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Investigation of turbulence mixed convection in air filled enclosures

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Mixed convection heat transfer is a process where natural and forced convection happens simultaneously. The effect of temperature difference in natural convection and momentum difference results in mechanical ventilation or pressure difference that result in indoor and outdoor flow. When used in an optimum way, it is one of the newly subjects in HVAC industry which is a prominent example of mixed convection heat transfer. In the present study, first, the laminar flow of natural convection of air in a room with forced convection results in mechanical ventilation and then the calculated results are compared with the results of other researchers. After showing validation of calculations, aforementioned flow is solved as a turbulent flow, using the valid turbulence models RNG $K-\epsilon$, standard $K-\epsilon$ and RSM. To solve governing differential equations for this kind of flow, method of finite volume was used. This method is a special kind of residual weighted method. The results show that turbulence intensity is limited to vertical walls and boundary condition, such that flow in center region enclosure is turbulence and laminar is close to walls. Moreover, it is discovered that when Richardson number increases the maximum local Nusselt decreases; and therefore heat transfer is decreased with increasing Ri number.

Key words: Mixed convection, turbulence, Richardson number, turbulent intensity, wall shear stress, enclosures.

INTRODUCTION

The heat transfer phenomenon, in which both natural and forced convections simultaneously exist, is called mixed convection. Mixed convection heat transfer occurs when its buoyant flow matters in a forced flow or when the effect of forced flow matters in a buoyant flow. Prevailing dimensionless numbers which are used to determine this type of flow are as follows: Grashof number (Gr), Reynolds number (Re), Rayleigh number (Ra), Prandtl number (Pr) and Richardson number (Ri). Richardson number is calculated by dividing natural convection effect by forced convection effect, and expressed by: $Ri=Gr/Re^2$ and from a physical standpoint, the Richardson number is the ratio of the convection to the forced convection.

In limit case where $Ri \rightarrow 0$ or $Ri \rightarrow \infty$, dominant heat

transfers are forced and natural convections, respectively. Mixed convection plays an important role in heat transfer from fan coils or package radiators into the room air, and in heat transfer from cooling systems into the air. Atmospheric and oceanic movements and the relevant heat transfer processes important in environmental sciences also involve mixed convection. Mixed convection is also employed for cooling modern electric and electronic systems (transistors, computers, transformers). In manufacturing and double-layer ceilings, mixed convection phenomenon can be used to calculate thermal loss.

There are different methods for creating mixed convection. One way includes the entry of hot (or cold) fluid from one side, passing isothermal walls, and exiting from the other side. In this case we could evaluate and compare the forced convection effect caused by the entry and exit of the fluid. Some scientists have applied thermal flux on the way fluid passes through the channel and also

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studied the effects. Among the studies, we can mention the ones done by Rahman et al. (2007), Saha et al. (2006), and Saha et al. (2008). Another method for creating mixed convections is to move enclosure walls in the presence of hot (cold) fluid inside the enclosure. This creates shear stresses and provides thermal and hydrodynamic boundary layers in the fluid inside the enclosure, and eventually creates forced convection flows in it. Numerous studies have been conducted in this field so far. For example, Oztop and Dagtekin (2004) studied a two-dimensional and square-shaped enclosure with vertical isothermal moving walls and insulated horizontal walls. In this work, different situations have been considered concerning the movement of vertical walls, and $0.01 \leq Ri \leq 100$ has been presupposed. Heat transfer rate has also been expressed in the form of Nusselt numbers. The results of this work suggest that in low Richardson values, if the moving walls move in the opposite directions, heat transfer from enclosure is much higher than when the walls slide converge. Basak et al. (2009) studied the mixed convection flow inside a square enclosure with left and right cold walls, insulated moving upper wall, and fixed lower hot wall by using finite element method. They suggested that by increasing Gr, with Pr and Re fixed, recirculation power will improve. In 2007, Sharif studied the laminar mixed convection in inclined rectangular enclosures with aspect ratio of 10, by using Fluent6. He let the Rayleigh number variable to be between 105 to 107, and Reynolds number fixed, to be 408.21. The fluid he used was water with Prandtl as 6, and the enclosure inclination angle to the horizon varied between 0 to 30°. The mentioned enclosure had hot moving upper wall, cold fixed lower wall, and adiabatic left and right walls. His study showed that local Nusselt number heightened by increasing the enclosure inclination angle.

Although progression of different sciences in the last decade has provided much subtle laboratory measuring tools and application of modern methods like parallel processing has enabled us efficiently use numerical analysis methods, analysis of turbulent flows inside the enclosure is still a challenging topic in fluid mechanics. This is because it is too difficult to reach ideal adiabatic wall condition in experimental situation. It is also so difficult to measure low speeds in mixed convection process using present sensors and probes. Although numerical methods like DES, LES, and DNS have seen much progression, it is still hardly possible to predict the stratification in the core of the enclosure. Non-linearity and coupling of the governing equations have also made the calculations complicated and time consuming. That is while in designing large enclosures, Rayleigh number is usually large, and so the flow nature is turbulent. The complexity of calculations in mixed convection situation has made scientists study the flow only in natural convection, among which we can mention the studies of Salat et al. (2004), Aounallah et al. (2007), Bessaih and

Kadja (2000), Ampofo (2004), and Xaman (2005). Tian and Karayiannis (2000) started an experimental study in South Bank University which was followed by Ampofo and Karayiannis in 2003. The data in this work were experimental benchmark data of natural convection flow inside a square enclosure, and were used for other studies. Whereas Peng and Davinson (2001) studied the mentioned flow by using LES, and Omri and Galanis (2007) used the SST k- ω to study the flow. Ampofo (2005) expanded his previous study in 2005 and tested the same enclosure with partitions and horizontal conducting walls and compared the results with his previous findings. Hsieh and Lien (2004) used turbulence models of steady RANS like Low-Re k- ϵ , and numerically analyzed the works done by Tian and Karayiannis (2000) and Betts and Bokhari (2000). In the present work, turbulent natural convection flow inside the enclosure is modeled first, and the results have been compared with the studies of Tian and Karayiannis (2000), Ampofo and Karayiannis (2003), Peng and Davinson (2001), Hsieh and Lien (2004), Omri and Galanis (2007). After the calculations are made valid, the Laminar mixed convection flow is solved in the square enclosure first, in comparison to Sharif (2007) and eventually turbulence mixed convection flow is modeled for the first time, using turbulence models like standard k- ϵ , RNG k- ϵ and RSM.

PROBLEM FORMULATION

For modeling the flow, continuity, momentum, energy, and turbulence equations have been studied. The properties have been considered as fixed. Density is calculated vertically by using variable density parameter for $\Delta T > 30$ and Boussinesq approximation for $\Delta T < 30$. The governing equations are as follow:

Continuity equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (1)$$

Momentum equations in X and Y direction:

$$\begin{aligned} \frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \\ \frac{\partial}{\partial x} (v + v_t) \left(2 \frac{\partial u}{\partial x} \right) + \frac{\partial}{\partial y} (v + v_t) \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right) \end{aligned} \quad (2)$$

$$\begin{aligned} \frac{\partial v}{\partial t} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial y} + g\beta(T - T_m) + \\ \frac{\partial}{\partial y} (v + v_t) \left(2 \frac{\partial v}{\partial y} \right) + \frac{\partial}{\partial x} (v + v_t) \left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y} \right) \end{aligned} \quad (3)$$

Energy equation:

$$\begin{aligned} \frac{\partial T}{\partial t} + u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \\ \frac{\partial}{\partial x} \left(\frac{v}{Pr} + \frac{v_t}{\sigma_T} \right) \frac{\partial T}{\partial x} + \frac{\partial}{\partial y} \left(\frac{v}{Pr} + \frac{v_t}{\sigma_T} \right) \frac{\partial T}{\partial y} \end{aligned} \quad (4)$$

Now for k-ε model, we will have:

Turbulent kinetic energy transport equation for k-ε model:

$$\begin{aligned} \frac{\partial k}{\partial t} + u \frac{\partial k}{\partial x} + v \frac{\partial k}{\partial y} = \frac{\partial}{\partial x} \left(v + \frac{v_t}{\sigma_k} \right) \frac{\partial k}{\partial x} \\ + \frac{\partial}{\partial y} \left(v + \frac{v_t}{\sigma_k} \right) \frac{\partial k}{\partial y} + P_k + G_k - \varepsilon \end{aligned} \quad (5)$$

Dissipation of turbulent kinetic energy transport equation for k-ε model:

$$\begin{aligned} \frac{\partial \varepsilon}{\partial t} + u \frac{\partial \varepsilon}{\partial x} + v \frac{\partial \varepsilon}{\partial y} = \frac{\partial}{\partial x} \left(v + \frac{v_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x} \\ + \frac{\partial}{\partial y} \left(v + \frac{v_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial y} + C_1 \frac{\varepsilon}{k} P_k + \\ C_2 \frac{\varepsilon^2}{k} + C_3 \frac{\varepsilon}{k} G_k - R_\varepsilon \end{aligned} \quad (6)$$

The eddy viscosity obtained from Prandtl- Kolomogorov relation:

$$v_t = C_\mu f_\mu \frac{k^2}{\varepsilon} \quad (7)$$

The stress production term, P_k , is modeled by:

$$P_k = v_t \left[2 \left(\frac{\partial u}{\partial x} \right)^2 + 2 \left(\frac{\partial v}{\partial x} \right)^2 + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)^2 \right] \quad (8)$$

The buoyancy term, G_k , is defined by:

$$G_k = -g\beta \frac{v_t}{\sigma_T} \frac{\partial T}{\partial y} \quad (9)$$

We will also have the following for RNG k-ε:

$$R_\varepsilon = \frac{C_\mu \rho \eta^3 \left(1 - \frac{\eta}{\eta_0} \right) \varepsilon^2}{1 + \beta \eta^3} \frac{1}{k} \quad (10)$$

That:

$$\eta = \frac{Sk}{\varepsilon} \quad (11)$$

The main difference between standard k-ε and RNG k-ε methods is in the ε equation, such that we can say the model RNG k-ε is the very same standard k-ε model, whose analytical formulas for turbulent Prandtl numbers have been improved. This is while these values in standard k-ε model are gained experimentally.

For RSM model, the turbulence equations are as follows: Reynolds stress transport equations:

$$\frac{D}{Dt} \left(\overline{u'_i u'_j} \right) = \frac{\partial (d_{ijk})}{\partial x_k} + P_{ij} + G_{ij} + \phi_{ij} - \varepsilon_{ij} \quad (12)$$

That:

$$\begin{aligned} \frac{D}{Dt} \left(\overline{u'_i u'_j} \right) &= \frac{\partial (\overline{u'_i u'_j})}{\partial t} + \overline{u_k} \frac{\partial (\overline{u'_i u'_j})}{\partial x_k} : \text{Advection (By Mean Flow)} \\ P_{ij} &= - \left[\left(\overline{u'_i u'_k} \right) \frac{\partial \overline{u'_j}}{\partial x_k} + \left(\overline{u'_j u'_k} \right) \frac{\partial \overline{u'_i}}{\partial x_k} \right] : \text{Production (By Mean Strain)} \\ G_{ij} &= \overline{u'_i f'_j + u'_j f'_i} : \text{Production (By Body Force)} \\ \phi_{ij} &= \frac{P'}{\rho} \left(\frac{\partial u'_i}{\partial x_j} + \frac{\partial u'_j}{\partial x_i} \right) = \frac{2P'}{\rho} S_{ij} : \text{Pressure-Strain Correlation} \\ \varepsilon_{ij} &= 2\nu \frac{\partial u'_i}{\partial x_k} \frac{\partial u'_j}{\partial x_k} : \text{Dissipation} \\ d_{ijk} &= \nu \frac{\partial (\overline{u'_i u'_j})}{\partial x_k} - \\ &\frac{P'}{\rho} \left(\overline{u'_i \delta_{jk} + u'_j \delta_{ik}} \right) - \overline{u'_i u'_j u'_k} : \text{Diffusion} \end{aligned}$$

Turbulent kinetic energy transport equation for RSM model:

$$\frac{Dk}{Dt} = \frac{\partial d_i^{(k)}}{\partial x_i} + P^{(k)} + G^{(k)} - \varepsilon \quad (13)$$

That:

$$\begin{aligned} P^{(k)} &= -\overline{u'_i u'_j} \frac{\partial \overline{u'_i}}{\partial x_j} : \text{Production (By Mean Strain)} \\ G^{(k)} &= \overline{u'_i f'_i} : \text{Production (By Body Force)} \\ \varepsilon &= \nu \left(\frac{\partial u'_i}{\partial x_j} \right)^2 : \text{Dissipation} \\ d_i^{(k)} &= \nu \frac{\partial k}{\partial x_i} - \frac{P' u'_i}{\rho} - \frac{1}{2} \overline{u'_j u'_j u'_i} : \text{Diffusion} \end{aligned}$$

Table 1. Coefficients for RNG k-ε turbulent model (Rahman et al., 2007).

Coefficient	Value
C_μ	0.0845
σ_k	1
σ_ε	1.3
C_1	1.42
C_2	1.68
η_0	4.38
β	0.012
K	0.41

Table 2. Coefficients for Standard k-ε turbulent model (Rahman et al., 2007).

Coefficient	Value
C_μ	0.0845
σ_k	1
σ_ε	1.3
C_1	1.42
C_2	1.68

Table 3. Coefficients for RSM turbulent model (Rahman et al., 2007).

Coefficient	Value
$c_2^{(w)}$	0.3
$c_1^{(w)}$	0.5
C_s	0.22
C_1	1.8
C_2	0.6
C_3	2.5

Except the terms Convection and Production in Reynolds stress transport equation, all the other terms have contributed in introducing a series of correlations, which have to be identified according to some known and unknown quantities, so that the equation system can be configured.

Diffusion term:

$$-\overline{u'_i u'_j u'_k} = C_s \frac{k}{\varepsilon} \overline{u'_k u'_i} \frac{\partial \overline{u'_i u'_j}}{\partial x_i} \quad (14)$$

Redistribution term:

$$\phi_{ij} = \phi_{ij}^{(1)} + \phi_{ij}^{(2)} + \phi_{ij}^{(w)} \quad (15)$$

That:

$$\phi_{ij}^{(1)} = -C_1 \frac{\varepsilon}{k} \left(\overline{u'_i u'_j} - \frac{2}{3} k \delta_{ij} \right)$$

$$\phi_{ij}^{(2)} = -C_2 \left(P_{ij} - \frac{1}{3} P_{kk} \delta_{ij} \right)$$

$$\phi_{ij}^{(w)} = \left(\tilde{\phi}_{ik} n_k n_j \delta_{ij} - \frac{3}{2} \tilde{\phi}_{ik} n_j n_k - \frac{3}{2} \tilde{\phi}_{jk} n_i n_k \right) \psi$$

$$\tilde{\phi}_{ij} = -C_1^{(w)} \frac{\varepsilon}{k} \overline{u'_i u'_j} + C_2^{(w)} \phi_{ij}^{(2)}$$

$$\psi = \frac{k^{\frac{3}{2}}}{C_3 y_n \varepsilon}$$

y_n is the distance from the wall. The role of the terms $\Phi_{ij}^{(2)}$, $\Phi_{ij}^{(1)}$ is to return isotropy (or terminating anisotropic flow with distributing kinetic energy of Reynolds' huge stresses among the stresses of smaller size). The terms $\Phi_{ij}^{(1)}$ and $\Phi_{ij}^{(2)}$ are called "return to isotropy" and "isotropization of production", respectively. The term $\Phi_{ij}^{(w)}$ is named as "wall reflection term".

For Dissipation term, we have:

$$\varepsilon_{ij} = \frac{2}{3} \delta_{ij} \varepsilon \quad (16)$$

The constants in the above relations have been presented in Table 1 for RNG k-ε, Table 2 for standard k-ε, and in Table 3 for RSM models.

In order to solve the differential equation that governs the flow, the finite volume method, which is explained in detail by Patankar (1980) was used. This method is a specific case among the residual of weighting methods. In this approach, the computational field is divided into some control volumes in a way that each node is surrounded by a control volume, and that control volumes have no volumes in common. The differential equation is then integrated on each control volume. Profiles in pieces which show changes (of a certain quantity like temperature, velocity, etc.) among the nodes, were used to calculate the integrals. The result is discretization equation, which includes quantities for a group of nodes (Patankar, 1980).

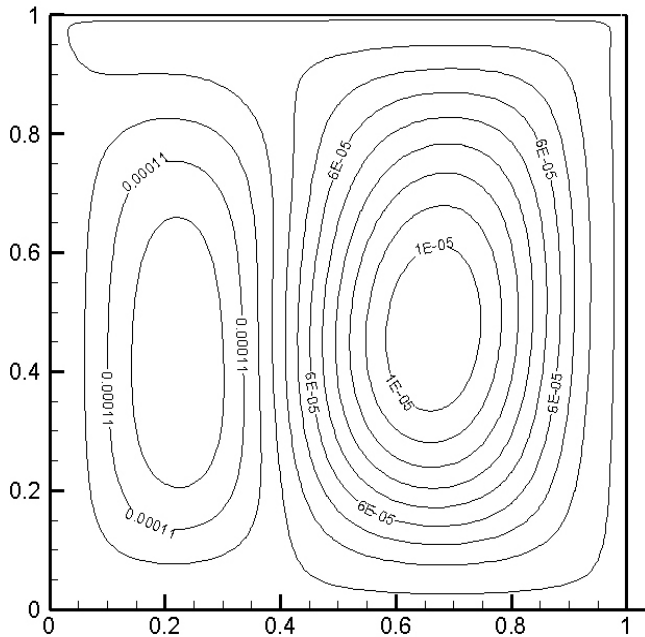
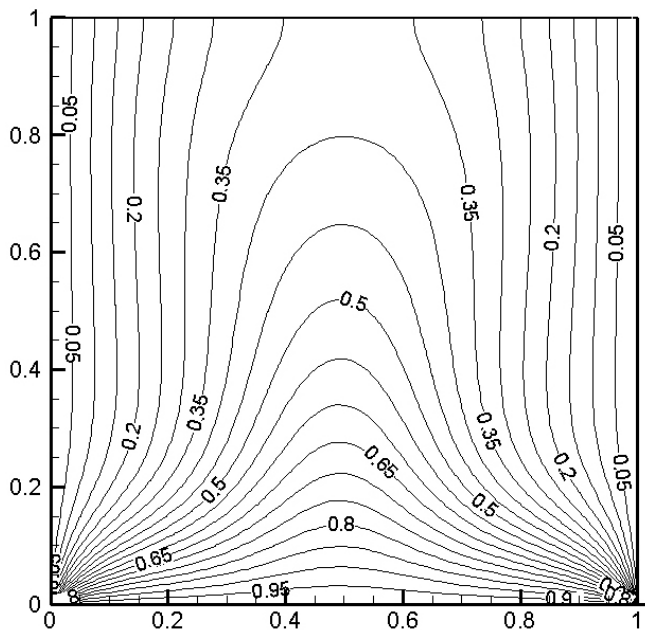
RESULTS

Grid independency

Grids designed to cover control volumes are square meshes provided on physical domain with different distances in order to reach independence. The mentioned mesh-independence for each turbulence

Table 4. Some meshes used for solving the problem.

	RNG k- ϵ			Standard k- ϵ			RSM		
Ri=0.1	57×57	101×101	161×161	61×61	119×119	181×181	75×75	119×119	201×201
Ri=1	68×68	135×135	201×201	81×81	160×160	241×241	90×90	160×160	270×270
Ri=10	75×75	150×150	225×225	90×90	175×175	270×270	105×105	180×180	300×300

**Figure 1.** Stream function with Re=10, in comparison with Basak et al. (2009) experiment.**Figure 2.** Temperature Contour with Re=1, in comparison with Basak et al. (2009) experiment.

model and any different Ri has been separately calculated. Table 4 shows some meshes used in this study.

Evaluating laminar mixed convection flow inside a square enclosure

In order to compare this work with the results of Basak et al. (2009), Pr and Gr have been considered as 0.7 and 10^4 , respectively. $1 < Re < 100$ has also been changed. Figures 1 and 2 present stream function and temperature contour in comparison with Basak et al. (2009). The acceptable harmony among the figures validates the accuracy of this study (Sharif, 2007)

Evaluation of turbulent mixed convection flow inside the enclosure

In order to prove the accuracy of the calculations, firstly, natural convection flow is solved into rectangular and square enclosures which are studied by other scientists like Tian and Karayiannis (2000), Ampofo and Karayiannis (2003), Peng and Davinson, (2001) and Omri and Galanis (2007), and then after the results are validated (Figures 3 and 4), the flow is solved inside the enclosure of mixed convection. In this case Ri varies from 0.1 to 10.

Figure 5 shows the schematics of the problem and also details of the Rectangular and Square Enclosures have been shown in Table 5.

Figure 6 illustrates the turbulent intensity for $y/H=0.5$ in a different Ri and turbulent models. It can be seen that turbulence intensity is limited to right and left walls and it reached a peak at the centre of cavity.

Figure 7 shows the Nusselt number counter on hot wall. As a result of this figure, it is clear that the Nusselt number is maximum for the case of Ri=0.1 and it moderately decreases due to increase in the value of Ri. It means that Ri=0.1, the forced convection governs fluid treatment. So the rate of heat transfer from cavity increased as a result of less Richardson number. Moreover, the rate of heat transfer by natural convection will grow if the Richardson number will increase. In large Richardson numbers, the natural convection is a major parameter of heat transfer in a cavity. The rate of heat transfer by mix and forced convection is much more than natural convection.

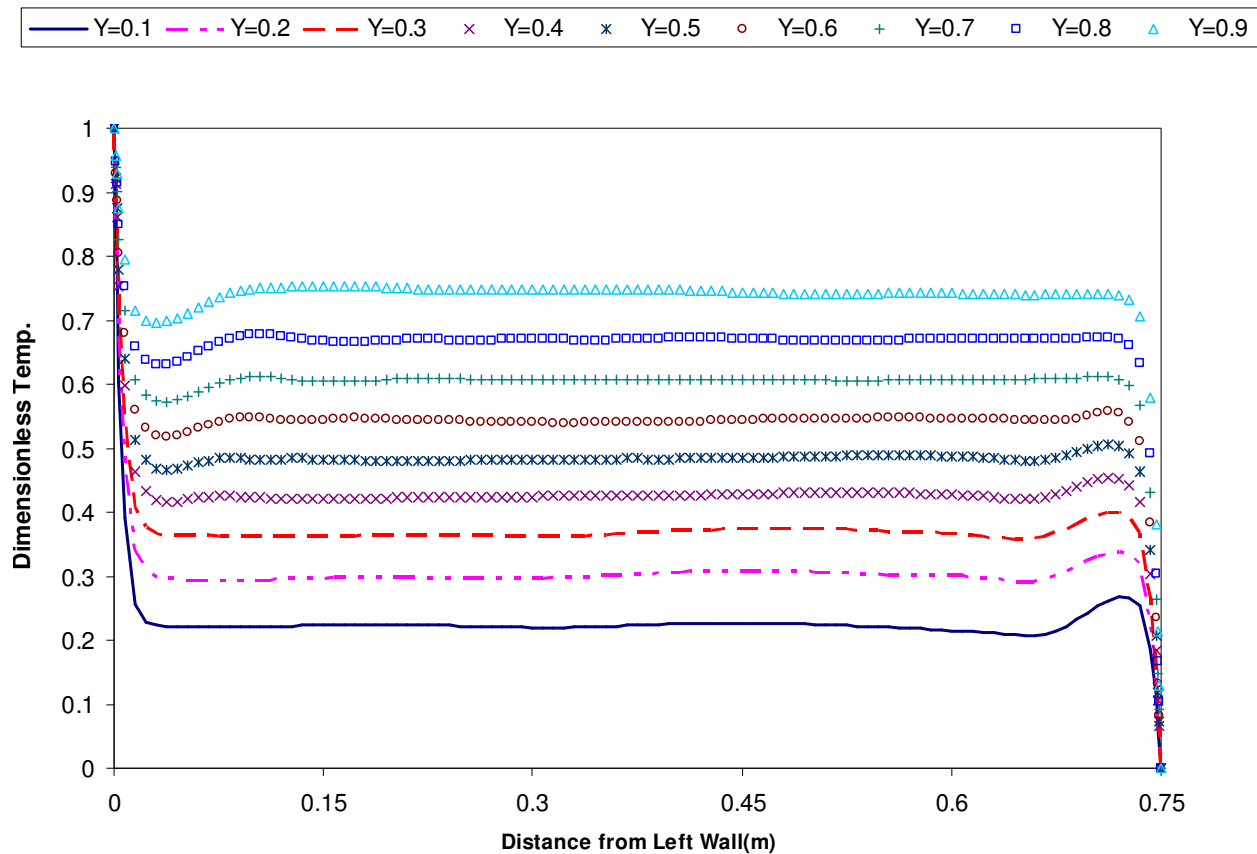


Figure 3. Temperature distribution at different height, in comparison with Tian and Karayiannis (2000) experiment.

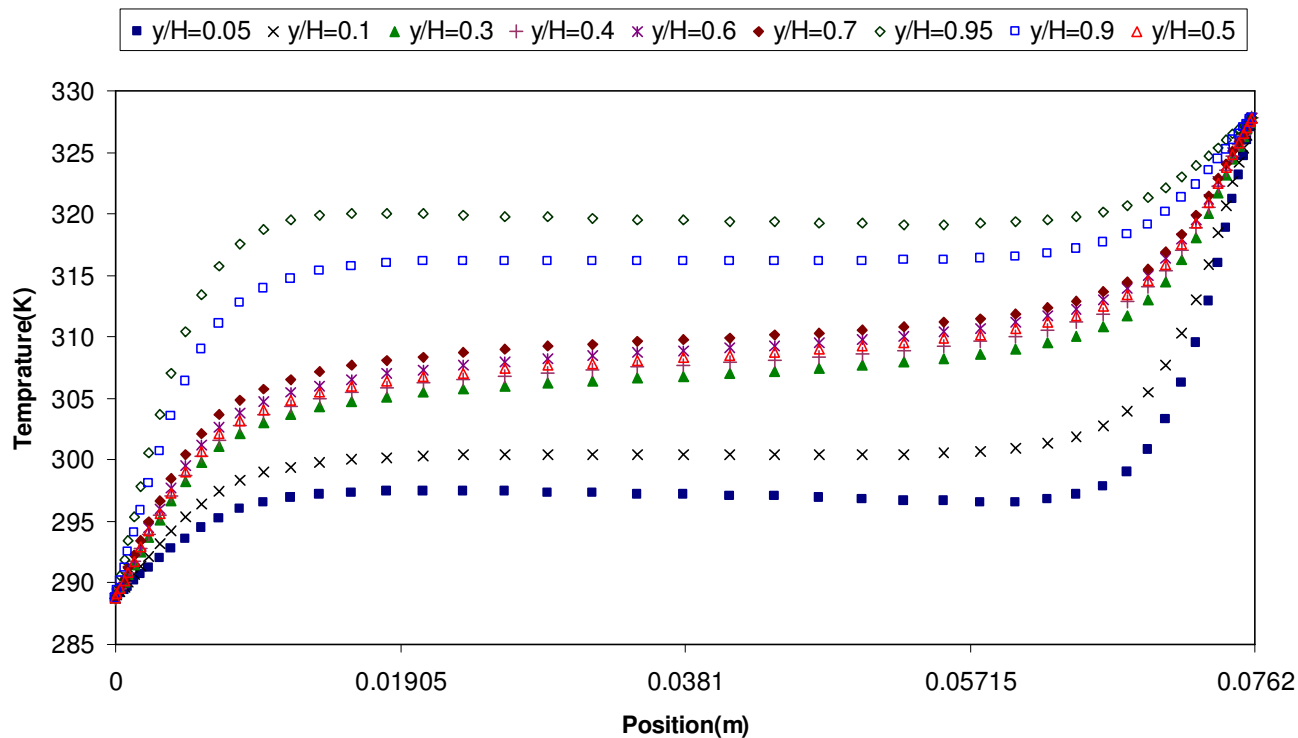


Figure 4. Variation of temperature in $Ra=1.43 \times 10^6$, in comparison with Betts and Bokhari (2000) experiment.

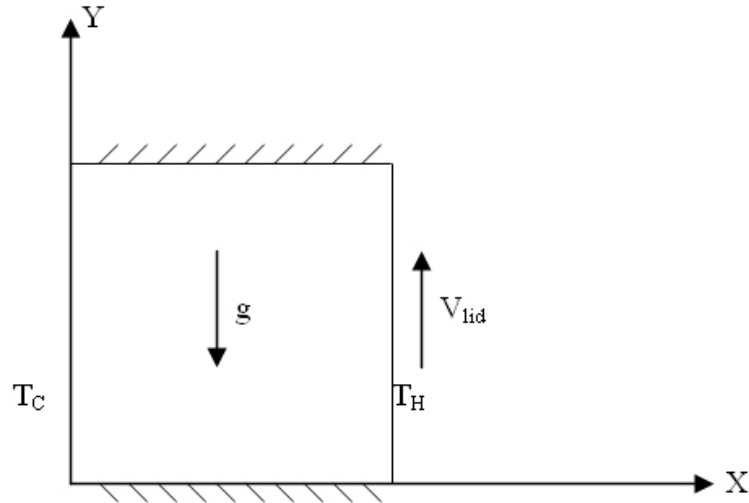


Figure 5. Schematic of the problem.

Table 5. Details of the rectangular and square enclosures.

Variable	Betts and Bokhari	Tian and Karayiannis
Rayleigh number	1.43×10^6	1.58×10^9
Length of enclosure (m)	2.18	0.75
Wide of enclosure (m)	0.076	0.75
Left wall temp.	15.6	50
Right wall temp.	54.7	10
Prandtl number	0.697	0.707

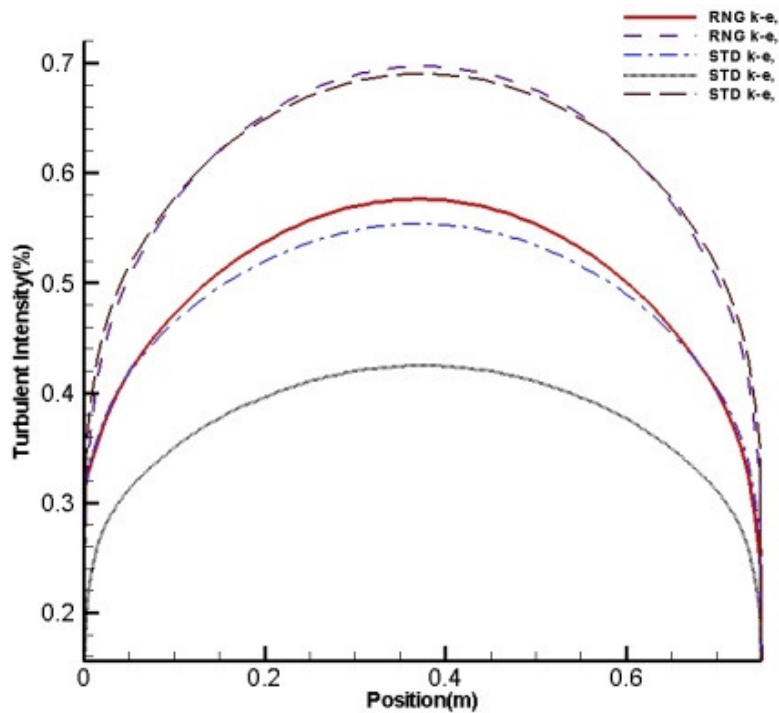


Figure 6. Turbulent Intensity at $y/H=0.5$.

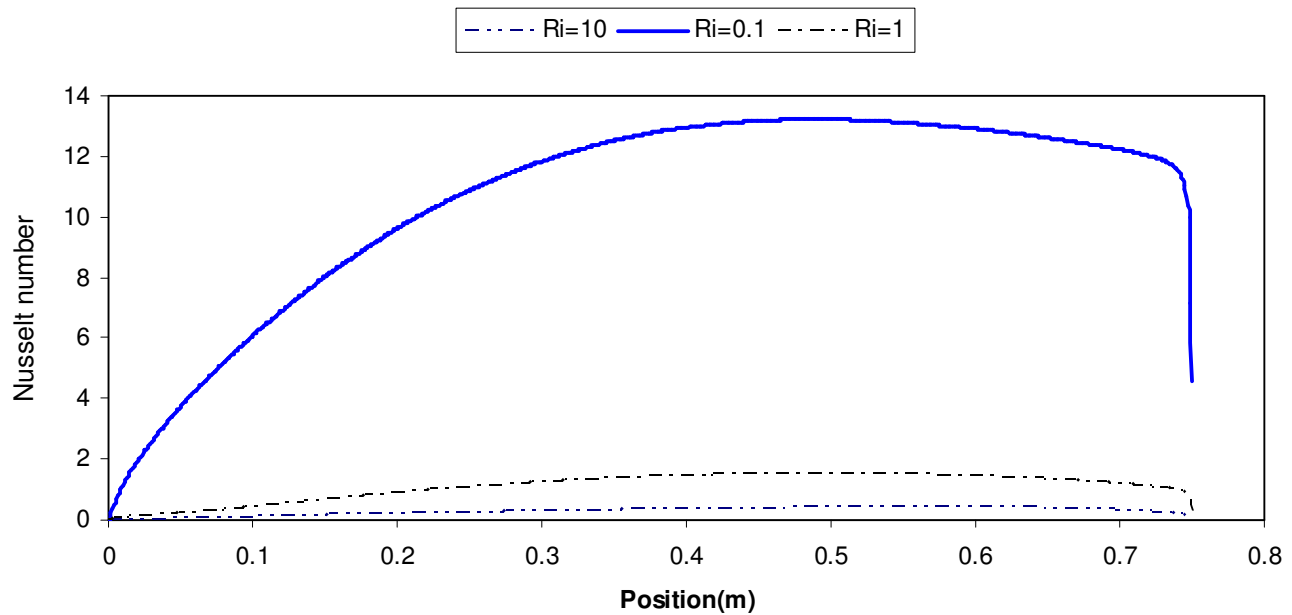


Figure 7. Nusselt number along the hot wall

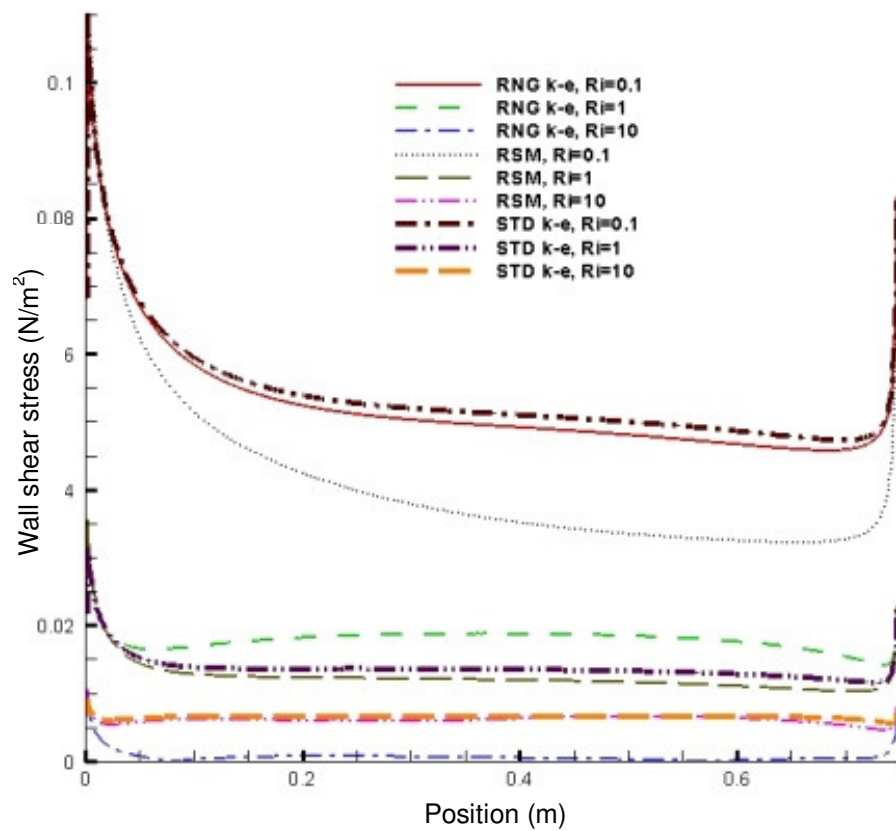


Figure 8. Wall shear stress at $y/H=0.5$.

The wall shear stress for $y/H=0.5$ and also in the different Ri and turbulence models is illustrated in Figure 8. It can

be seen from this figure that, in a boundary layer of the left wall, due to the non-existence of slip condition, the

value of shear stress hugely increased linearly; therefore, the value of shear stress decreased moderately at the boundary layer of the right wall, and also at this point, its value linearly increased hugely due to lack of slip condition. Another important point is that, if the value of Ri increases, the effect of fluid viscosity is low. As a result, it is reasonable to reduce the value of shear stress.

Conclusions

Mixed convection usually induced in enclosures or cavities heating elements on one of its walls or both is important from both theoretical and practical point of view. In fact, this configuration can be encountered in various engineering application. Numerous studies related to mixed convection in enclosures have been reported in order to investigate the heat transfer and fluid flow in such geometries. The following conclusions can be drawn from the results of the present work:

- 1) In large Richardson numbers, the natural convection is a major parameter of heat transfer in a cavity.
- 2) Heat transfer is decreased with increasing Ri number.
- 3) When the Reynolds number increases, the circulation of flow vortices increases and becomes stronger, making the forced convection effect more dominant for different values of Richardson numbers.
- 4) If the value of Ri increases, the effect of fluid viscosity is low.

Nomenclature

u, v : velocities in x and y directions (m/s); x, y : Cartesian coordinates (m); W : height of the cavity (m); P pressure (N/m^2); T temprature (K); t Time (Sec); g gravitational acceleration (m^2/s); K Turbulent kinetic energy transport (m^2/s^2); k , ($W/m.k$) thermal conductivity; Re , Reynolds number; Ri , Richardson number; Gr , Grashof number; Nu Nusselt number; Pr , Prandtl number, Ra , Rayleigh number; ϵ , Dissipation of turbulent kinetic energy transport (m^2/s^3); ν_t turbulent kinematic viscosity (m^2/s); σ_T turbulent thermal diffusivity (m^2/s); β thermal expansion coefficient ($1/K$); ρ density (kg/m^3); ν (m^2/s) kinematic viscosity; h =hot wall; c cold wall; m mean; lid Lid.

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