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Abstract: Enclosures are frequently encountered in practice, and heat transfer through them is of practical interest. Heat transfer in enclosed space is complicated by the fact that fluid in the enclosure, in general, does not remain constant. The fluid adjacent to the hotter surface rises and the fluid adjacent to the cooler one falls, setting a rotionary motion within the enclosure that enhances the heat transfer through the enclosure. This paper describes a numerical predication of heat transfer and fluid flow characteristics inside an enclosure bounded by horizontal wavy walls and two periodic straight vertical walls. Governing equation were discretized using an implicit finite difference method, based on finite volume approach. Simulation was carried out for a range of Rayleigh number (10^4-10^6) and Aspect ratio (0.35-0.75) for the fluid having Prandtl number 0.71. Results are presented by streamlines, isotherms and local Nusselt numbers. It is observed that flow and thermal field inside the enclosure are affected by the shape of enclosure and heat transfer rate increases as Rayleigh number increase.

1. INTRODUTION

Buoyancy-driven cellular flow structure in a bottom heated horizontal wavy enclosures, noted as Rayleigh Benard convection, has been extensively investigated over the past four decades due to the important role it plays in fundamental fluid mechanics and heat transfer study in technological applications such as solar collectors, cooling of microelectronics equipments, heat transfer in nuclear reactors, electric machinery, double wall thermal insulation, natural circulation in atmosphere. Rayleigh-Benard convection is the instability of a fluid layer which is confined between two thermally conducting plates, and is heated from below to produce a fixed temperature difference. Mahmud and Islam[4] solved the laminar free convection and entropy generation inside an inclined wavy enclosure. They obtained that the inclination angle of cavity affects the entropy generation. Das and Mahmud[5]Analyzed the free convection inside both the bottom and ceiling wavy enclosure. They found the heat transfer rate rises when amplitude wave length ratio changes from zero to other values .Varol and Oztop [6]Investigated the effects of inclination angle on the laminar free convection. They observed that the inclination angle is most important and effective parameter. Mahmad et al.,[7] Have shown the effect of surface waviness on natural convection heat transfer and fluid flow inside a vertical wavy walled enclosed for range of wave ratio ($0.00 \leq \lambda \leq 0.4$) and aspect ratio ($1.0 \leq A \leq 2.0$) .They observed that aspect ratio is the most important parameter for heat and fluid transfer and higher heat transfer is obtained forced convection heat transfer.

2. PROBLEM SPECIFICATION

The schematic configuration of horizontal wavy enclosure of length L and height H are considered in a two different cases. First the hot bottom wavy wall and the cold straight top wall as shown in fig.1 and second the both the hot bottom and cold top walls are considered as wavy as shown in fig.2 .are kept isothermal at temperature equal respectively to T_h and T_c and the two vertical straight walls are considered periodical. The fluid is assumed to be of constant properties and the Boussinesq approximation is employed for the gravity terms. Stress-work are neglected. The dimension less temperature function is defined as $\theta = (T\text{-}T_r)/\Delta T$, in which $\Delta T = T_h\text{-}T_c$ stands for the temperature difference of two isothermal walls.

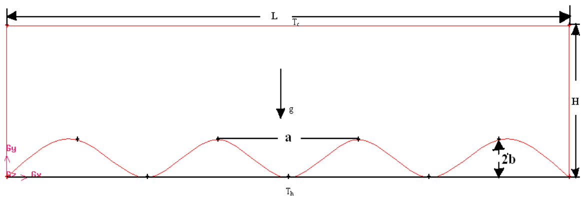

Fig.1.Schematic diagram of bottom wavy

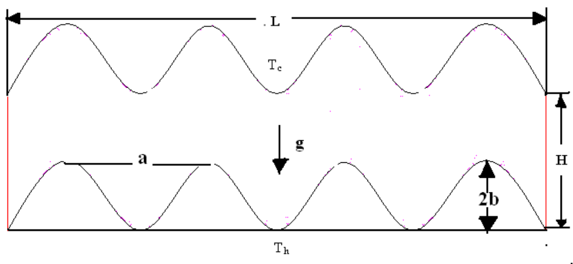

Fig.2. Schematic diagram of both top and bottom wavy

3. GOVERNING EQUATIONS AND BOUNDARY CONDITIONS

Natural convection is governed by the differential equations expressing the conservation of mass, momentum, and energy. The present flow is considered steady, laminar, incompressible and two-dimensional. The momentum equations are simplified using the Boussinesq approximation, in which all fluid properties are assumed constants except the density in its contribution to the buoyancy forces. The Resulting non-dimensional governing equations and the boundary conditions are:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0$$

$$\frac{\partial U}{\partial \tau}+U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=\Pr.(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2})-\frac{\partial P}{\partial X}$$

$$\frac{\partial V}{\partial \tau}+U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=\Pr.(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2})-\frac{\partial P}{\partial Y}+Ra.\Pr.\theta$$

$$\frac{\partial \theta}{\partial \tau}+U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2})$$

Where A = (2.b/a)υ is the aspect ratio of the enclosure. Ra = $g\beta L^3(T_h-T_c)/\upsilon\alpha$ is the Rayleigh number. Pr = υ/α is the prandtl number. β is the thermal expansion coefficient and υ is the kinematic viscosity. The two opposite wall (bottom and top) are kept at the uniform temperature T_h and T_c respectively. The boundary conditions for the system of equation are,

U = V/ X = θ/ X =0; at two side walls.

U=V=θ =1; at hot wavy bottom wall.

U=V= θ = 0; at cold straight top wall

Heat transfer rate is measured by local (NuX) and average (Nu) Nusselt numbers. Following equations are used to calculate the local and average nusselt numbers:

NuX = - — $_w$

Nu =

Grid independence study and validation:

A grid independence test was performed using three combination of control volumes (30 113,74 289,82 321) for Ra=5 10^5. To check the validity of the present numerical procedure, thermally driven rectangular cavity were solved. For validation, the numerical resultants are compared with the resultants of Soong et al (1996) solutions.Comparsions lies in Fig.3&4. present predication shows a very good agreement with the result of Soong et al. (1996).

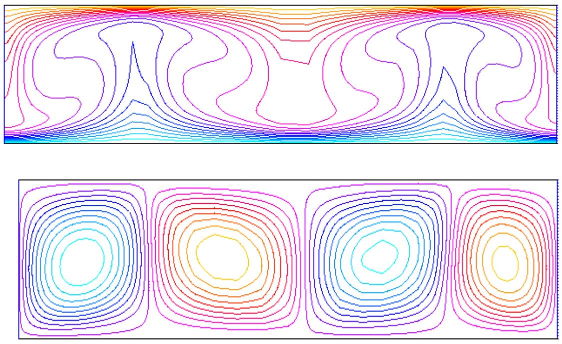

Fig.3. Isotherms (top) and streamlines (bottom) for A=4, As=0.5, Ra=5 10^5, and Pr=0.71

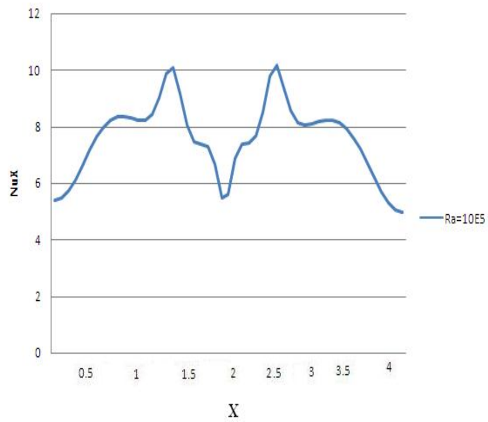

Fig.4 Validation of Nusselt number for Rectangular cavity at A=4, Ra=5 10^5 and Pr=0.71.

4. RESULTS AND DISCUSSION

A numerical study is made to stimulate of natural convection flow and thermal fields inside the horizontal wavy enclosures. A comparison is performed from the heat transfer point of view. Parameters effective on natural convection are non-dimensional Aspect ratio of 0.5, the Rayleigh number which changes between 5 10^4 to 5 10^6 and the non-dimensional length chosen to be 4. Pr number is taken as 0.71 which corresponds to air.

Two symmetric counter rotating vortices are observed at top and bottom of the wave wall on the flow field due to the uniform temperature graident. At low rayleigh number circulation inside the cavity is weak. Isothermal lines are nearly parallel to each other and follow the geometry of the surfaces. The effects of Rayleigh number are presented in fig.(5-8) for the two different wavycases.

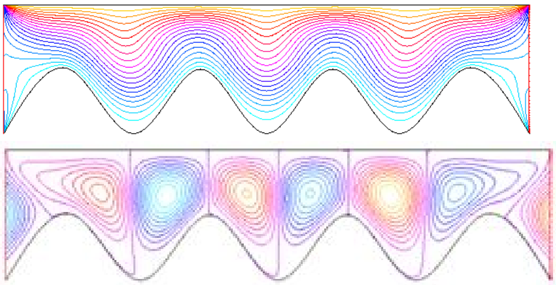

Fig.5. Isotherms (top) and streamlines (bottom) for A=4, As=0.5, Ra=5 10^5, and Pr=0.71

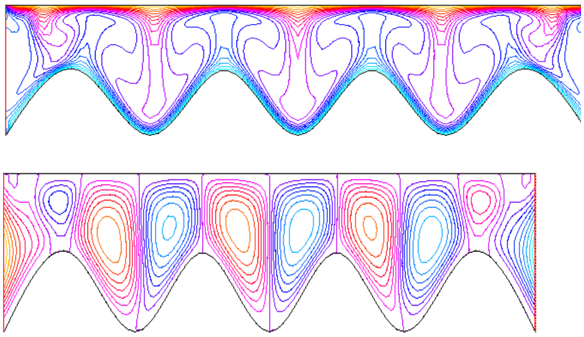

Fig.6. Isotherms (top) and streamlines (bottom) for A=4, As=0.5, Ra=5 10^6, and Pr=0.71

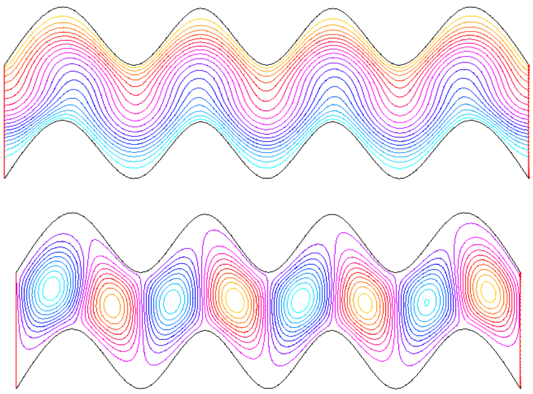

Fig.7. Isotherms (top) and Streamlines (bottom) for A=4, As=0.5, Ra=5 10^5 and Pr=0.71.

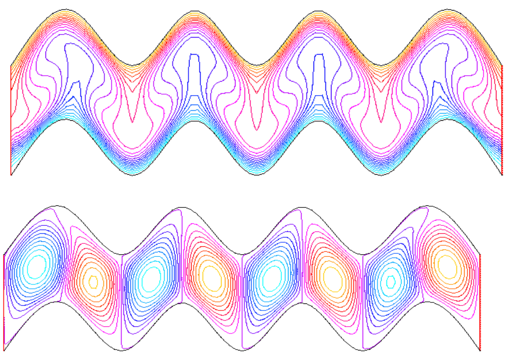

Fig.8. Isotherms (top) and streamlines (bottom) for A=4, As=0.5, Ra=5 10^6, and Pr=0.71

As the results the comparison from two different wavy cases it is seen that a further increase of Rayleigh number increase the circulation strength inside the enclosure. The uniform temperature profile is changed and three high gradient spot is observed at top and bottom of the hot wall due to the rapid circulation of fluid inside the cavity.

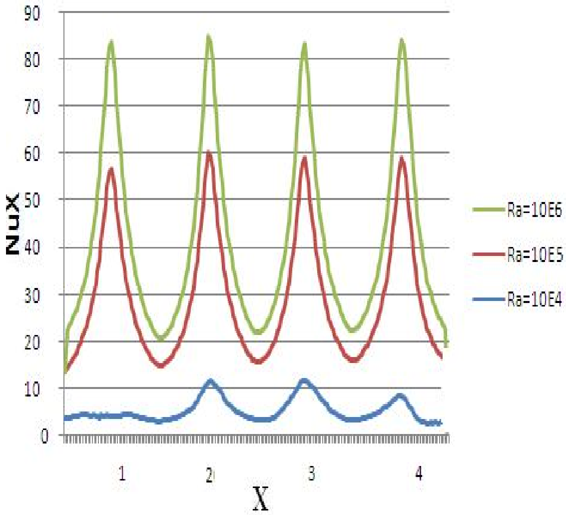

Fig.9. Variation of local nusselt number for bottom wavy at As=0.5, A=4, Pr=0.71.

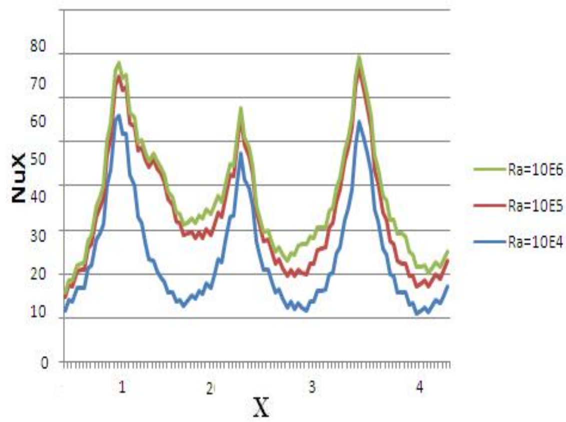

Fig.10. Variation of local nusselt number for Both top and bottom wavy at As=0.5, A=4, Pr=0.71.

Fig.9&10 compares the local nusselt number along the two different cases. First the hot bottom wavy wall and the cold straight top wall and second the both the hot bottom and cold top walls are considered as wavy with different Rayleigh number .As expected, their values increase with the increase of the Rayleigh number. The same increase can be seen on the value of local nusselt number for wavy wall but their distribution show wavy behavior. They have maximum value on the top of wave. But in the cavity part of wave , they decrease and a minimum point can be seen the value of which is smaller due to the low velocity in that part.

5. CONCLUSION

Laminar steady natural convection heat transfer and the fluid flow in a horizontal wavy enclosure are investigated numerically. Some observations can be drawn as follows:

- Natural convection heat transfer and fluid flow is strongly affected by the geometrical parameters and Rayleigh number.
- Two circulation cells are obtained at the enclosure cavity of wave in different directions especially at the high Rayleigh number.
- Variation of local Nusselt number is higher for top and bottom wavy than the rectangular and bottom wavy enclosure.

NOMENCLATURE

A	non dimensional length, L/H
L	length of the enclosure (m)
H	height of the enclosure (m)
As	aspect ratio, 2b/a
b	amplitude height (m)
a	wave length of the wavy wall(m)
g	gravitional acceleration, (m sec^{-2})
Nu	mean nusselt number
NuX	local nusselt number
Pr	Prandtl number
Ra	Rayleigh number
p,P	dimensional pressure and pressure (N m^{-2})
u,v	dimensional velocity components (m sec^{-1})
U,V	dimensionless velocity components
x,y	Cartesian coordinates (m)
X,Y	dimensionless coordinates.

Greek symbols

α	Thermal diffusivity (m^2 sec)
β	Thermal expansion coefficient (K^{-1})
θ	Dimensionless temperature function $\theta = (T-T_r)/(T_h-T_c)$
λ	Thermal conductivity (w.m^1.K^1)
Ψ	Dimensionless stream function

Subscripts

h	Hot
c	Cold

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