrally-inserted with a solid block and subjected to an unknown heat flux on one side and to known conditions on the remaining sides. The accuracy of the heat flux profile estimations was shown to depend, strongly, on the thermal Rayleigh number, body size and relative thermal conductivity of the solid material.

The present study investigates, numerically, the natural convection inside the channel between the flat-plate cover and wavelike absorber of the cross-corrugated enclosure. This configure ration simulates the solar air heater. These kinds of study are crucial for minimizing the heat loss due to natural convection and improving the efficiency of the solar air heater. A cross-corrugated solar air heater is developed, whose absorber and back plate (they form the channel where the air is heated) are placed across from each other and both have the sine-wave shape. This study is different from the others due to the shallow geometry. Rayleigh number, inclination angle, characteristic height ratio and characteristic geometric ratio of enclosure are considered as study parameters. The characteristic height ratio of the channel, A, is defined as the ratio of the channel height to the amplitude height of the wavelike absorber. While the characteristic geometric ratio, L, is defined as the ratio of one-fourth of the wavelength to the amplitude height of the absorber.

2. Problem formulation

The dimensions of the tilted wavy enclosure are shown in Figure. 1, along with the coordinate system and boundary conditions. The computational domain is chosen as the upper part of the model. The problem treated is a two-dimensional heat transfer in an inclined closed cavity. The hot bottom wall, wavelike absorber, is wavy with a constant temperature T_H. The cold top wall, cover, is opposite to the latter with a constant temperature T_C while the other sides are adiabatic of average height $\overline{H}$. In this figure, a, λ and ϕ denote amplitude, wavelength and inclination angle respectively. The shape of the wavy bottom wall is taken as sinusoidal with fixed wavelength of 0.4 m for all cases. The Rayleigh number is varied up to 10^6 while Prandtl number is equal to 0.71.

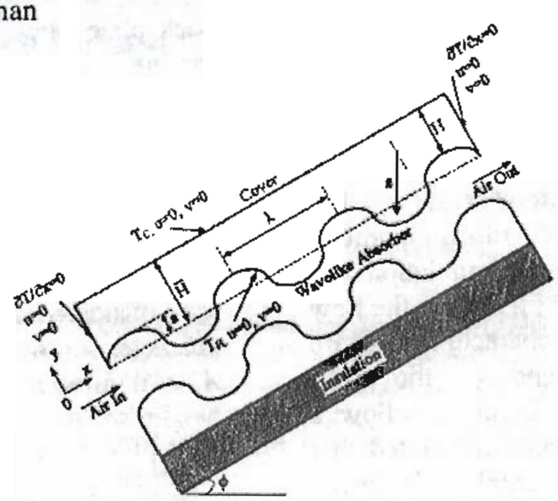


Figure. 1. Schematic diagram of tilted wavy enclosure.

The steady-state viscous incompressible flow inside a closed cavity and a temperature distribution is described by the Navier–Stokes and the energy equations. The Boussinesq approximation is used with the assumptions of constant properties and negligible viscous dissipation. To put the governing equations and their boundary and initial conditions in dimensionless form, one introduces the following dimensionless independent and dependent variables as;

$$x=\frac{x^{*}}{\overline{H}},\qquad y=\frac{y^{*}}{\overline{H}},\qquad u=\frac{u^{*}\overline{H}}{\upsilon},\qquad v=\frac{v^{*}\overline{H}}{\upsilon},$$

$$p=\frac{p^{*}\overline{H}^{2}}{\rho\upsilon^{2}}\ \text{and}\qquad T=\frac{T^{*}-T_{C}^{*}}{\Delta T^{*}} \qquad (1)$$

Dimensionless continuity equation:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \qquad (2)$$

Dimensionless u-momentum equation:

$$\frac{\partial(u^{2})}{\partial x}+\frac{\partial(uv)}{\partial y}=-\frac{\partial p}{\partial x}+\left(\frac{\partial^{2}u}{\partial x^{2}}+\frac{\partial^{2}u}{\partial y^{2}}\right)+\frac{Ra}{\Pr}T\cos\phi \qquad (3)$$

Dimensionless v-momentum equation:

$$\frac{\partial(v^{2})}{\partial x}+\frac{\partial(uv)}{\partial y}=-\frac{\partial p}{\partial y}+\left(\frac{\partial^{2}v}{\partial x^{2}}+\frac{\partial^{2}v}{\partial y^{2}}\right)+\frac{Ra}{\Pr}T\sin\phi \qquad (4)$$

Dimensionless energy equation:

$$\frac{\partial(uT)}{\partial x}+\frac{\partial(vT)}{\partial y}=\frac{1}{\Pr}\left(\frac{\partial^{2}T}{\partial x^{2}}+\frac{\partial^{2}T}{\partial y^{2}}\right) \qquad (5)$$

Where, u and v are the dimensionless x-velocity and y-velocity components, p is the dimensionless pressure. In addition, the velocity and temperature boundary conditions, take the following form:
On all solid walls $u = 0$ and $v = 0$,

On the wavy surface (absorber), $T = T_H = 1$,
On the top surface (cover), $T = T_C = 0$,
On vertical boundaries, $\partial T / \partial x = 0$ .... (6)

Also, the dimensionless number Pr and Ra are Prandtl and Rayleigh numbers, are defined according to the following relations:

$$Pr = \frac{\upsilon}{\alpha} \quad \& \quad Ra = \frac{g \beta \Delta T \overline{H}^3}{\upsilon^2} Pr \qquad (7)$$

Where, β, α and υ are the coefficient of thermal expansion, thermal diffusivity, and kinematics viscosity of the fluid, respectively. To solve the dimensionless governing equations, one has to know the pressure field which is difficult to obtain.

To avoid solving the pressure field directly, one employ the widely used vorticity-stream function method [13]. The dimensionless stream function (ψ) and vorticity (ω) are defined as follows:-

$$u = \frac{\partial \psi}{\partial y} \quad \& \quad v = -\frac{\partial \psi}{\partial x} \qquad (8)$$

$$\omega = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \qquad (9)$$

Substituting in equations (2-4) with the corresponding expressions of velocity components (7-8), one can eliminate the pressure from both momentum equations.

Then the following transport governing equations for ψ and ω are obtained:

$$\frac{\partial^2 \psi}{\partial x^2} + \frac{\partial^2 \psi}{\partial y^2} = 0 \qquad (10)$$

$$\frac{\partial (u\omega)}{\partial x} + \frac{\partial (v\omega)}{\partial y} = \left(\frac{\partial^2 \omega}{\partial x^2} + \frac{\partial^2 \omega}{\partial y^2} \right) + Gr \left(\frac{\partial T}{\partial x} \cos\phi - \frac{\partial T}{\partial y} \sin\phi \right) \qquad (11)$$

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \frac{1}{\Pr} \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \qquad (12)$$

The average Nusselt number is the average of local Nusselt number along the wavy wall and is defined by the following equation:

$$Nu = \frac{1}{s} \int_0^s Nu_x \, ds \qquad (13)$$

Where, s is the total length along the wavy sinusoidal wall.

3. Numerical method

Governing equations are solved numerically using finite difference technique. According to this technique, partial differential equations which, describing the flow field are transformed to associated sets of linear algebraic equations. To transfer the final form of differential equations to difference equations, the derivatives of the problem are approximated by finite centered differences [13, 14]. The system of algebraic equations is solved using Gauss-Siedel iteration method to obtain the steady state solution of the problem considered. The calculations are done on finer grid size distribution. To ensure convergence, an under relaxation was needed for the non-linearity and strong coupling of the equations. The relaxation factors are used for stream function, vorticity and temperature equations and are, respectively, 0.9, 0.2 and 0.5. Variations by less than 10^{-6} over all grid points were adopted as a convergence criterion. A computer program in FORTRAN language is developed to perform the numerical solution according to the above-mentioned technique.

Numerical grid generation has now become a fairly common tool for use in the numerical solution of partial differential equations on arbitrarily shaped regions [15]. A curvilinear mesh is generated over the physical domain such that one member of each family of curvilinear coordinate lines is coincident with the boundary contour of the physical domain. The results were checked for grid independence. Grid sizes change from 420×30 to 420×120 according to the characteristic height ratio of the enclosure.

3.1. Grid independence study of the problem concerned

The grid independence test is performed using successively sized grids, 400×110, 420×120 and 440×130 for $Ra = 5\times10^5$, $L = 1$, $A = 4$ and $\phi = 60°$. The value of average Nusselt number at wavy wall is shown in Table 1. It is clearly seen that there is a little difference between the three grid results and the grid of 420×120 is used in all subsequent calculations considering the same channel characteristic height ratio. This procedure is repeated for all examined characteristic height ratio.

Table 1

Comparison of average Nusselt number at several grids for $Ra = 5\times10^5$, $L = 1$, $A = 4$ and $\phi = 60°$

	400×110	420×120	440×130
Nu	2.41	2.21	2.22

3.2. Code validation

The present code is validated for natural convection heat transfer by comparing the results of laminar heat transfer in a closed cavity with differentially heated walls. The bottom wavy wall was kept hot while the top wall was cooled. The left and right walls are insulated. In the present work numerical predictions using the developed algorithm, are obtained for Rayleigh numbers between 10^3 and 10^6. Table 2 compares the present results with those by de Gao et al. [7], Varol et al. [16] and Dalal et al. [8]. The results are in very good agreement with the benchmark solution. More over, a comparison is made between the present study and Claudio et al. [17] considering flat plate square enclosure. A good agreement is found, specially, at lower values of inclination angles.

Table 2 Comparison of solutions for natural convection in wavy wall and flat wall enclosures

	Parameters	Present study (numerical)	Gao et al. [7] (numerical)	Varol et al.[16] (CFD)	Dalal et al.[8] (numerical)	Cianfrini[17] (flat wall)
h	(a) $Ra = 5\times10^4$, $L = 1$, $A = 2$, $\phi = 30°$. (b) $Ra = 5\times10^4$, $L = 2$, $A = 2$, $\phi = 30°$.	4.26 4.17	4.10 4.00			
Nu	(a) $Ra = 3\times10^5$, $L = 4$, $A = 4$, $\phi = 40°$. (b) $Ra = 1\times10^6$, $L = 2$, $A = 3$, $\phi = 20°$.	14.07 13.36		13.8 12.9		
Nu	$Ra = 10^5$, $L = 2$, $A = 20$, $\phi = 90°$.	3.7 (4-undulations)			3.51 (3-undulations)	
Nu	(a) $Ra = 10^4$, $\phi = 20°$. (b) $Ra = 10^6$, $\phi = 30°$.	4.82 6.47				4.5 6.2

4. Results and discussions

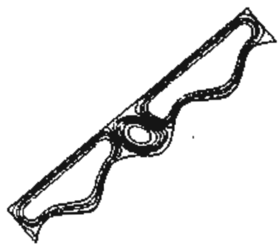
A numerical study is carried out to determine the influence of inclination angle (ϕ), Rayleigh numbers (Ra), characteristic height ratio (A) and

Characteristic geometric ratio (L) on the flow field and heat transfer in wavy enclosure. Inclination angle is considered in the range of 10–60° in steps of 10° to obtain the inclination. The Ra is varied between 5×10^3 and 10^6 to cover a large range. The influence of the

Streamlines are given at different values of Rayleigh numbers in Figure 2 for the case of A=4, L=2 and ϕ = 40°. Circulation cells form in each cavity of the wavy wall in counter clockwise and clockwise directions because hot fluid

moves up along the hot wall and it circulates to the left. It impinges on the top wall and moves downward. The fluid is stagnant between cells.

(a) Ra= 5×10^4



(b) Ra= 5×10^5

Fig. 2. Streamlines for different Rayleigh numbers at A=4, L=2 and ϕ = 40°

characteristic height ratio is examined for the values in between 1 and 5 considering different values of characteristic geometric ratio.

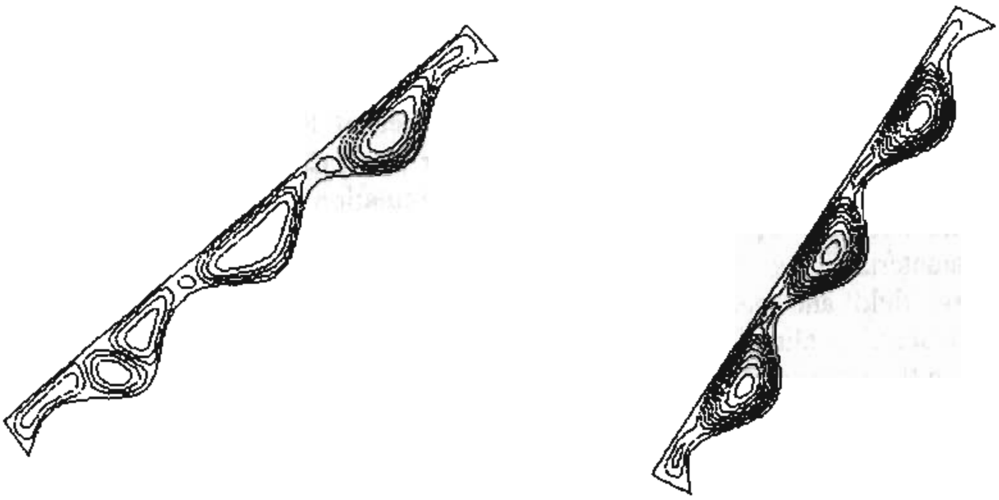
4.1 Flow field

Figure 3 shows the effect of inclination angles for the case of Ra = 10^5, L=2 and A=2. The flow is more complex for small inclination angles. As seen in Figure. 3a, seven cells are formed inside the enclosure and the biggest one is located at the top end of the cavity. The length of cells is increased with increasing inclination angle. Similar results were obtained by Mahmud et al. [2].

Another important parameter which affects the flow field is the characteristic height ratio since the distance between the hot and cold walls is

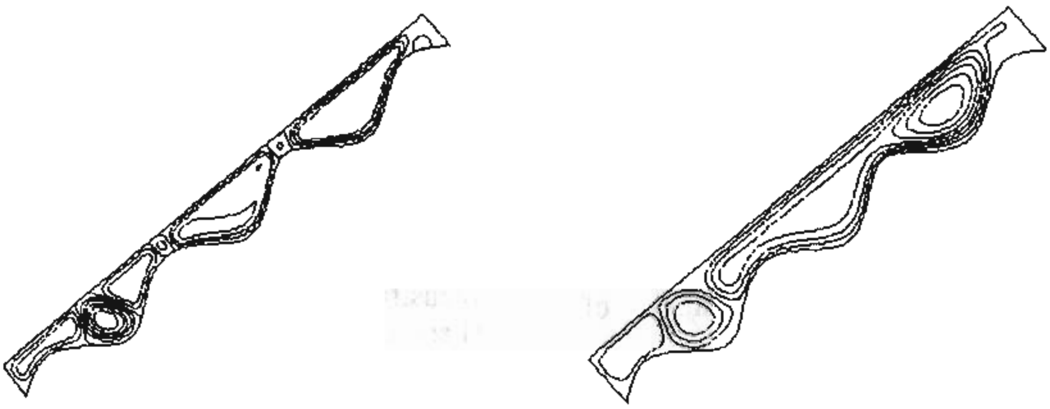
While Ra number is increased as indicated in Figure 2b, the flow is intensified at the middle of the enclosure. There are mainly two cells encompassing at the enclosure ends, complete domain with a small one at middle.

the most important parameter for heat transfer. The effect of characteristic height ratio for Ra = 10^6, L=2 and ϕ = 40° is shown in Figure. 4. Generally, streamlines strongly follow the boundaries except for the lowest part of the enclosure. At the right corner of the enclosure, the flow is almost stagnant due to the insulation boundary conditions. Considering lower values of characteristic height ratio, the flow is very complex and circulation cells are distorted because of the high velocity as shown in Figure 4a. Considering Figure. 4b, the length of circulation cells in each undulations along wavy wall are changed.



(a) $\phi = 40°$ (b) $\phi = 60°$

Figure. 3. Streamlines for different inclination angles at $Ra = 10^5$, L=2 and A=2.



(a) A=2 (b) A=3

Figure. 4. Streamlines for different characteristic height ratio at $Ra = 10^6$, L=2 and $\phi = 40°$.

4.2 Local Nusselt number

The local Nusselt number distribution, Nu_x, is plotted versus dimension x^* for different problem parameters as shown in Figure. 5 to Figure 8. Generally, local Nusselt number distribution shows rise and fall in each cycle and this tendency depends on the examined parameters. The presence of hill in the wavy wall, which leads to narrow flow passage, is the suggested reason for Nu_x rise. Figure. 5 shows the effects of inclination angle of the enclosure on the distribution of local Nusselt number at A = 2, L = 2 and Ra = 5×10^4. In general, as inclination angle is increased, the value of peak Nusselt number is decreased along the wavy wall. For the values of $\phi = 20°$ and $\phi = 40°$, the local Nusselt numbers follow the same pattern. The maximum Nusselt number is obtained for $\phi = 20°$ at the third hill (x^* = 0.9). Also, the maximum Nusselt number is obtained for $\phi = 40°$ near the second hill (x^* = 0.6). Figure 6 illustrates the distribution of local Nusselt number along the wavy wall at A = 4, L = 4 and Ra = 5×10^4 for different values of inclination angles. The local Nusselt number has peak values near hills position at $\phi = 20°$ and $\phi = 40°$. Comparing Figure 5 and Figure 6, for the same Rayleigh number, the local Nusselt number values at A =2 and L = 2 are higher than that the case of A = 4 and L = 4 due to increasing distance between hot and cold walls. In addition at

A =2 and L = 2, the number of peak values of Nu_x is larger than the case at A = 4 and L = 4. The flow reattachment is the suggest reason for these additional peak.

Figure 7 illustrates the distribution of Nu_x along the wavy wall at A = 2, ϕ = 20° and Ra = 10^5 for different values of characteristic geometric ratio. It is clear that at L = 1, small peak values of Nu_x exist along the wavy wall. On the other hand, large peak values of Nu_x exist at L = 2 and L = 3. In the case of L = 3, the local Nusselt number has a maximum value near the left vertical wall and it has four other peak values. Considering the present study, similar trend is observed at different inclination angles [8 and 16].

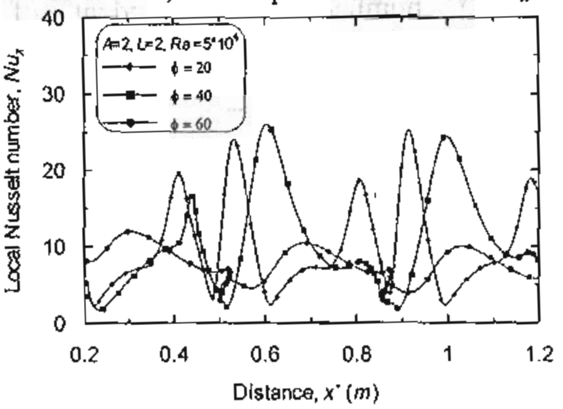


Figure. 5. Distribution of local Nusselt number along the wavy wall for different values of inclination angle at A=2, L=2.

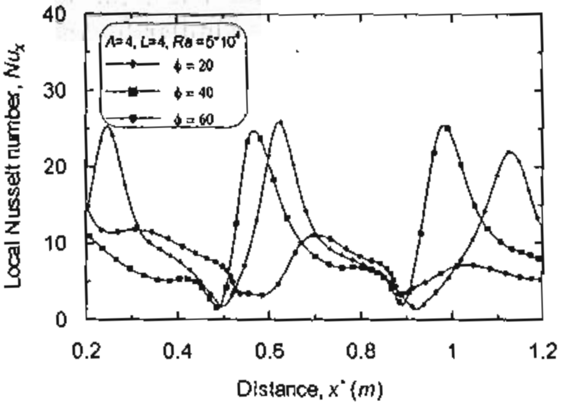


Figure. 6. Distribution of local Nusselt number along the wavy wall for different values of inclination angle at A=4, L=4.

As early indicated, the characteristic height ratio is another important parameter affecting the heat transfer and flow field inside the enclosure. Figure. 8 shows the variation of local Nusselt number along the wavy wall at L = 2, ϕ = 20° and Ra = 5×10^5 for different values of characteristic height ratio. The trend of local Nusselt number is almost the same for the examined values of A. The comparison between the local Nusselt numbers at the examined values of A showed that, Nu_x for A = 2 is higher than Nu_x for A = 3 and 4. The peak values of Nu_x is increased with the decrease of A. Examining the definition of A, it is decreased with the decrease in average height of the channel ($\overline{H}$). Which in turn leads to narrow passage at hill and consequently Nu_x is increased. Finally, the highest heat transfer is found at the middle section of the enclosure.

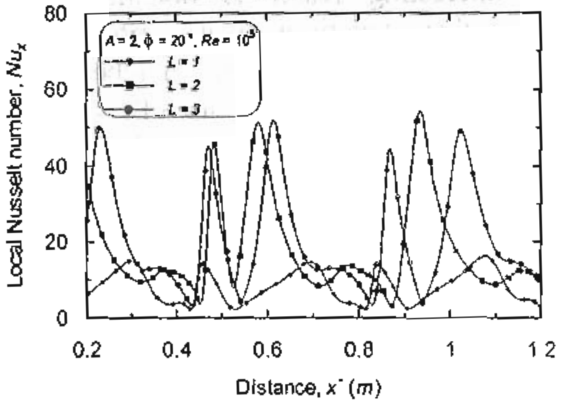


Figure. 7. Distribution of local Nusselt number along the wavy wall for different values of characteristic geometric ratio.

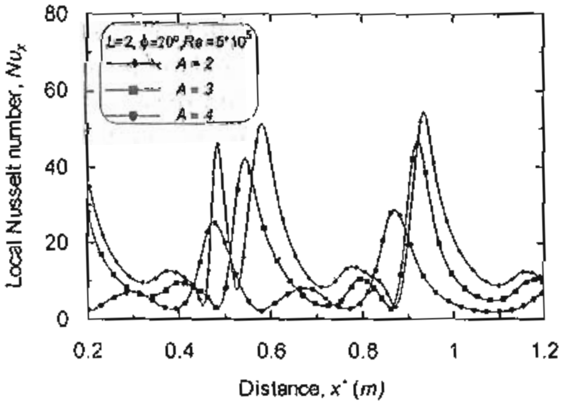


Figure. 8. Distribution of local Nusselt number along the wavy wall for different values of characteristic height ratio.

4.3 Average Nusselt number

To obtain the heat transfer performance, the average Nusselt number is present for different problem parameters as shown in Figure 9 to Figure 12.

Figure 9 illustrates the average Nusselt number, *Nu,* versus Rayleigh number at A = 2 and L = 1 for different values of inclination angle. The trend of average Nusselt number is similar for the examined inclination angle and is almost linear. The value of average Nusselt number is increased with increasing Rayleigh number as expected. Generally, the larger average Nusselt number is obtained for the smaller inclination angle. The higher values of *Nu* are occurred at ϕ = 20°. On the other hand, at ϕ = 10° lowest value of *Nu* is obtained [2, 8, 16]. It is clear that, at $Ra \leq 5\times10^3$, the average Nusselt number is independent of inclination angle.

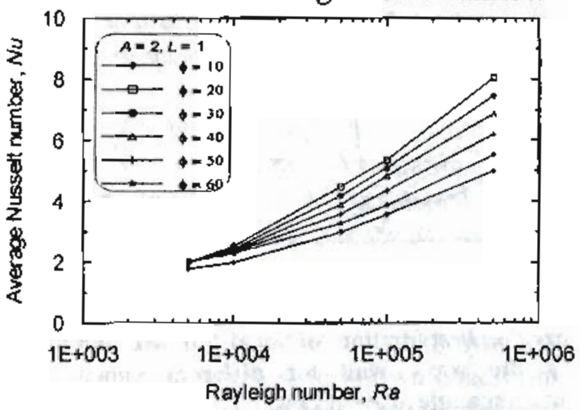


Figure. 9. Average Nusselt numbers versus characteristic geometric ratio at different values of inclination angle A=2, L=1.

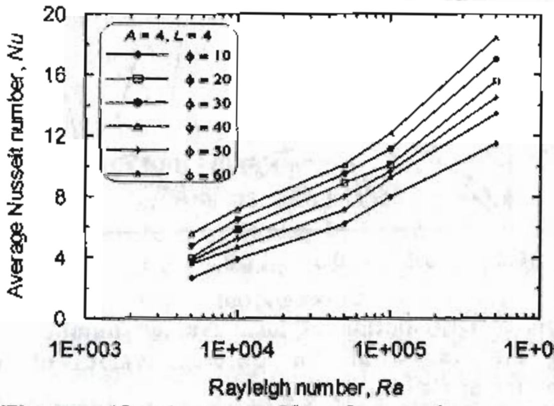


Figure. 10. Average Nusselt numbers versus characteristic geometric ratio at different values of inclination angle A=4, L=4.

In the same manner, Figure. 10 shows *Nu* versus *Ra* at A = 4 and L = 4 for different values of inclination angle. The same previous results are obtained, except that the higher values of *Nu* occurs at ϕ = 40°. Comparing Figure 9 and Figure. 10, considering the same Rayleigh number, the average Nusselt number values at A =4 and L = 4 are higher than that the case of A = 2 and L = 1. Decreasing of channel height and consequently the distance between hot and cold walls is the suggested reason for this increase. In present study, the highest average Nusselt number is obtained for L = 4 and A = 4 at the value of ϕ = 40°. It is concluded that, the inclination angle can be a control parameter for heat transfer.

Figure 11 shows the average Nusselt number versus characteristic geometric ratio, L, at A = 4 and Ra = 5×10^5 for different values of inclination angle. The value of average Nusselt number is increased, linearly, with increasing characteristic geometric ratio. An interesting result is obtained for higher characteristic geometric ratio. The highest average Nusselt number is achieved at ϕ = 40° in the case of $L > 2$ as illustrated in Figure. 11. On the other hand, this highest value of average Nusselt number is achieved at ϕ = 20° in the case of $L \leq 2$.

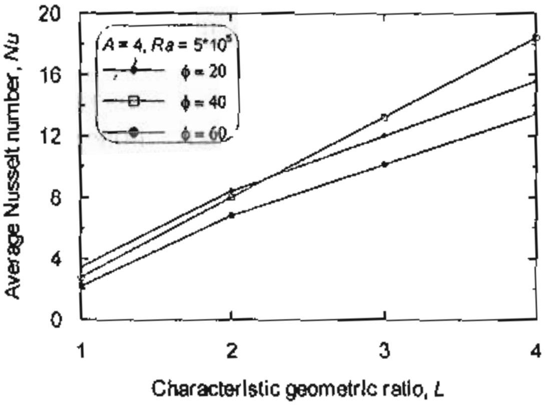


Figure. 11. Average Nusselt numbers versus characteristic geometric ratio at different values of inclination angle.

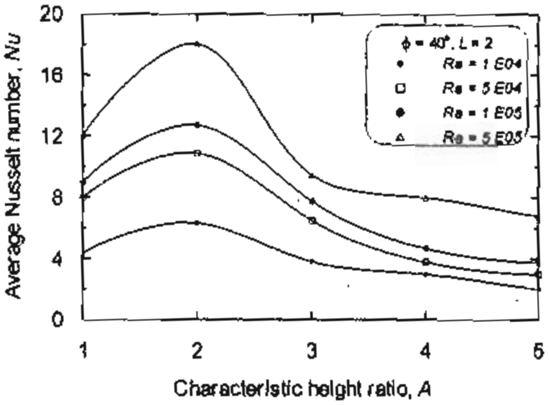


Figure. 12. Average Nusselt numbers versus characteristic height ratio at different values of Rayleigh number.

Finally, Figure. 12 show the average Nusselt number versus characteristic height ratio, A, at $L = 2$ and $\phi = 40^{\circ}$ for different values of Rayleigh number. It is found that Nu increases with A when $A < 2$ but decreases thereafter, and this decreasing rate is almost linear. From the figure, it is apparent that A should be equal 2 to minimize the heat loss through natural convection heat transfer inside the channel.

Conclusions

A numerical study is conducted to examine the steady state heat transfer by natural convection inside the channel between the flat-plate cover and the sine-wave absorber in a tilted wavy enclosure. The effects of system parameters on the natural convection heat transfer inside the tilted wavy enclosure are simulated. Rayleigh number (Ra), inclination angle (ϕ), characteristic height ratio (A) and characteristic geometric ratio (L) are considered as problem parameters with air as working fluid. Where, A is defined as the ratio of the channel height to the amplitude height of the sine wave absorber and L is defined as the ratio of one-fourth of the wavelength to the amplitude height of the absorber. The results indicated that, the flow field is affected by wavy wall. The more complicated flow is obtained for lower inclination angle. Heat transfer is increased with the increase of characteristic geometric ratio and Rayleigh number for the examined inclination angles. It is found that, to minimize the heat loss through natural convection heat transfer inside the channel between the cover and the absorber, A should be equal to 2, L larger than 1 and ϕ between 20° and 40°. The wavy wall can be used to control of heat transfer and flow field in addition to inclination angles.

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