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A Novel Passive Thermal Diode Based on Natural Convection Using an Internal Baffle and a One-Way Valve: A Numerical Investigation

Mostafa Rahimi[1,*], Arezoo Sadat Bathaei[1]

[1] Department of Mechanical Engineering, University of Mohaghegh Ardabili, Ardabil, Iran; rahimi@uma.ac.ir; bathaei.a91@gmail.com.

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Abstract

The present paper deals with a detailed numerical analysis of natural convective heat transfer in a rectangular enclosure subjected to heating from one side. The analysis is done independently for two different fluids: water (Pr = 7) and air (Pr = 0.7) by considering the Rayleigh number variation between 10^3 and 4×10^5 to cover both asymptotic and early laminar convection regimes. The solution of the governing equations (continuity, momentum, and energy) is obtained using a finite difference scheme with the Boussinesq approximation in MATLAB. The validation of the numerical method is successfully carried out with the help of benchmark solutions of De Vahl Davis and highly accurate solutions of Wan et al. After that, an obstacle (baffle) of small thickness is fixed at the mid-height of the cavity without hindering the flow, and the effect of the same on the overall heat transfer rate is analyzed. Finally, a low-density flap is attached to the upper edge of the obstacle to allow one-directional flow. Such a configuration allows designing a one-way thermal insulator; convective heat flow can be easily allowed from one side, while it is efficiently restricted from the opposite side when the valve is shut. These findings suggest that the presence of the baffle alone has no significant impact on the heat transfer process, which is reduced only by 2% for air and 0.2% for water due to the use of the baffle alone. In comparison, the heat transfer process is decreased by up to 95% for air and 92% for water with the help of the baffle-valve system.

Keywords: Natural convection, Rectangular enclosure, One-way heat transfer, Baffle, Thermal insulation, Numerical simulation.

1 | Introduction

Buoyant convection in enclosed geometries of fluids in rectangular enclosures heated on one side has been an area of great interest since its extensive use in engineering and environmental systems. The flows that exist

Corresponding Author: rahimi@uma.ac.ir
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in the rectangular cavity arise from complicated mechanisms associated with the interaction of the fluid, the temperature gradients, and the boundaries. These flows play a key role in several applications including solar energy collection, double-pane windows, cooling of electronics, building ventilation, and thermal regulation of electrical machines [1], [2].

Early accurate simulations of this problem have been carried out by Saitoh and Hirose [3] and De Vahl Davis [4]. They provide benchmark solutions to the problem of natural convection in a square cavity heated on the vertical walls while maintaining adiabatic boundaries on the horizontal walls. Their work has provided a standard for validating the computational algorithms. Subsequently, Nithyadevi et al. [5] studied natural convection in a rectangular enclosure with partially active side walls.

Several research works have been conducted on the impact of internal barriers or partitions on the thermal behavior of enclosures. For example, Raji et al. [6] investigated a square cavity filled with various square-shaped blocks. They found that heat transfer and flow intensity decrease as the number and thermal conductivity of the blocks increase. The natural convection in the presence of a large temperature difference in a rectangular cavity around a square solid body located at the bottom of the cavity has been investigated by Bouafia and Daube [7]. It was found that at low Rayleigh numbers, a symmetric flow is formed, while at high Rayleigh numbers, asymmetry occurs and flow oscillations take place.

The impact of a horizontal fin attached to the hot side of a cubic enclosure was investigated by Da Silva and Gosselin [8], and it was revealed that the mean value of heat flux rises as a function of fin length. Zhang et al. [9] conducted numerical analysis on the effect of a permeable screen in a vertical cavity.

Laguerre et al. [10] conducted both experimental and numerical investigations of the cavity with obstacles of cylindrical form, emphasizing the great influence of moisture on increasing the air velocity and that the radiation exchange between obstacles should not be ignored.

Among other references, we can mention research by Ostrach [1] devoted to basic principles of buoyancy-driven flows, as well as the review [1] devoted to natural convection in enclosed volumes conducted by Catton. Gebhart et al. [11] give a detailed description of buoyancy-driven flows and transport processes. Incropera et al. [2] provide detailed correlations for natural convection in different geometries. Some numerical results for high-accuracy benchmarks for buoyancy-driven cavities were obtained in [12]. Natural convection in a square cavity with a local heat source was studied by Sheremet and Pop [13]. As an example of a systematic approach for analysis of heat transfer in a vertical layer, we can quote the work by Bejan [14]. In this work, the classical correlation for the average Nusselt number is given. In a recent study, Hussein et al. [15] have investigated the effect of baffle length on natural convection in a nanofluid-filled enclosure. In contrast, Barman and Rao [16] have studied the natural convection in a wavy porous enclosure having an insulated obstacle.

On the other hand, Rachedi et al. [17] have numerically analyzed natural convection in a rectangular enclosure with discrete heat flux. Also, El Hamri et al. [18] have performed the analysis of entropy generation in the cavities containing a horizontal partition. Furthermore, recently some research has been conducted in relation to applied science regarding smart passive insulation [19] and one-way heat wall experiments [20].

Although considerable attention has been paid to natural convection in enclosures with obstacles, the idea of developing a unidirectional thermal insulator in which heat flows freely in one direction and gets blocked in the other direction has never been explored before. The objective of this paper is to introduce such a simple passive mechanism, which has an internal baffle along with a one-way valve. The numerical analysis will be carried out at a wide range of Rayleigh numbers and for two fluids.

2 | Problem Statement

Consideration is given to a rectangular cavity with height L and width δ (schematically presented in *Fig. 1*). The cavity is filled with air or water. Vertical walls are maintained at the constant temperatures of T_h (hot) and Tc (cold), whereas horizontal walls are adiabatic. The Rayleigh number is varied from 10^3 to 4×10^5 in order to cover the asymptotic and early stages of laminar convection flow.

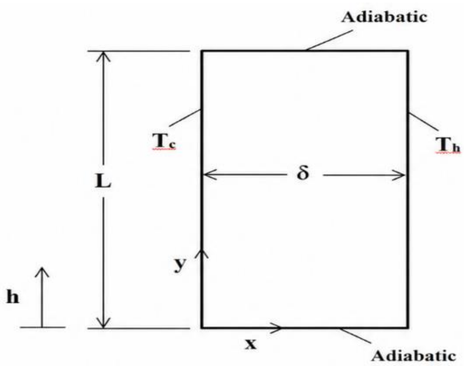


Fig. 1. Schematic of the computational cavity under investigation.

Then a thin solid barrier (obstacle) is added to the mid-height of the cavity (presented schematically in *Fig. 2a*). An investigation of the impact of such an obstacle on the flow and temperature fields is performed. Next, a thin light barrier (valve) is installed at the upper end of the barrier, closing the upper gap (*Fig. 2b*). Such a valve enables free flow only to the left, but blocks any reverse flow, which leads to the formation of one-way insulation.

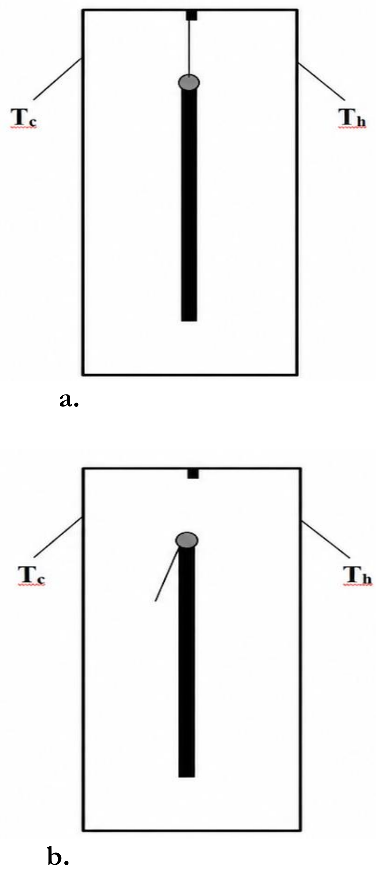


Fig. 2. Schematic of the cavity with an obstacle;
a. front view and b. side view.

3 | Governing Equations and Numerical Method

The flow is assumed steady, laminar, and two-dimensional. The fluid properties are constant except for the density variation in the buoyancy term, which is modelled using the Boussinesq approximation. The governing equations are written as:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0. \tag{1}$$

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial P}{\partial x}+\frac{\mu}{\rho}\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right)-g\frac{\partial h}{\partial x}. \tag{2}$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho}\frac{\partial P}{\partial y}+\frac{\mu}{\rho}\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)-g\frac{\partial h}{\partial y}. \tag{3}$$

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right). \tag{4}$$

In the employed numerical method, after discretizing the computational domain into a mesh, the velocity components, temperature, and pressure were defined at each grid point. The four governing differential equations were discretized using the central difference scheme. The resulting system of algebraic equations was solved simultaneously by imposing the prescribed boundary conditions, including constant-temperature conditions on the vertical walls, adiabatic conditions on the horizontal walls, and specifying the pressure at a reference point within the enclosure. The obtained values of the dependent variables were then used as updated initial guesses, and the iterative procedure was repeated until convergence was achieved. The density, ϱ\rhoϱ, at each computational node was determined according to the local temperature. Owing to the fully implicit numerical formulation, the convergence rate was satisfactory, and convergence was generally achieved within fewer than 20 iterations for the different geometrical configurations.

To validate the accuracy and reliability of the numerical procedure, the governing equations were first solved for a benchmark geometry previously investigated in the literature, and the obtained results were compared with published data. The benchmark problem considered was the natural convection heat transfer in a square enclosure containing a centrally located heated obstacle, as reported by Nazari and Ramezani [21]. The configuration and geometric specifications of the enclosure are illustrated in *Fig. 3*.

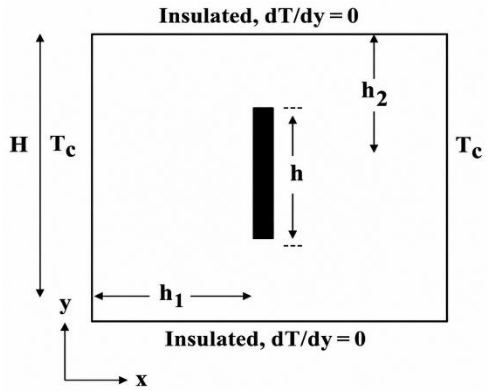


Fig. 3. Schematic diagram and geometrical specifications of the benchmark problem adopted from [21].

The heat transfer rate within the enclosure shown in *Fig. 3* was evaluated by calculating the average Nusselt number. The obtained results are summarized in *Table 1*.

Table 1. Comparison of the average Nusselt number obtained in the present study with the results reported by Nazari and Ramezani [21].

Enclosure Side Length (mm)	Grashof Number (Gr)	Present Study	Nazari and Ramezani [21]	Relative Error (%)
7.71	10310^{3}103	1.7926	1.738	3.0
16.6	10410^{4}104	2.1517	2.535	15.0

As can be observed from *Table 1*, the predicted average Nusselt number at Gr=10^3 shows excellent agreement with the results reported by Nazari and Ramezani [21], with only a minor deviation. For the higher Grashof number (Gr=10^4), a slightly larger discrepancy is observed; however, the overall agreement remains satisfactory. These results demonstrate the accuracy and reliability of the numerical methodology employed in the present study.

To further validate the numerical model, the streamline and isotherm contours for Gr=10^3 were compared with those reported by Nazari and Ramezani [21]. The comparisons are presented in *Figs. 4* and *5*, respectively.

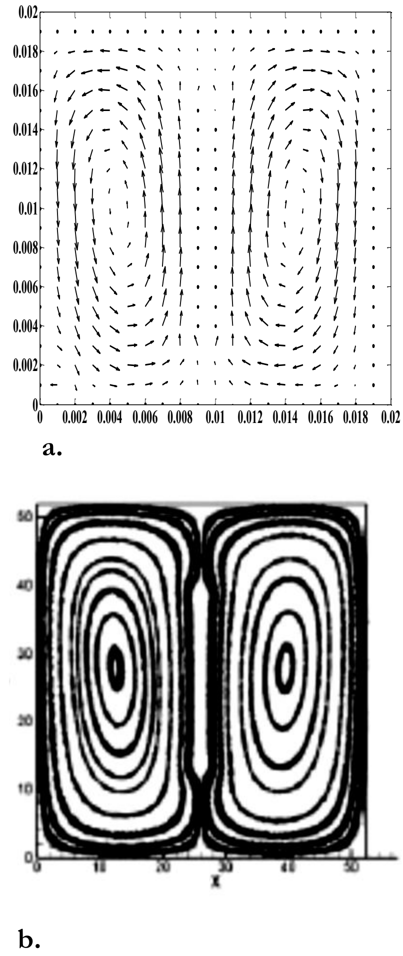
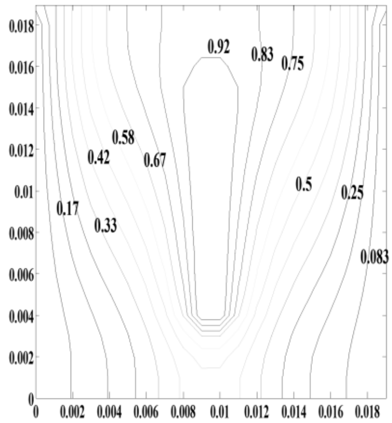
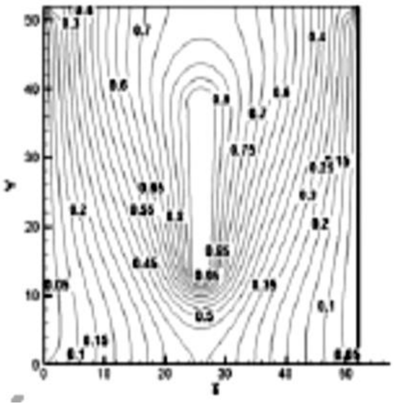


Fig. 4. Comparison of streamline contours at Gr = 10^3; a. present study, and b. Nazari and Ramezani [21].



a.



b.

Fig. 5. Comparison of isotherm contours at Gr = 10^3; a. present study, and b. Nazari and Ramezani [21].

To further assess the accuracy of the present numerical model, an additional validation was performed by comparing the predicted average Nusselt numbers for different enclosure geometries and working fluids with well-established empirical correlations available in the literature. The comparisons, summarized in *Tables 2* and *3*, cover the Rayleigh number range of $10^3 \leq Ra \leq 10^5$, corresponding to the conduction-dominated and laminar natural convection regimes illustrated in *Fig. 3*. The correlation for the average Nusselt number proposed by Ostrach [1] was employed as the reference and is given by *Eq. (5)*:

$$\overline{Nu} = \frac{q'}{q'_{conduction}} \rightarrow \overline{Nu} = \frac{q'}{kL\Delta T/\delta} = 0.364\frac{\delta}{L}Ra_L^{\frac{1}{4}}. \tag{5}$$

The average Nusselt number was then evaluated using both *Eq. (5)* and the present numerical code for different Rayleigh numbers. The corresponding comparisons for air and water are presented in *Tables 2* and *3*, respectively.

Table 2. Comparison of the theoretical and numerical average Nusselt numbers for air.

$\frac{L}{\delta}$	Rayleigh Number (Ra)	Theoretical Nusselt Number	Numerical Nusselt Number	Relative Error (%)
0.10	(1.63625908×10^7)	1.30	1.1083	15.4
0.20	(1.309007264×10^8)	2.19	1.8760	14.3
0.30	(4.417899516×10^8)	2.97	2.6610	10.4
0.40	(1.047205811×10^9)	3.68	3.3244	10.0
0.50	(2.04532385×10^9)	4.35	3.9590	9.0
0.60	(3.534319613×10^9)	4.995	4.5321	3.3

Table 3. Comparison of the theoretical and numerical average Nusselt numbers for water.

(Ra)	Prandtl Number (Pr)	Theoretical Nusselt Number	Numerical Nusselt Number	Relative Error (%)
1.802×10^5	0.75	1.0087	1.25	25.6
1.44×10^6	1.26	1.0960	1.13	3.1
4.86×10^6	1.71	1.4174	1.17	17.1
1.152×10^7	2.12	1.8762	1.65	11.5
2.25×10^7	2.51	2.3025	2.11	8.2
3.888×10^7	2.87	2.6755	2.50	6.7
6.174×10^7	3.23	3.0135	2.81	6.7
9.216×10^7	3.57	3.3313	3.11	6.7

As shown in *Tables 2* and *3*, the maximum discrepancy between the numerical predictions of the present study and the theoretical values does not exceed 12% for the investigated geometries. This level of agreement confirms the validity and satisfactory accuracy of the developed numerical model over the considered range of Rayleigh numbers.

4 | Results and Discussion

4.1 | Empty Cavity (Baseline Case)

First, an analysis of the cavity without baffles is performed. In the case of air (2 m × 0.02 m), with the applied temperature difference across the walls, the heat transfer value equals 7.35 W. As can be seen from *Fig. 6*, there exists thermal stratification in the cavity, which means that the hot fluid rises along the hot wall, moves towards the cold wall along the ceiling, cools down, and flows back towards the hot wall along the floor.

In the case of water (0.08 m × 0.008 m), due to the high thermal conductivity and volumetric heat capacity, the heat transfer is very high and equals 285.33 W. In the case of water, the temperature distribution (*Fig. 7*) is similar to that in the case of air. However, the temperature gradients in the neighborhood of the vertical walls are much higher, and the thermal boundary layer is very thin. This means that the Nusselt number for water is much higher than for air, as the Prandtl number of water is much higher (Pr = 7).

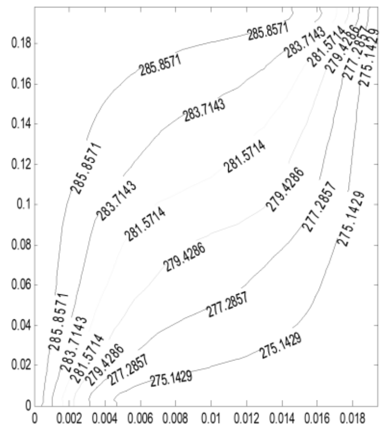


Fig. 6. Temperature distribution in the obstacle-free cavity filled with air.

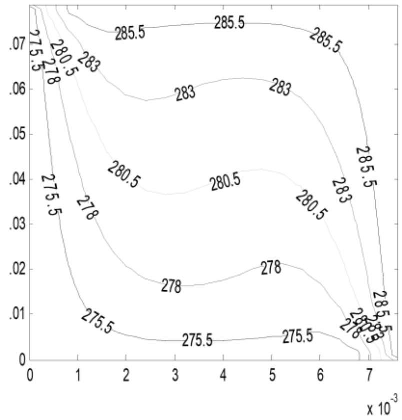


Fig. 7. Temperature distribution in the obstacle-free cavity filled with water.

4.2 | Cavity with Central Baffle (Open Valve)

Secondly, a thin solid baffle is positioned in the center of the cavity. In the case of air, the thickness of the baffle is 1 mm, and its height is 12.75 cm (around 58% of the height of the cavity). In the case of water, the thickness of the baffle is 0.4 mm, and its height is 5.1 cm (around 64% of the height of the cavity). An adiabatic boundary condition is assumed for the baffle to prevent heat transfer via conduction through the baffle.

From the figures (*Figs. 8* and *9*), it is evident that there is no effect of the baffle on the flow and heat transfer patterns. The fluid can still easily pass through the two openings above and below the baffle. The results from the temperature plots show that a small stagnant region forms directly behind the baffle, but the major convection region that runs along the whole height of the enclosure does not break up significantly, with only a slight distortion of the major flow pattern. The heat transfer rate of 7.19 W for air and 284.73 W for water is observed, with only a marginal reduction in the heat transfer rate of about 2.2% and 0.2%, respectively.

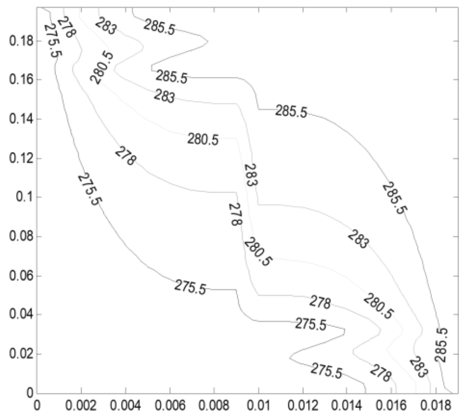


Fig. 8. Temperature distribution in the cavity with an obstacle and an open vent, filled with air.

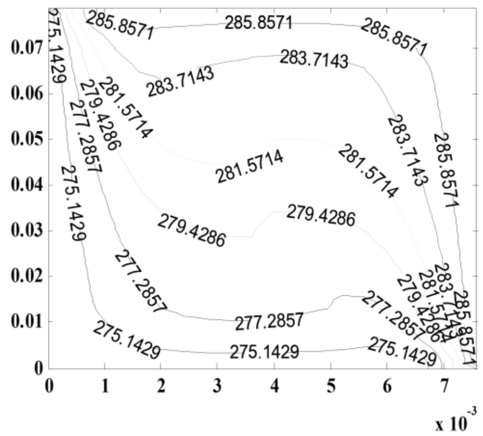


Fig. 9. Temperature distribution in the cavity with an obstacle and an open vent, filled with water.

4.3 | Cavity with Baffle and Closed One-Way Valve (Unidirectional Insulator)

In the final step, a lightweight flap is attached to the upper end of the baffle, completely blocking the upper flow passage. This flap is designed to allow flow only from right to left and completely closes in the opposite direction, simulating a one-way check valve.

4.3.1 | Velocity distribution

Fig. 10 and *Fig. 12* clearly reveal a dramatic change in the flow structure. The fluid circulation is now strictly confined to the lower half of the cavity, below the baffle. The upper half of the cavity becomes almost entirely stagnant. In this configuration, the hot fluid rising from the hot wall cannot pass over the top of the baffle to reach the cold wall. Instead, it becomes trapped behind the baffle, forming a small, weak recirculation cell in the bottom corner. This phenomenon, known as "fluid entrapment" [9], effectively decouples the hot and cold regions of the enclosure.

4.3.2 | Temperature distribution

Fig. 11 and *Fig. 13* corroborate the velocity field. In the upper half of the cavity, a very strong thermal stratification develops, with the vertical temperature difference reaching its maximum. Since there is no convective mixing in this region, heat transfer occurs predominantly by pure conduction across a thick stagnant fluid layer, which is significantly less efficient than convection. The isotherms in the lower half are densely packed near the hot wall, but the upper half exhibits nearly horizontal, parallel isotherms indicative of a conduction-dominated regime.

4.3.3 | Heat transfer rates

The overall heat transfer rate in this blocked configuration drops drastically. For air, it reaches only 0.3914 W, which represents a 95% reduction compared to the open-valve configuration (7.19 W). For water, the heat transfer rate drops to 22.3964 W, representing a 92% reduction compared to the open-valve configuration (284.73 W). This significant reduction demonstrates that the combined baffle and one-way valve system performs exceptionally well as a unidirectional thermal insulator: heat is readily transferred in the direction the valve opens. However, it is effectively suppressed in the opposite, blocked direction.

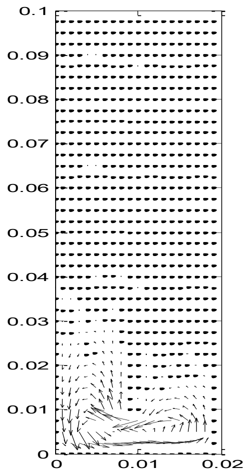


Fig. 10. Velocity distribution in the cavity with an obstacle and a closed vent, filled with air.

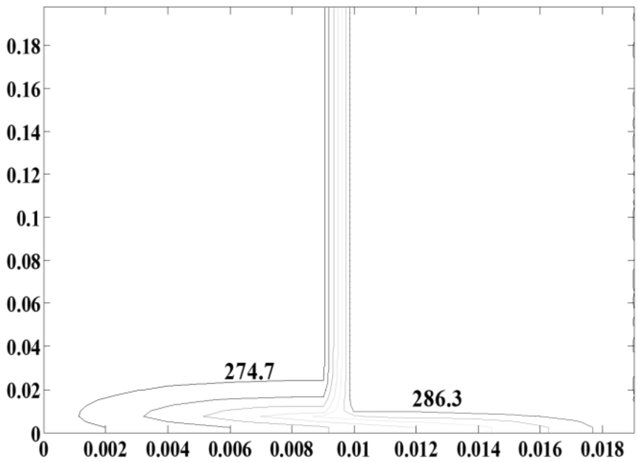


Fig. 11. Temperature distribution in the cavity with an obstacle and a closed vent, filled with air.

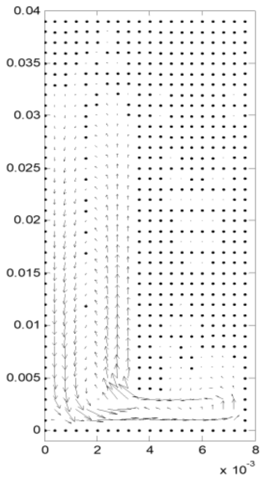


Fig. 12. Velocity distribution in the cavity with an obstacle and a closed vent, filled with water.

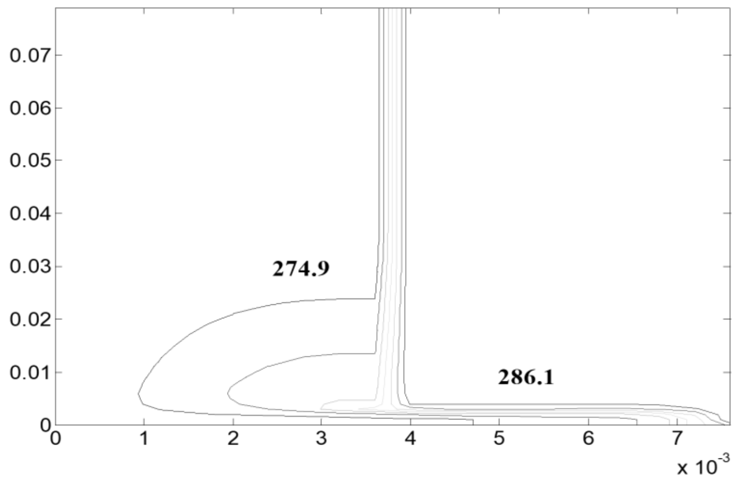


Fig. 13. Temperature distribution in the cavity with an obstacle and a closed vent, filled with water.

5 | Conclusion

A numerical analysis was carried out to study the natural convective heat transfer in a rectangular cavity with an internal baffle and a one-way valve. The important conclusions are as follows:

I. The effect of a small-sized internal baffle on the heat transfer alone is negligible as far as the fluid is allowed to circulate in the upper and lower passages. The decrease in heat transfer is less than 3% in both cases of air and water.

II. If the upper passage is blocked using a flap allowing flow in only one direction, then the process of circulation is disrupted. Hence, the thermal resistance is greatly increased. The heat transfer is reduced to up to 95% in the case of air and 92% in the case of water in comparison with the other direction.

III. The above design will essentially operate as a one-way thermal insulator because heat transfer through natural convection is easy from one side but not from the other side. This method of passive control does not require any form of power or mechanical movement; it just uses buoyancy force and the valve.

IV. The idea is applicable in construction, solar thermal energy, electronics, and any other system in which directionality of heat is desirable. Future research may consider the influence of various designs of valves and different properties of fluids under transient conditions.

Conflict of Interest

The authors declare no conflict of interest.

Data Availability

All data are included in the text.

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