Application of Improved Blocked-Off Method to Simulate the Interacting Influences of Obstacle Shape and Wall Velocity on the Turbulent Mixed Convection Flow in a Trapezoidal Cavity

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Abstract: In the current research, interaction influences of obstacle shape and top wall velocity on the hydrothermal behaviours of the turbulent mixed convection flow in a trapezoidal cavity are numerically simulated. To achieve this goal, three different shapes of the obstacles including semicircular, triangular, rectangular are considered. Dimensions of these obstacles are chosen so that the environment around all three of them is same. The RNG $k-\varepsilon$ model is chosen to simulate the turbulent flow. To model the inclined or curved walls of trapezoidal cavity and obstacles, the improved blocked-off method is applied. Results show that the obstacle shape and top wall velocity have a significant influence on the thermal and hydrodynamic behaviours. In fact, the highest magnitude of heat transfer rate along the bottom wall occurs in the cavity with the rectangular obstacle and for the highest magnitude of top wall velocity; whilst its lowest magnitude is related to the pure free convection and for the cavity with the semicircular obstacle. Besides, the lowest and highest magnitudes of temperatures fields occur for the cavities with rectangular and triangular obstacles, respectively.

Keywords: Improved Blocked-Off Method, Rectangular Obstacle, RNG $k-\varepsilon$ MethodTurbulent Flow, Semicircular Obstacle, Triangular Obstacle

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1 INTRODUCTION

The study of laminar and turbulent forced, free and mixed convection fluid flow in cavities and enclosures has long been considered by a large number of thermal science researchers [1-8]. Koufi et al. [9] numerically investigated the influences of inlet and outlet openings on the turbulent mixed convection flow in an open cavity. Miroshnichenko et al. [10] simulated the coupling between the turbulent convection flow and surface thermal radiation in a square enclosure having local heater.

Huang et al. [11] analyzed the influence of buoyancy force on the turbulent mixed convection flow inside a cavity using the large eddy simulation (LES) approach. It can be concluded that the results of LES method are in good agreement with experimental data. Olazo-Gómez et al. [12] studied the impact of a vertical glazed wall on the conjugate laminar and turbulent convection flow in a cavity in the presence of surface thermal radiation. Rodrigues and de Lemos [13] applied the $k-\varepsilon$ and local thermal non-equilibrium approaches to analysis the turbulent convection flow in a cavity filled with a porous medium. This attention has been due to the importance of these types of flows in design of many engineering and industrial applications such as chemical catalytic reactors, chemical vapor deposition equipment, furnace engineering, solar collectors, nuclear reactors, and electronics cooling systems [14-19].

In addition, it should be noted that the laminar and turbulent convection fluid flow inside the cavities and enclosures is known as a benchmark problem and it is used to validate various issues [20-26]. Finding suitable methods to control the thermal and hydrodynamic behaviours in different systems is one of the main goals of thermal science researchers. So far, several different approaches have been proposed in this regard. One of the effective and useful methods proposed is to use an obstacle in the flow domain [27-34]. As a result, several studies have been conducted to investigate the effects of different obstacles on the heat transfer rates and friction coefficients in the various geometers such as ducts, cavities and enclosures for laminar and turbulent fluid flow regimes [35-42].

Among these studies, Kareem and Gao [43] numerically investigated the effects of a rotating cylinder on the hydrothermal behaviours of turbulent mixed convection flow in a lid-driven cavity. Motlagh and Sarvari [44] used the large eddy simulation and proper orthogonal decomposition (POD) methods to study the turbulent mixed convection flow in the ventilated cavity with an obstacle. Barman and Dash [45] investigated the effects of obstacle positions on the turbulent forced convection fluid flow in a duct with two forward facing steps. Menni et al. [46-48] analyzed the characteristics of various baffles on the trends of turbulent convection heat transfer inside channels under different conditions.

Siba and Jehhef [49] numerically studied the impacts of a triangular obstacle on the turbulent forced convection heat transfer in a channel with a sudden expansion. Although so far, several studies have been done to investigate the influences of obstacles on the hydrothermal behaviours of laminar or turbulent convection fluid flow in different geometries; but, to the best of the authors' knowledge, analysis of the interacting effects of obstacle shape and top wall velocity on the turbulent mixed convection flow in a trapezoidal cavity have not been investigated by other researchers. It should be mentioned that the obstacles shape and wall velocity can reinforce or weaken each other's effects on the hydrodynamic and thermal behaviours. Also, it is important to note that in this study, calculations are performed for three shapes of obstacle (rectangular, triangular and semicircular) at different values of top wall velocity. However, the popular and efficient RNG $k-\varepsilon$ approach is used to model the turbulent fluid flow, whereas, the improved blocked-off method is applied to simulate the inclined or curved walls of trapezoidal cavity and obstacles.

2 PROBLEM DESCRIPTION

As it is mentioned earlier, this paper attempts to analysis the interaction impacts of the top wall velocity and shape of various obstacles (rectangular, triangular and semi-circular) on the hydrodynamic and thermal behaviours of mixed turbulent convection flow in trapezoidal cavities. The geometries of these trapezoidal cavities are shown in "Figs. 1(a to c)".

It should be noted that the size of the cavities walls (except the size of obstacles walls) is the same and equal to:

$a = 4cm, b = 2cm, c = 4.7cm$ and $d = 14cm$

The dimensions of the obstacles are considered so that the environment around all three of them is equal to each other. Therefore, the radius of semicircular obstacle, the width of rectangular obstacle and the height of triangular obstacle are equal to:

$$R = \frac{b}{2} = 1cm, W = R\left(\frac{\pi}{2} - 1\right) = 0.57cm \text{ and}$$

$$H = R\sqrt{\frac{\pi^2}{4} - 1} = 1.21cm$$

Also, the flow and thermal boundary conditions for all three cavities are presented in "Table 1".

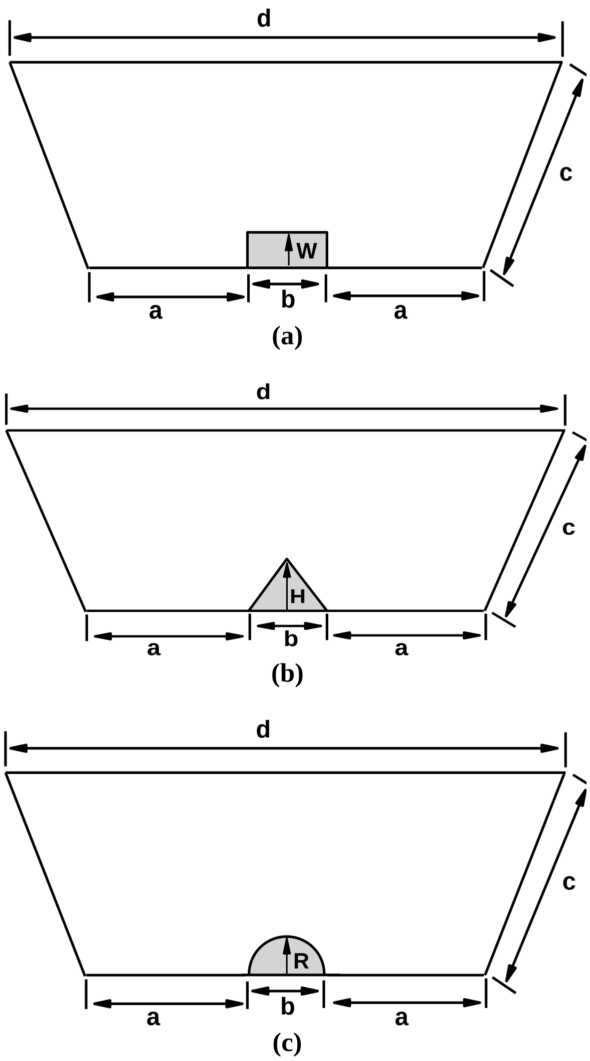


Fig. 1 Cavity with the semicircular obstacle: Geometry of the studied trapezoidal cavities.

Table 1 Flow and thermal boundary conditions for all studied cavities

	Flow boundary conditions	Thermal boundary conditions
Top walls	$u = U_0, v = 0$	$T = 300K$
Inclined side walls	$u = v = 0$	$\frac{\partial T}{\partial n} = 0$
Bottom walls including the obstacles surfaces	$u = v = 0$	$T = 400K$

3 GOVERNING EQUATIONS

The basic equations for turbulent mixed convection flow in cavity under study can be written in vector forms as follows:

$$\frac{\partial U_j}{\partial x_j} = 0 \tag{1}$$

$$\frac{\partial (U_j U_i)}{\partial x_j} = -\frac{1}{\rho}\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\upsilon\left(\frac{\partial U_i}{\partial x_j}\right) - \overline{u_i u_j}\right] + B_i \tag{2}$$

$$\frac{\partial (U_j T)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\frac{\upsilon}{Pr} - \frac{\upsilon_t}{Pr_t}\right)\frac{\partial T}{\partial x_j}\right] \tag{3}$$

In these equations, the $\overline{u_i u_j}, \upsilon_t$ and Pr_t terms are respectively the Reynolds stress, turbulence viscosity and turbulence Prandtl number. Also, the B_i term is related to the contribution of buoyancy force. In this research, the turbulence terms are calculated using the RNG $k - \varepsilon$ turbulence model. According to this model:

$$\overline{u_i u_j} = -\upsilon_t\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_i}{\partial x_j}\right) + \frac{2}{3}\delta_{ij}k \tag{4}$$

$$d\left(\frac{\rho^2}{\sqrt{\varepsilon\mu}}\right) = 1.72\frac{\upsilon_t}{\sqrt{{\upsilon_t}^3 - 1 + C_v}}d\upsilon_t \tag{5}$$

In the above equations, the k and ε parameters are respectively the turbulent kinetic energy and the turbulent energy dissipation rate. The following equations are used to calculate these terms:

$$\rho\frac{\partial}{\partial x_j}(kU_i) = \frac{\partial}{\partial x_j}\left(a_k\mu_{eff}\frac{\partial k}{\partial x_j}\right) + G_k + G_b - \rho\varepsilon + S_k \tag{6}$$

$$\rho\frac{\partial}{\partial x_i}(\varepsilon U_i) = \frac{\partial}{\partial x_j}\left(a_\varepsilon\mu_{eff}\frac{\partial \varepsilon}{\partial x_j}\right) + C_{1\varepsilon}\frac{\varepsilon}{k}(G_k + C_{3\varepsilon}G_b) - C_{2\varepsilon}\,\rho\frac{\varepsilon^2}{k} - R_\varepsilon + S_\varepsilon \tag{7}$$

In these equations:

$$G_k = -\rho\overline{u_i u_j}\frac{\partial u_j}{\partial x_i} \tag{8}$$

$$R_\varepsilon = \frac{C_\mu\rho\eta^3(1 - \frac{\eta}{\eta_0})}{1 + \beta\eta^3}\frac{\varepsilon^2}{k} \tag{9}$$

$$\eta = \frac{S_k}{\varepsilon} \quad (10)$$

$$\mu_{eff} = \rho C_\mu \frac{k^2}{\varepsilon} \quad (11)$$

The constants shown in the above equations are presented in "Table 2".

Table 2 The constants used in the $k-\varepsilon$ equations [50-52]

β	c_v	η_0	c_μ	$C_{2\varepsilon}$	$C_{1\varepsilon}$	Pr_t
0.012	100	4.38	0.0845	1.68	1.42	0.85

However, more details of the RNG $k-\varepsilon$ turbulence model are presented in Refs. [50-52].

4 NUMERICAL SOLUTIONS AND CODE VALIDATION

To obtain the hydrothermal behaviours of turbulent mixed convection flow in the studied cavity, the basic equations (continuity, Navier-Stokes, energy, turbulent kinetic energy and turbulent energy dissipation) are discretized by integration on the volume of each element (FVM). Then, the obtained discrete equations are solved by using the line-by-line iterative approach and three-diagonal matrix method. It should be mentioned that the Simple algorithm is applied to couple the velocity and pressure fields.
The optimum meshes used to numerically solve the governing equations in all three cavities under study are presented in "Table 3". It is important to note that the considered meshes are concentrated near the cavities and obstacles walls to achieve more accurate results.

Table 3 The optimum meshes used to solve the governing equations in the studied cavities

Cavity with rectangular obstacle	Cavity with triangular obstacle	Cavity with semicircular obstacle
280×140	300×160	280×150

Also, the computational times for solving the governing equations by using the provided meshes are presented in "Table 4". As can be seen from this table, the cavities with triangular and rectangular obstacles have the highest and lowest computational times, respectively.
In the present study, the improved blocked-off method [53-54] is applied to model the inclined side walls of the trapezoidal cavity and obstacles surfaces. The difference between this method and the blocked-off method [55-58] is related to model the curved or inclined surfaces. In fact, the improved blocked-off method simulates exactly the inclined or curved walls same as the real irregular walls. A schematic of the simulation of curved or inclined surfaces using the improved blocked-off method is shown in "Fig. 2". More details of this approach were fully explained in Refs. [53-54]. Therefore, these explanations are not presented here to avoid repetition.

Table 4 The computational times for solving the governing equations in the studied cavities

Cavity with rectangular obstacle	Cavity with triangular obstacle	Cavity with semicircular obstacle
$24\ min$	$31\ min$	$27\ min$

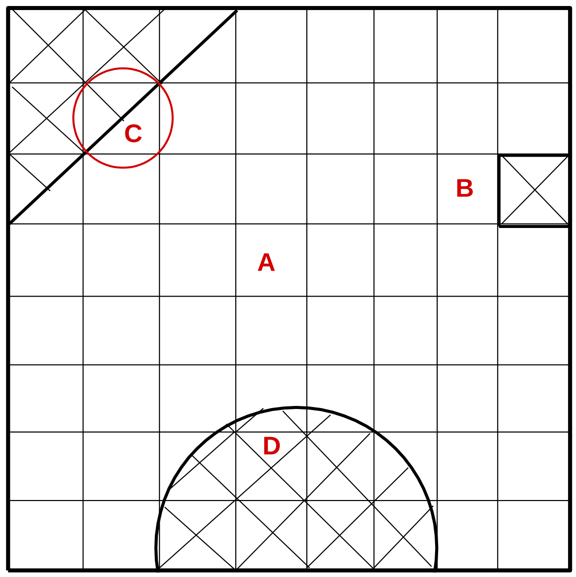


Fig. 2 A schematic of the simulation of curved or inclined surfaces using the improved blocked-off method.

Also, the convergence criterions considered to numerically solve the governing equations are as follows:

$$Max\left|\frac{\Lambda^{\omega}(m,n) - \Lambda^{\omega-1}(m,n)}{\Lambda^{\omega}(m,n)}\right| \le 10^{-6} \quad (12)$$

$$\sum_{m=1}^{m=M}\sum_{n=1}^{n=N}\left|\Lambda^{\omega}(m,n) - \Lambda^{\omega-1}(m,n)\right| \le 10^{-5} \quad (13)$$

In these relations, the Λ parameter denotes the velocity, pressure, temperature, turbulent kinetic energy and turbulent energy dissipation fields. Also, the ω symbol is the iteration step.

Furthermore, it should be mentioned that all computations in this research are done by using a homemade code written in Fortran 90 and a personal computer Intel (R), Core (TM) i5-7400, CPU 3.0 GHz and 8.00 GB of RAM.
To ensure the accuracy of the numerical methods and algorithms considered in this research, the results of this study are compared with the results presented by Promvonge [27] and Fawaz et al. [29]. These comparisons are shown in “Fig. 3”. As can be seen from this figure, the numerical approach considered in this research has a high accuracy.

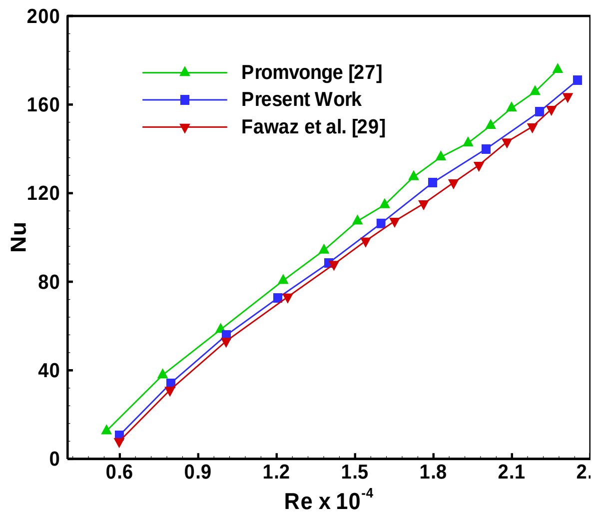
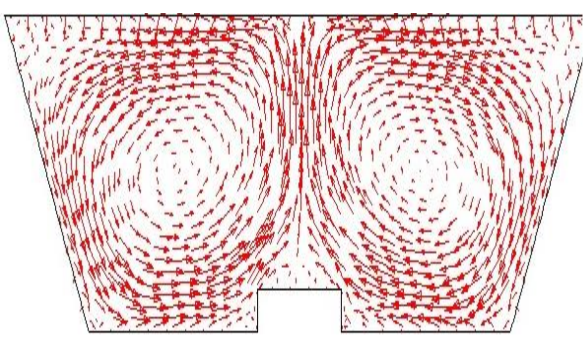


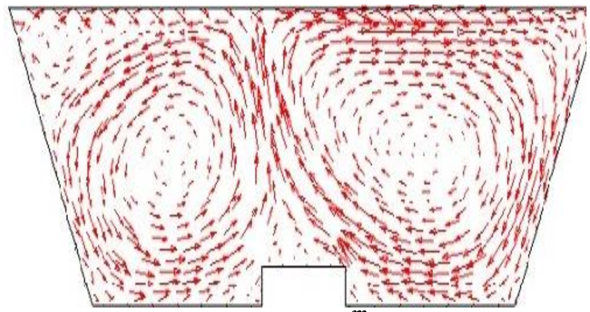
Fig. 3 Comparison of the magnitudes of average Nusselt number at various values of Reynolds numbers with the findings of Promvonge [27] and Fawaz et al. [29].

5 RESULTS AND DISCUSSIONS

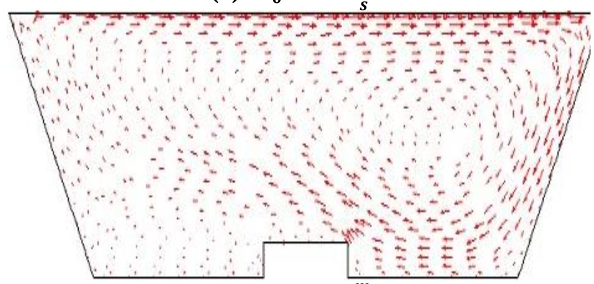
First, to analyze the interaction impacts of obstacle shape and top wall velocity on the flow features, distributions of the velocity vectors in the trapezoidal cavity are shown in “Figs. 4 to 6” for three shapes of obstacle (rectangular, triangular and semicircular) at different values of U_0.



(a): $U_0 = 0$

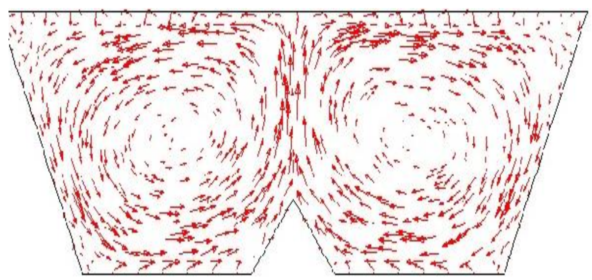


(b): $U_0 = 0.1\,\frac{m}{s}$

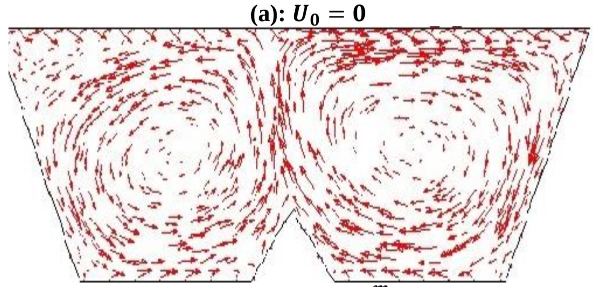


(c): $U_0 = 0.5\,\frac{m}{s}$

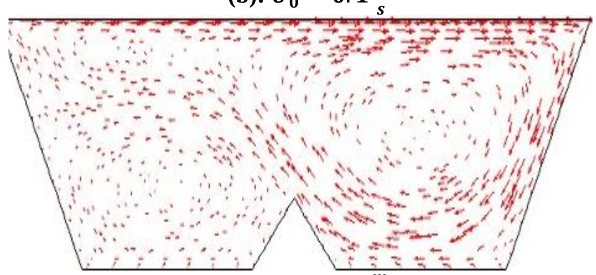
Fig. 4 Impacts of top wall velocity on the distributions of the velocity vectors in the trapezoidal cavity with the rectangular obstacle.



(a): $U_0 = 0$

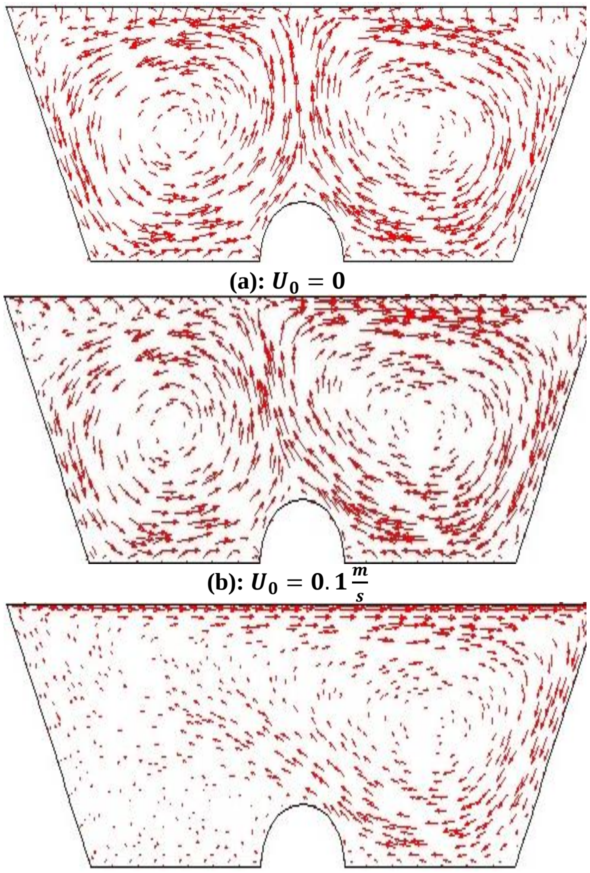


(b): $U_0 = 0.1\,\frac{m}{s}$



(c): $U_0 = 0.5\,\frac{m}{s}$

Fig. 5 Impacts of top wall velocity on the distributions of the velocity vectors in the trapezoidal cavity with the triangular obstacle.



(a): $U_0 = 0$

(b): $U_0 = 0.1\frac{m}{s}$

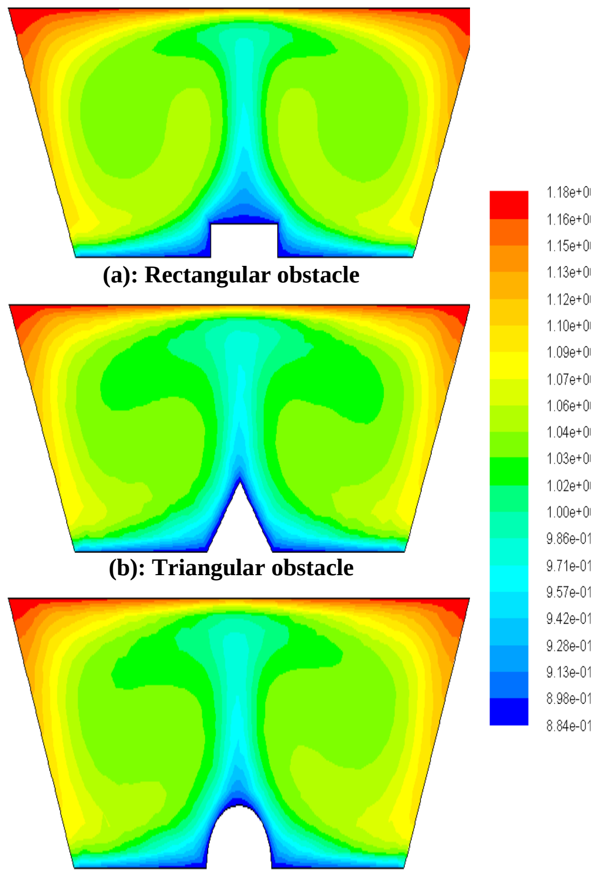
(c): $U_0 = 0.5\frac{m}{s}$

Fig. 6 Impacts of top wall velocity on the distributions of the velocity vectors in the trapezoidal cavity with the semicircular obstacle.

Accurate analysis of these figures clearly shows that for all three shapes of obstacle, the flow pattern is significantly dependent on the magnitudes of top wall velocity. In the case of $U_0 = 0$ (pure free convection), the streamlines are symmetrical such that two equal recirculation zones are formed on the left and right hands of the different obstacles. It should be noted that in this case, the movement of fluid flow in the cavity is affected by the buoyancy force and changes of fluid density in different parts of the cavity.

In "Fig. 7", distributions of fluid density changes in the cavity are shown for all three obstacles geometries.

As it is seen from these Figures, the obstacle shape has a considerable effect on the density changes in the cavity domain. In the cases of $U_0 \neq 0$ (mixed convection), the streamlines are not symmetrical and the size of the right recirculation zones enhances by increasing the top wall velocity. In fact, in these cases, the buoyancy force and the top wall movement lead to the movement of fluid flow in the cavity. Of course, it should be noted that the role of the buoyancy force decreases as the top wall velocity increases. Another noteworthy point in these figures is that the maximum and minimum effect of top wall velocity on the streamlines is related to the cavities with the rectangular and triangular obstacles, respectively.



(a): Rectangular obstacle

(b): Triangular obstacle

(c): Semicircular obstacle

Fig. 7 Distributions of density changes in the trapezoidal cavity for all three obstacles geometries ($\boldsymbol{U_0 = 0}$).

To illustrate the influences of obstacle shape and top wall velocity on the thermal characteristics in the trapezoidal cavity under study, distributions of temperature fields are displayed in "Figs. 8 to 10".

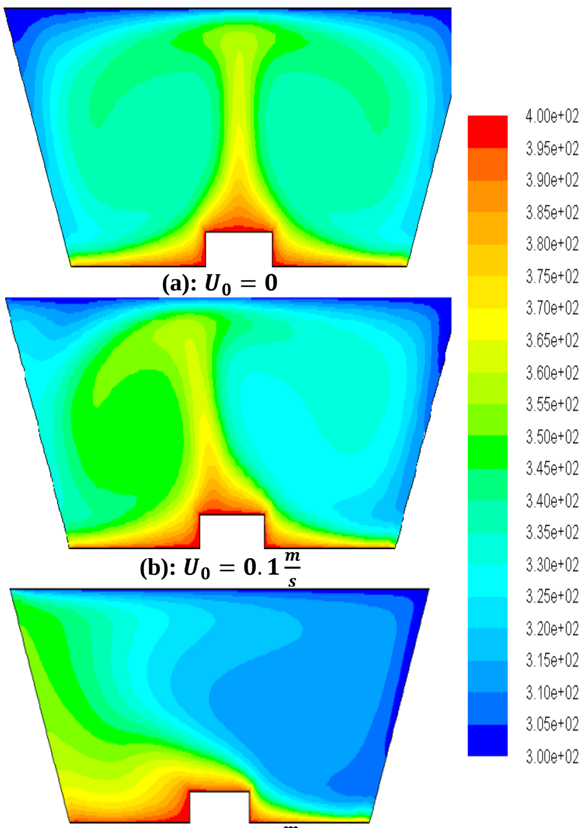
Careful examination of these figures clearly shows that the top wall velocity has an important effect on the trends and values of the temperature distributions inside the cavity. In the absence of forced convection heat transfer ($U_0 = 0$), the temperature distributions inside the cavity are symmetrical for all three shapes of obstacles.

In this case, the highest values of fluid temperature are related to the areas near the cavity bottom wall and the areas above the obstacles. It should be mentioned that in these areas, the fluid density has its lowest values.

In other words, in this case, variations of temperature field are the opposite of the variations of fluid density in the cavity.

In the presence of both forced and free convection heat transfer mechanisms ($U_0 \neq 0$), the temperature field inside the cavity is not symmetrical. In these cases, by moving the top wall from left to right, the hot areas are transferred to the left side of cavity domain. In fact, by enhancing the top wall velocity, more hot areas are located on the left side of the cavity.

Another interesting point in "Figs. 8 to 10" is that the lowest and highest values of temperatures field in the cavities are respectively related to the rectangular and triangular obstacles.

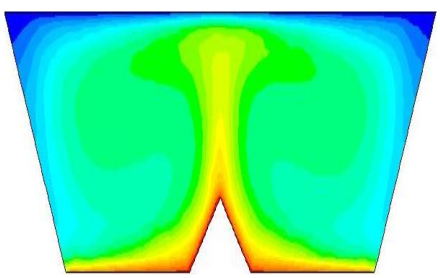


(a): $U_0 = 0$

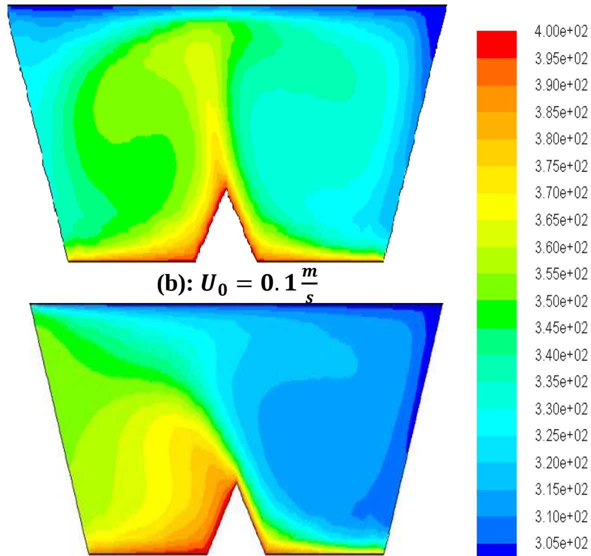
(b): $U_0 = 0.1\,\frac{m}{s}$

(c): $U_0 = 0.5\,\frac{m}{s}$

Fig. 8 Impacts of top wall velocity on the temperature distributions in the trapezoidal cavity with the rectangular obstacle.



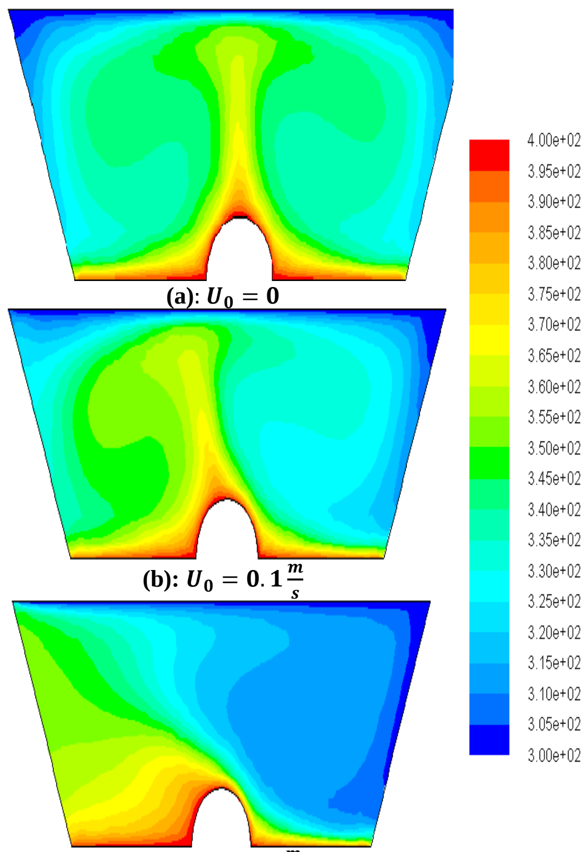
(a): $U_0 = 0$



(b): $U_0 = 0.1\,\frac{m}{s}$

(c): $U_0 = 0.5\,\frac{m}{s}$

Fig. 9 Impacts of top wall velocity on the temperature distributions in the trapezoidal cavity with the triangular obstacle.



(a): $U_0 = 0$

(b): $U_0 = 0.1\,\frac{m}{s}$

(c): $U_0 = 0.5\,\frac{m}{s}$

Fig. 10 Impacts of top wall velocity on the temperature distributions in the trapezoidal cavity with the semicircular obstacle.

To further study the interaction influences of obstacle shape and top wall velocity on the thermal behaviours in the cavities under study, the heat transfer rates along the cavities bottom wall (including the obstacle surfaces) are shown in “Fig. 11”.

A comparison of the data presented in this figure clearly shows that the magnitudes of heat transfer rate increase with augmentation of the top wall velocity. Also, it is quite clear that the highest and lowest values of heat transfer rates on the bottom wall are related to cavities with the rectangular and semicircular obstacles.

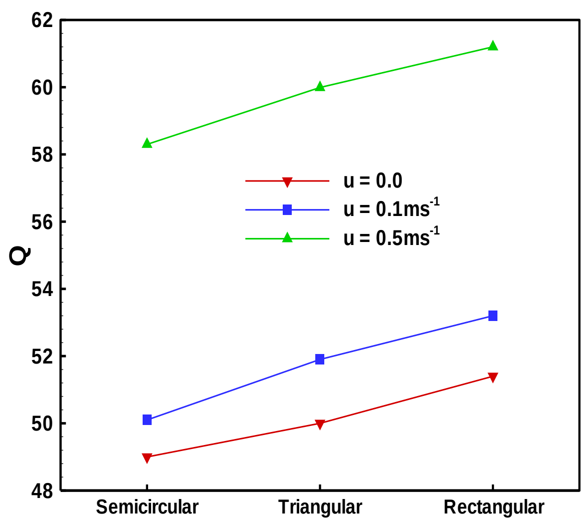


Fig. 11 Interacting influences of obstacle shape and top wall velocity on the heat transfer rates along the cavity bottom wall.

6 CONCLUSIONS

Interacting influences of obstacle shape (rectangular, triangular and semicircular) and top wall velocity on the hydrothermal features of turbulent mixed convection flow in a trapezoidal cavity are analyzed with details. The main results of this study can be summarized as follows:

✓ For the case of pure free convection heat transfer, distributions of velocity vectors and temperature fields inside the cavity are symmetrical for all three shapes of obstacles. Also, an enhancement in the top wall velocity leads to a decrease in the role of buoyancy force on the distributions of velocity vectors and temperature fields.

✓ The maximum and minimum influences of top wall velocity on the velocity vectors occur respectively in the cavities with rectangular and triangular obstacles.

✓ By increasing the top wall velocity, more hot areas are located on the left side of the cavity.

✓ The lowest and highest values of temperatures fields are respectively related to the cavities with rectangular and triangular obstacles.

✓ The highest values of heat transfer rate on the bottom wall occur for the highest values of top wall velocity and for the cavity with the rectangular obstacle.

✓ The lowest magnitudes of heat transfer rate on the bottom wall occur in the absence of forced convection flow (pure free convection, $U_0 = 0$) and for the cavity with the semicircular obstacle.

7 NOMENCLATURE

b	Obstacles length on cavities bottom wall, (cm)
B_i	Buoyancy force
H	Height of triangular obstacle, (cm)
k	Turbulent kinetic energy
P	Pressure, $(N.m^{-2})$
Pr	Prandtl number
Pr_t	Turbulence Prandtl number
Q	Heat transfer rate, (W)
R	Radius of semicircular obstacle, (cm)
Re	Reynolds number
T	Temperature, (K)
U_0	Velocity of top wall, $(m.s^{-1})$
U_i or U_j	Velocity vector, $(m.s^{-1})$
$\overline{u_i u_j}$	Reynolds stress
W	Width of rectangular obstacle, (cm)
x_i or x_j	Cartesian coordinates, (m)

Greek Symbols

ε	Turbulent energy dissipation rate
υ	Viscosity, $(m^2.s^{-1})$
υ_t	Turbulence viscosity
ρ	Density, $(Kg.m^{-3})$

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