Effect of Insulated up and Down Lid Motion on the Heat Transfer of a Lid-Driven Cavity with an Attached Fin

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This study investigates the effect of lid motion on the optimal characteristics a thin rectangular fin attached on the hot wall of a square lid-driven cavity with active vertical walls. The optimal fin position is studied for Richardson numbers of 0.1-10. The effect of mounting a rectangular fin with a thermal conductivity of 1 and 1000 on minimization and maximization of heat transfer through such cavity is explored. Mixed convection equations are solved using the control volume method with the help of the SIMPLER algorithm. The particle swarm optimization algorithm is used to determine the fin characteristics that minimizes or maximizes the heat transfer to the cold wall. The results show that optimal fin length and position is influenced by the position of the lid driven on the top or bottom of the cavity as well as lid velocity direction. The greatest reduction and increase in the Nusselt number are related to Richardson number of 0.1 with the bottom lid moving in the negative direction and Richardson number of 10 with the bottom lid moving in the positive direction, respectively.

Key words: Lid driven cavity, Optimal fin characteristics, Up lid moving, Down lid moving, PSO algorithm

1 Introduction

Mixed convection heat transfer in lid-driven cavities is an interesting subject of research with important applications in several fields of engineering [1-3]. In general, the studies in this area can be divided into three groups. The articles in the first group are focused on the heat transfer in such cavities with different temperature boundary conditions and different velocity magnitudes and direction on different surfaces. Notable among these studies is the research of Moallemi et al. [4], where they examined the characteristics of fluid flow and mixed convection heat transfer in a lid-driven cavity with insulated walls that was uniformly heated from the bottom. Ogut [5] studied the mixed convection in an inclined square cavity with a hot wall at the bottom, insulated side walls, and cold moving lid at the top.

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He concluded that in high Richardson numbers, cavity angle has an immense impact on the heat transfer through the cavity. He also discovered that heat conduction occurs at an angle of 180 degrees and in Richardson number of 10. Basak [6,7] analyzed the mixed convection in cavities with an insulated lid that are heated uniformly or non-uniformly from the bottom. The mixed convection in cavities with moving vertical walls was studied by Chamkha [8]. In that research, the cavity was considered to be filled with a hydromagnetic fluid; and its heat transfer was studied in the presence, and then in the absence of a heat source. Aydin [9] compared the aiding and opposing mechanisms of free and forced convection in a cavity. The numerical studies of Sharif [10], Morzynski [11], Freitas [12,13], and Iwatsu et al. [14], and Mohamad and Viskanta [15] on the effect of horizontal motion of a wall of the cavity showed that the ratio of the velocity of the moving lid to the buoyancy force is the most important parameter in the fluid flow. Oztop and Dagtekin [16] and Alleborn et al. [17] were the first researchers to study the cavities with two moving lids. The works of these researchers showed that lid motion direction plays a particularly important role in free convection in these cavities. The articles in second group involve the effect of fins on the control over mixed convection heat transfer, energy conservation, and the quality of thermal systems. Unfortunately, only a limited number of articles are concentrated on this particular branch of research. Notable among these articles are the works of Shi and Khodadadi [18-20], where they showed that mounting an oscillating fin on a stationary wall can facilitate the control over the fluid flow and heat transfer.

Oztop [21] studied the fluid flow and heat control using an insulated rectangular object. Dagtekin and Oztop [22] analyzed the cooling of electronic devices in a lid-driven cavity. Mahapatra et al. [23] investigated the problem of mixed convection heat transfer in a square cavity with two partitions installed on its insulated surfaces. They solved this problem using the commercial computational fluid dynamics software Fluent. This study found that at Richardson number of 1.0, the partition has a moderate impact on heat transfer, but at Richardson number of 0.1, this impact is much more pronounced. Mansutti et al. [24] developed a potential vector model for incompressible viscous flow in an inclined lid-driven cavity containing a block. Sun et.al [25] investigated effect of a triangular fin attached on the wall of a square lid driven cavity. They studied three positions for attachment of a triangular fin. They attached fin to the center of the hot wall firstly, then they attached it to the center of cold wall and insulated down wall. They investigated effect of fin attachment in three Richardson numbers 0.1, 1.0, 10.0 and for two directions of up lid driven. The most of decrease related to attachment of triangular fin to the center of hot wall.

The third group of works in this avenue of research includes the articles concentrated on optimization of such cavities or their conditions to maximize or minimize the heat transfer. Our exploration in this branch of the literature showed that not much work has been done in this area. One of the two most notable articles in this group is the study of Rahman et al. [26], where they searched for the optimal parameters of a lid-driven cavity with hot bottom wall and cold lid on the top. Their goal was to obtain the optimal flow and heat parameters in the presence of a solid block and with internal heat generation in order to maximize or minimize the heat transfer and the drag force (the force acting on the block), and they pursued this goal by assessing the best results obtained from several tests with different parameters. The second notable work in this group is the study of Lorenzini et al. [27], where they examined the optimal dimensions of a highly conductive hot fin mounted on the bottom wall of a square cavity where all surfaces except the cold moving lid were insulated. In this study, the ratio of fin area to cavity area was fixed and the fin was positioned in the center of the wall. The goal of this study was to calculate, for the rectangular fin, the length-to-width ratio that gives the highest heat transfer to the cold surface.

The optimal fin shape that increases the heat transfer through the described cavity was obtained by analyzing the placement of highly conductive fin for a range of Rayleigh and Reynolds numbers within the domain of the study.
A review of the past studies indicates that, so far, no attention has been paid to the notion of controlling the heat transfer by simultaneous optimization of the position and length of a fin in a lid-driven cavity for different fin thermal conductivities.
The present paper is focused on the control of mixed convection heat transfer, and more specifically, on the effect of the top and bottom lid motion velocity and direction on the optimal fin specifications. We also investigate the effect of Richardson number on the optimal specifications in the range of Ri=0.1 to Ri=10. to achieve the research objectives, in Section (5), we first validate the analysis for the cavity without the fin and then analyze the cavity with a triangular fin to determine the optimal fin position in section (6). Then, we compare the optimization outcomes with the results of Ref. [25]. In Section (6), we analyze the cavity with a thick rectangular fin with the same area as the fin analyzed earlier and compare the optimal position of this fin with that one.

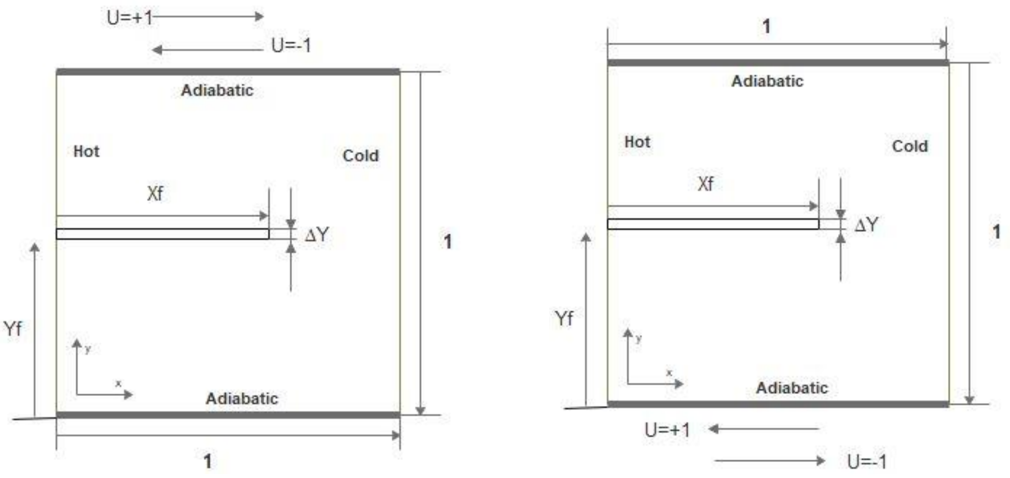
Finally, we determine the optimal position and length of very thin rectangular fins with the thermal conductivity of 1000 and 1, which are mounted to maximize and minimize the heat transfer, respectively. In the final section, we solve the problem with the motion of top and bottom lids considered in both directions. All optimizations of this study are performed using the particle swarm optimization algorithm. But given the effect of mixed free and forced convection, increasing nonlinearity of the problem, and the possibility of convergence to local optima, the results are obtained for 4 sample spaces, and ultimately the results obtained with 60 particles in 30 iterations are reported.

2 Problem definition

Consider the square cavity shown in Figures (1-a) and (1-b). The left wall is hot, the right wall is cold, and the horizontal walls are insulated and one of them is moving. The goal is to investigate the impact of velocity and direction of the motion of top and bottom (insulated) walls on the fin position and length that optimize (maximize or minimize) the heat discharged by the cold wall. Heat control with this method can contribute to the proper use of such cavities and also their energy conservation.

The cavity is filled with air with Prandtl number of 0.71. The analyses are conducted with Grashof number of 10^5. The analyses of velocity, however, are conducted using the Reynolds numbers of 100, 362.32 and 1000. Therefore, the problem is analyzed for Richardson numbers of 0.1, 1.0 and 10.0.

In a research conducted by Azimifar et al. [28], they found that as the number of fins increases, the algorithm accuracy decreases and it becomes more likely to converge to the local minima. Thus, to obtain a global minimum, they investigated the performance that can be achieved with a higher number of particles and iterations and ultimately found that better results can be obtained by the use of 60 particles instead of previously recommended 20-40 particles. In the present study, heat transfer optimization is performed for only one fin, but given the presence of moving lid as well as free convection, the mixed convection is much more complex. Thus, despite the presence of only one thin rectangular fin, to determine the optimal length and position of the fin, each optimization problem is solved in 4 modes with 20, 40, 60 and 80 particles in 20, 20, 30 and 40 iterations respectively.



a) b)

Figure 1 Schematic figure of lid-driven cavity with attached fin on the hot wall
a)up lid moving b)down lid moving

3 Governing equations

The governing equations consist of continuity, momentum and energy equations for the fluid and the energy equation for the solid. These equations are solved based on the assumption of two-dimensional steady incompressible fluid and using the Boussinesq approximation. After applying the following assumptions and normalizing the equations with the following parameters, the dimensionless form of the equations will be as follows:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0.0 \tag{1}$$

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\frac{\psi}{\mathrm{Re}}\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right) \tag{2a}$$

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\frac{\psi}{\mathrm{Re}}\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)+Ri\theta \tag{2b}$$

$$U\frac{\partial T}{\partial X}+V\frac{\partial T}{\partial Y}=\frac{R_k}{pr\,\mathrm{Re}}\left(\frac{\partial^2 T}{\partial X^2}+\frac{\partial^2 T}{\partial Y^2}\right) \tag{3}$$

Non-dimensional parameters are as follows:

$$\theta=\frac{T-T_0}{T_h-T_c},\ \mathrm{Re}=\frac{U_0 L}{\nu},\ \mathrm{Pr}=\frac{\nu}{\alpha},Y=\frac{y}{L},X=\frac{x}{L},\ V=\frac{v}{U_0},\ U=\frac{u}{U_0},Gr=\frac{g\beta\left(T_h-T_c\right)H^3}{\nu^2} \tag{4}$$

$$P=\frac{p}{\rho U_0^2},\ Ri=\frac{Gr}{\mathrm{Re}^2}=\frac{g\beta(T_w-T_\infty)L^3/\nu^2}{(\frac{U_\infty L}{\nu})^2}$$

4 Optimization problem

4.1 objective function and design parameters

In the optimization problem of this study, fin position $YP_{f,i}$, fin length $XP_{f,i}$ are unknown, while the average Nusselt number of the cold wall $\varpi \overline{Nu}_d$ is known.
To solve this optimization problem, the unknown coordinates ($XP_{f,i}$, $YP_{f,i}$) for the i-th fin on the hot wall is selected from a continuous space. To solve the governing equations by finite volume method, these fins are linked to the closest node on the mesh.

In the displacement subprogram, the applied fins are assigned with a dimensionless viscosity and thermal conductivity set equal to infinity and R_k respectively. The displacement subprogram is then solved for solid-fluid interaction.

$$XP_{f,m} = \left\{ XP_{f,1}, XP_{f,2}, ..., XP_{f,i}, XP_{f,M_f} \right\} \tag{5}$$

$$YP_{f,m} = \left\{ YP_{f,1}, YP_{f,2}, ..., YP_{f,i}, YP_{f,M_f} \right\} \tag{6}$$

Here, $XP_{f,i}$, $YP_{f,i}$ denoting respectively the fin length, position

$$\overline{Nu}c = \frac{1}{H} \sum_{nd=1}^{R} Nu_{d,nd} \Delta Y \qquad where\; Nu_{d,nd} = \left.\frac{\partial t^*}{\partial X}\right)_{d,nd} \quad at\; cold\; wall\;\; for\; without\;\; fin \tag{7}$$

$$\overline{Nu}e = \frac{1}{H} \sum_{nd=1}^{R} Nu_{e,nd} \Delta Y \qquad where\; Nu_{d,nd} = \left.\frac{\partial t^*}{\partial X}\right)_{e,nd} \quad at\; cold\; wall \tag{8}$$

Where $Nu_{d,nd}$ and $Nu_{e,nd}$ are the desired and estimated Nusselt numbers for the cold wall.
Here, R is the number of nodes on the cold wall.
The solution of this problem is obtained by minimizing an objective function expressed by Eq. (9):

$$GO(XP_{f,m}, YP_{f,m}) = \left| \varpi \overline{Nu}_d - \overline{Nu}_e (XP_{f,m}, YP_{f,m}) \right| \tag{9}$$

Where $\overline{Nu}_d$ is the average Nusselt number for the cold wall without fin attachment to the hot wall, and ϖ is a constant. For heat transfer minimization objective, ϖ <1 and $\varpi \overline{Nu}_d$ = 1, and for heat transfer maximization objective function ϖ >1 and $\varpi \overline{Nu}_d = 1000$

4.2 optimization algorithm

In recent decades, Particle Swarm Optimization (PSO) algorithm has become extremely popular among many research communities mainly because of its simplicity as well as powerful search capabilities. PSO generates a random population of solutions and then improve them iteratively by searching for the best solution among the population. In this algorithm, each bird is called a particle. In the course of optimization, particles fly, typically at the same velocity, toward the best position found collectively by the swarm.

In each iteration, velocity of each particle will be updated based on its current velocity, the best position found by that individual particle, and the best position found by all particles collectively [29].
For particle i, the position, velocity and the best position are expressed as follows:

$$\mathbf{XI}_i(iter) = [xi_{i1}(iter), xi_{i2}(iter), ..., xi_{iN}(iter)] \tag{10}$$

$$\mathbf{VI}_i(iter) = [vi_{i1}(iter), vi_{i2}(iter), ..., vi_{iN}(iter)] \tag{11}$$

$$\mathbf{PI}_i(iter) = [pi_{i1}(iter), pi_{i2}(iter), ..., pi_{iN}(iter)] \tag{12}$$

Also, the best position found at each iteration by all particles is expressed as:

$$\mathbf{PI}_g(iter) = [pi_{g1}(iter), pi_{g2}(iter), ..., pi_{gN}(iter)] \tag{13}$$

Position and velocity of particles are updated by the following equations [30-31]:

$$\begin{aligned} vi_{ij}(t+1) &= \omega(t) vi_{ij}(t) + \\ &C_1 r_1 \left[pi_{ij}(t) - xi_{ij}(t) \right] + C_2 r_2 \left[pi_{gj}(t) - xi_{ij}(t) \right] \end{aligned} \tag{14}$$

$$xi_{ij}(iter+1) = xi_{ij}(iter) + vi_{ij}(iter) \tag{15}$$

Where C_1 and C_2 are positive acceleration constants also known as cognitive and social scaling constants respectively. r_1 and r_2 are random values between zero and one. Domain Nt and C_1 and C_2 are defined as [32]:

$$C_1 + C_2 \le 4 \tag{16}$$

$$20 < Nt < 80 \tag{17}$$

Where N and Nt are the number of variables and particles respectively.
Inertia weight ω is the parameter controlling the effect of previous velocity of a particle on the updated velocity. The best approach in regard to this parameter is to start with a large ω to accelerate and improve the global search, and then decrease it gradually to improve convergence at local scale. To prevent jumping around the global optimum, ω is typically set to decrease linearly in the range [0.4,0.9] or [0.1,0.7]; but in this article, ω is set to decrease at each iteration with a constant multiplier defined as follows:

$$\omega(\mathrm{t}+1) = 0.99 * \omega(\mathrm{t}) \tag{18}$$

Where $t_{\max}$ is the maximum iteration.
Generally, $XI_i(t)$ and $VI_i(t)$ should remain within the ranges $[XI_{\min}, XI_{\max}]$ and $[-VI_{\max}, VI_{\max}]$, where $VI_{\max}$ and $XI_{\max}$ are the upper bounds of design variables and particle velocity respectively. In this study, $VI_{\max}$ is defined as follows:

$$\mathbf{VI}_{\max}(t) = 0.1 \times (XI_{\max} - XI_{\min}) \tag{19}$$

In this study, optimization with PSO consists of the following steps:

1- Initializing the position and length of fin and particle velocity over the entire search space
2- Setting the fins to the closest point on the displacement subprogram of mixed convection
3- Obtaining the objective function value according to particle positions using Eqs.(1-3)
4- Updating the best individual position $\mathrm{PI}_i(t)$ and global position at each iteration

5-Updating the particle velocity based on the previous values.
6-Decreasing the ω-value as described.
7-Repeating steps 2 to 6 until a stopping condition is met. (The difference between global best objective function and objective function in each iteration to be less than 10^{-9}. Also, maximum iteration set to final iteration)
8-Printing the global optimum $P_g(t)$ (optimum length and position)

5 Results

5.1 Validation of the direct solution (Fin-less cavity)

As shown in Figure (2), we consider a square cavity, where the left wall is hot, the right wall is cold, and the top and bottom walls are insulated and top wall can move in the positive or negative direction. The above figure is used for grid independence analysis and validation.
In the mixed convection heat transfer, Richardson number (Ri) varies in the range of 0.1 <Ri <10. Therefore, grid independence analysis and validation were carried out for Ri =0.1, 1, and 10. For validation, we utilized the results of research conducted by Sun et al. [25]. For grid independence analysis, we began the analysis with 80 nodes and increased the number of nodes step by step. Finally, we plotted the average Nusselt number versus the number of nodes. This plot showed that when Ri =0.1, increasing the number of nodes beyond n=140 makes no significant change in the results. For other Richardson numbers, this threshold was found to be n=120. Therefore, the rest of the analyses were carried out using 140 nodes when Ri =0.1, Ri =1.0 and Ri =10. The grid independence analysis plots for two lid velocity directions and three Richardson numbers are provided in Figures (3-a) to (3-c).
To validate the accuracy of the used computer code and equations, the results needed to be compared with the results provided elsewhere.
In Table (1), the average Nusselt numbers obtained in the present study are compared with these were reported by Sun et al. [25]. As shown in Table (1), the deviation of our results from the results of Sun et al. [25] remains lower than 3%, and this indicates the appropriate accuracy of the numerical solution.

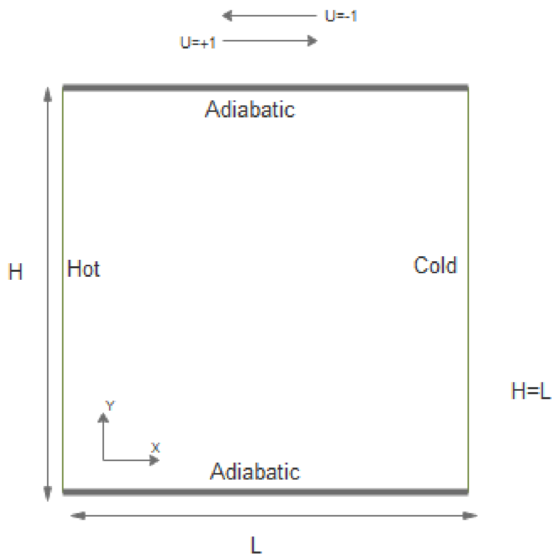


Figure 2 Schematic diagram of lid-driven cavity with mixed convection

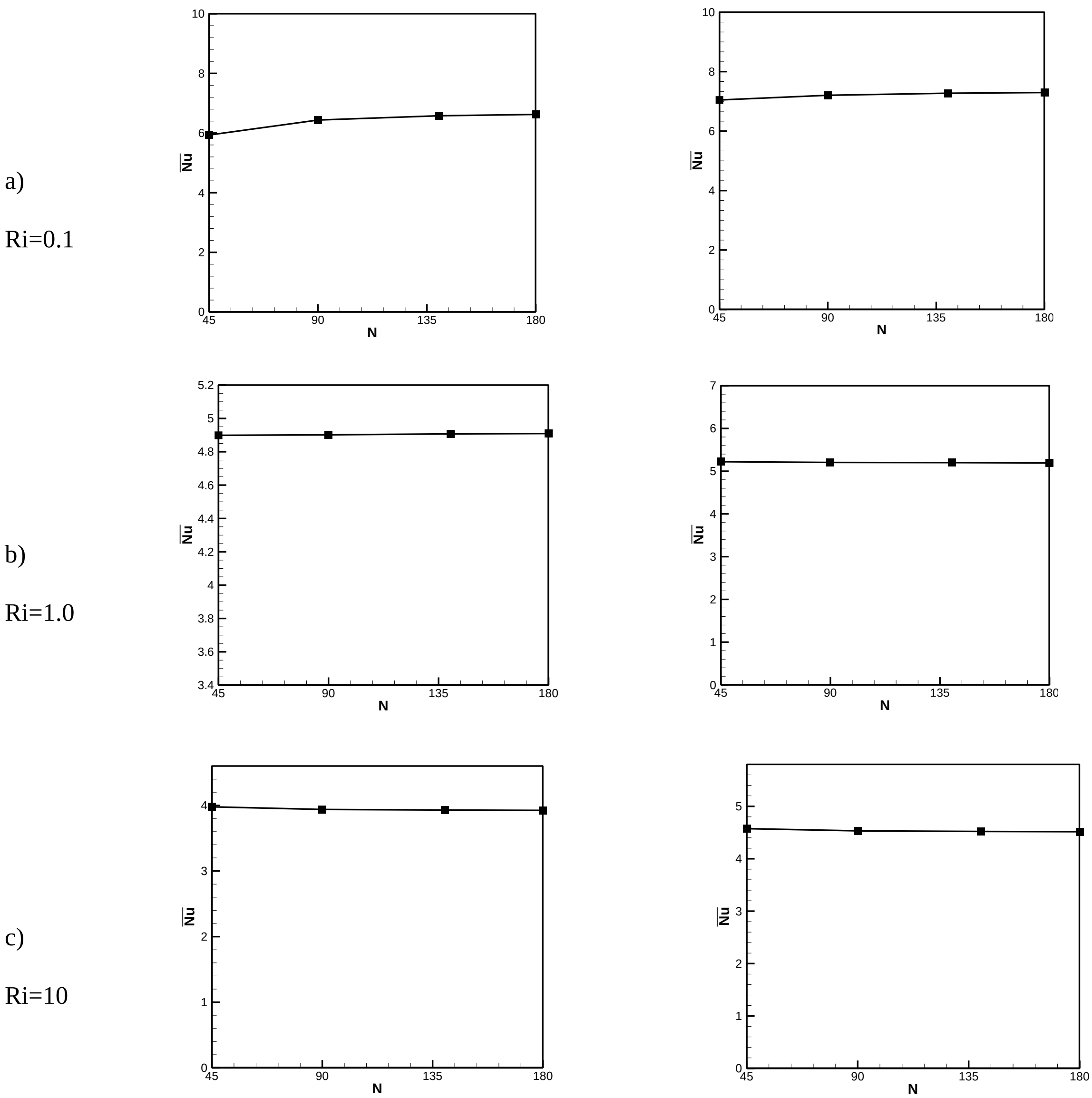


Figure 3 Average Nusselt number versus grid node number for: a) Ri=0.1 b) Ri=1.0 and c) Ri=10.0 when U has a negative direction (left) and U has a positive direction (right)

Table 1 Comparison of the average Nusselt numbers in the present study and in the research of Sun et al.[25]

	U=+1			U=-1		
Ri	0.1	1.0	10.0	0.1	1.0	10.0
Present study	7.25	5.20	4.53	6.57	4.89	3.94
Sun. et.al[25]	7.46	5.29	4.61	6.75	5.01	4.01
Error%	2.81	1.70	1.73	2.66	2.39	1.74

5.2 Cavity with a triangular fin on the center of the hot wall

In this section, the problem consists of a square cavity with a triangular fin mounted on the left wall are considered as seen in Figure (4) the left wall is hot, the right wall is cold, and the horizontal walls are insulated. The top horizontal wall moves at the velocity U in the positive or negative direction of the X-axis.

We used the above figure for validation. In the mixed convection heat transfer, Richardson number (Ri) varies in the range of 0.1 <Ri <10. Thus, validation was carried out for Ri =0.1, 1, and 10 using the results of Sun et al. [25]. The used computer code and equations were validated using the results provided in this reference. Table (2) compares the average Nusselt numbers obtained in the present study and those provided in the study of Sun et al. [25].
As an example, the flow patterns obtained for the cavity the lid moving in the positive direction at Richardson number 0.1 are compared with the results of Sun et al.[25] (see Figure (5)). Also isothermal lines for two works are compared in Figures (6). For positive direction of up wall motion at Richardson 0.1.

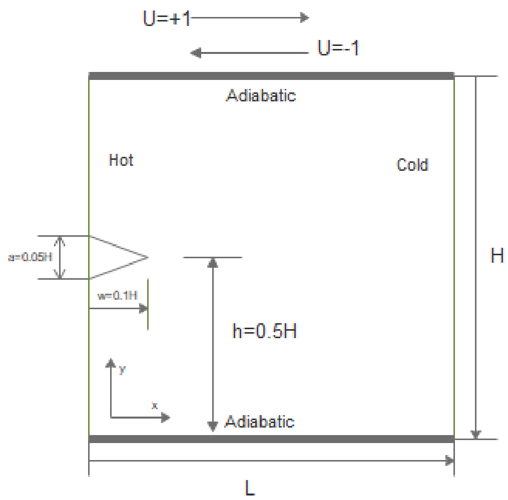


Figure 4 Schematic figure of the lid-driven cavity with an attached triangular fin on the hot wall

Table 2 Comparison of the average Nusselt numbers in the present study and in the research of Sun et al[25]. with an attached triangular fin on the center of hot wall

	U=+1			U=-1		
Ri	0.1	1.0	10.0	0.1	1.0	10.0
Present study	6.14	4.68	3.56	5.87	4.75	4.15
Sun.et.al[25]	6.16	4.75	3.64	5.50	4.57	4.12
Error%	0.32	1.47	2.19	6.72	3.93	0.72

a)

Ri=0.1

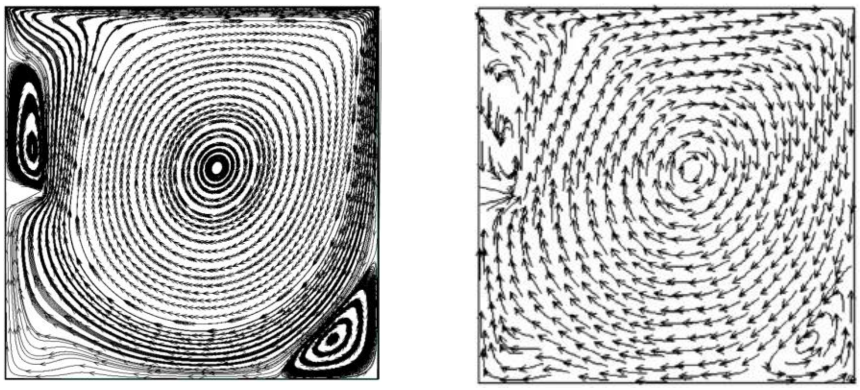


Figure 5 flow patterns in the cavity the lid moving in the positive direction in the present study (left) and in the study of Sun et al[25]. (right) at Ri=0.1

a)

Ri=.1

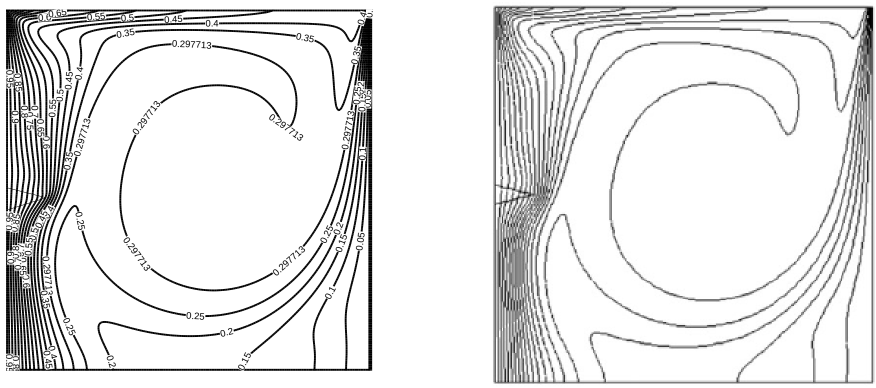


Figure 6 Isothermal lines in the cavity the lid moving in the positive direction in the present study (left) and in the study of Sun et al [25]. (right) at Ri=0.1

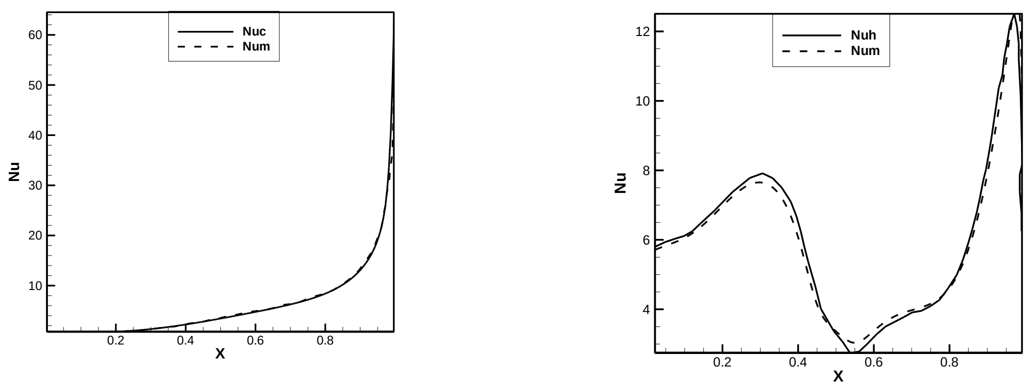


Figure7 the local Nusselt numbers on the hot (right wall) and cold (left wall) walls for the cavity with triangular fin and positive lid motion direction are compared for Richardson number 0.1 with the results of Sun et al. [25].

In Figures (7), the local Nusselt numbers on the hot and cold walls for the cavity with triangular fin and positive lid motion direction are compared for Richardson number 0.1 with the results of Sun et al. [25]. In these diagrams, the Nusselt numbers reported in the mentioned reference are denoted by *Num* and the Nusselt numbers obtained in this study for the cold and hot walls are represented by *Nuc* and *Nuh*, respectively.
As is evident the described numerical model is well able to model the triangular fin with any thermal conductivity coefficient, and thus, there is no need to solve the solid and fluid equations separately.

6 Optimization

6.1 Validation of optimization code

To validate the optimization solution, we consider the problem discussed in Section (5.2). With the fin dimensions considered to be the same as the triangular fin, the goal is to determine the fin position that minimizes the heat transfer through the cavity. This optimization was performed for both positive and negative lid motion and Richardson numbers of 0.1, 1.0 and 10. As shown in Table (3), when the lid moves in the positive direction, for all Richardson numbers, the optimized position of the fin with a thermal conductivity of 1 is not different from the un-optimized position. But when the lid moves in the opposite direction, with the conflict of free and forced convection and the dominance of forced convection in Richardson numbers of 0.1 and 1.0, the top of the hot wall becomes the optimal fin position. Under the dominance of forced convection, the highest velocities take place just above the hot wall and near the moving lid, which means a short fin located in this area to have the greatest impact on the heat transfer (because the fin in this position block vertical velocities). Also, in Richardson number of 1.0, when forced convection is still somewhat effective, the optimal fin position remains in the top section of the hot wall. But in the Richardson number of 10.0, when the free convection is dominant, it is the center of the hot wall that becomes optimal fin location. As expected, with the dominance of free convection, the optimal fin position become independent of the lid motion velocity. Table (3) presents the decrease in the heat transfer relative to the unoptimized state.

To assess the validity of the solution provided by the optimization code, in Figure (8), the average Nusselt number on the cold wall of the cavity with the lid moving in the negative direction is plotted against the fin position at (0.1-0.9) on the hot wall for three Richardson numbers of 0.1, 1.0 and 10. As can be seen, the best fin position in this diagram matches the position determined by the optimization algorithm.

Table 3 Comparison of the optimization and non-optimization average Nusselt numbers with an attached triangular fin on the optimal place and center of the hot wall

	U=+1				U=-1			
Ri	Nu_{opt}	$Nu_{opt}/Nu_{without\_fin}$	$Nu_{opt}/Nu_{\_non\_opt}$	Y_{opt}	Nu_{opt}	$Nu_{opt}/Nu_{without\_fin}$	$Nu_{opt}/Nu_{\_non\_opt}$	Y_{opt}
0.1	5.847	19.35	6.52	0.466	5.773	6.19	5.97	0.91
1.0	4.751	8.63	0.021	0.567	4.271	12.65	8.73	0.93
10.	4.146	8.45	0.096	0.567	3.567	9.46	0.196	0.456

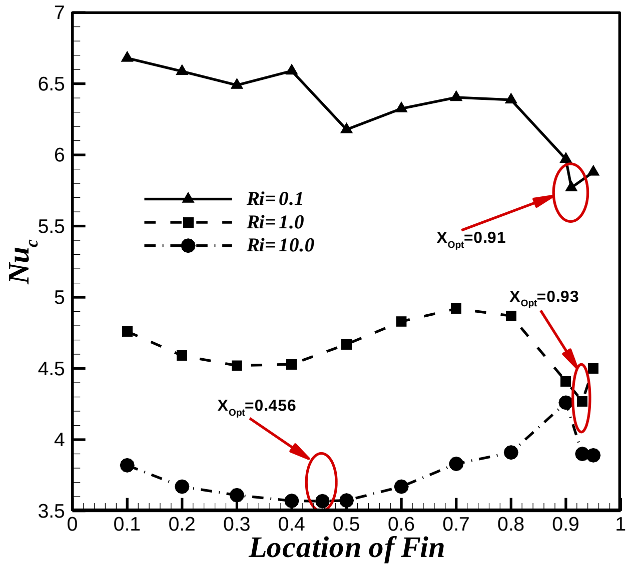


Figure 8 The average Nusselt number on the cold wall of the cavity with the lid moving in the negative direction is plotted against the fin position at (0.1-0.9) on the hot wall for three Richardson numbers of 0.1, 1.0 and 10

6.2 Optimal position of the Rectangular fin for heat transfer minimization

Consider the square cavity of Figure (9), where the horizontal walls are both insulated, the left wall is hot, and the right wall is cold, and the moving lid is positioned at the top. The goal is to investigate the impact of velocity and direction of the motion of the top lid on the position of a rectangular fin of known length and thickness (the area that give the rectangular fin the same area as triangular fin discussed in the previous section (section 5.2)) that minimizes the heat discharged by the cold wall. The cavity is filled with air with Prandtl number of 0.71, and all analyses are conducted with Grashof number of 10^5.

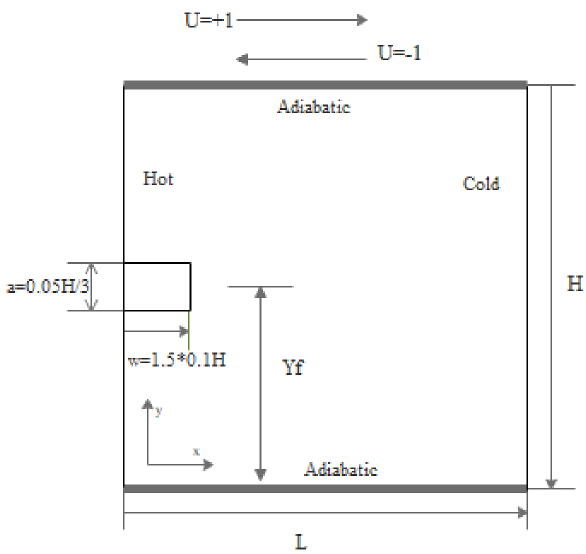


Figure 9 Schematic figure of the lid-driven cavity with an attached thick rectangular fin on the hot wall

Table 4a Results of optimization of rectangular fin and comparison with validation results for the cavity with the lid moving in the positive direction

Ri	$\overline{Nu}_{opt}$	$Y_{opt,rec}$	$Error_{opt/withoutfin}$	$Error_{opt\_rec/opt\_tri}$
0.1	5.429	0.436	33.542	7.14
1.0	4.586	0.536	13.388	3.472
10.0	4.009	0.551	12.995	3.30

Table 4b Results of optimization of rectangular fin and comparison with validation results for the cavity with the lid moving in the negative direction

Ri	$\overline{Nu}_{opt}$	$Y_{opt,rec}$	$Error_{opt/withoutfin}$	$Error_{opt\_rec/opt\_tri}$
0.1	5.476	0.91	19.978	4.95
1.0	3.993	0.93	22.464	6.50
10.0	3.423	0.447	15.103	4.03

The analyses of velocity are conducted using the Reynolds numbers of 100,362.32, and 1000. The problem is solved for Richardson numbers of 0.1, 1.0 and 10.0.

As shown in Table (4a) and (4b), irrespective of the lid motion velocity and for all Richardson numbers, the rectangular fin with a thermal conductivity ratio of 1 has the same optimal position as the triangular fin of equal area, but makes a greater reduction in heat transfer. The best heat transfer reduction achieved with this fin is 33.5% at the Richardson number of 0.1.

7 Optimal position and length of a rectangular thin fin

The results obtained from section 6.2 are shown a rectangular fin can be have a better performance therefore in the final section to find the optimal length and position of rectangular fin for controlling the mixed convection. As told in definition problem section, In the present study, heat transfer optimization is performed for only one fin, but given the presence of moving lid as well as free convection, the mixed convection is much more complex. Thus, despite the presence of only one thin rectangular fin, to determine the optimal length and position of the fin, each optimization problem is solved in 4 modes with 20, 40, 60 and 80 particles in 20, 20, 30 and 40 iterations respectively.

Figures (10a-10d) summarize the average Nusselt numbers obtained by solving the optimization problems with 20, 40, 60, and 80 particles and maximum iterations 20,20,30 and 40 for each problem respectively to determine the optimal length and position of a thin fin (on the hot wall) for the cavities with bottom lid moving in the positive and negative directions for decrease(10a) and increase(10b) of heat transfer from the cavity, also for cavities with top lid moving in positive and negative directions for decrease (10c) and increase (10d) of heat transfer. As seen in Figures (10a-10d) the average Nusselt numbers for the 60 and 80 particles are same, therefore 60 particles and maximum 30 iterations are selected for next sections.

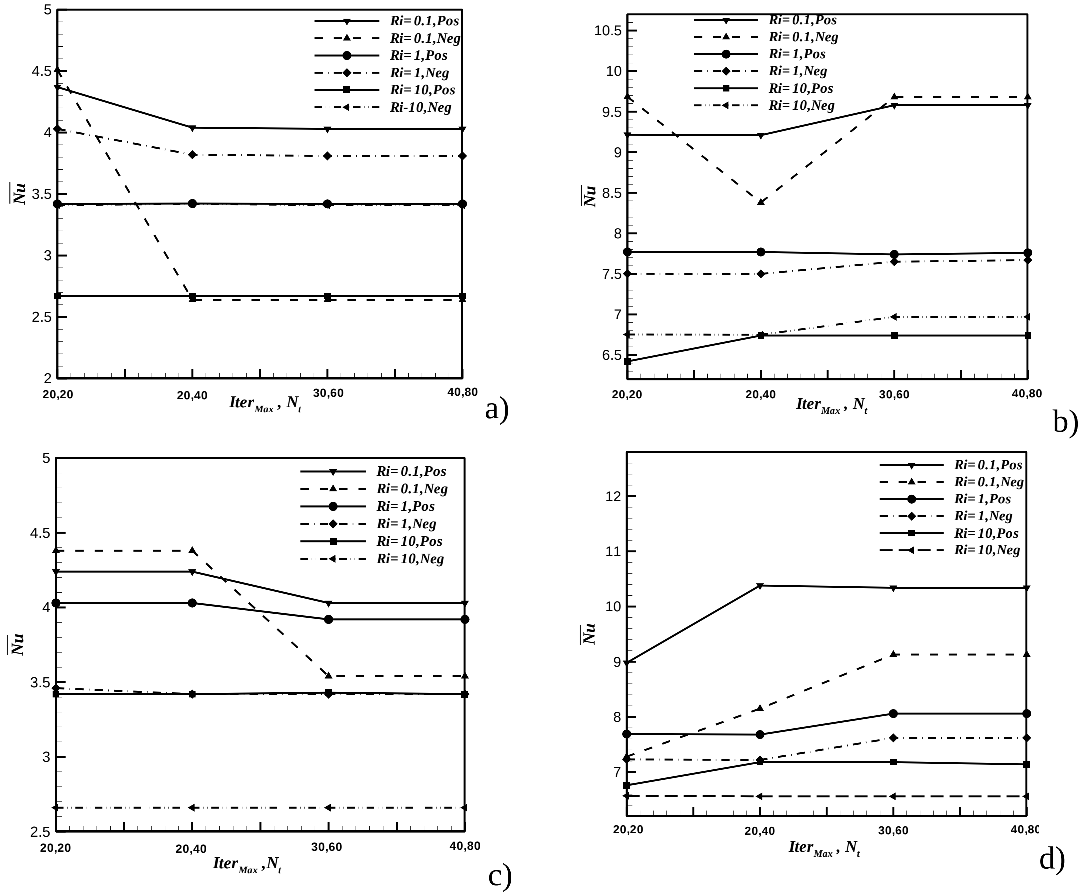


Figure10 effect of particles and maximum iterations on a,c) decrease of the heat transfer for a)down lid driven c)up lid driven, b,d) increase of heat transfer b)down lid driven d) up lid driven

8 Impact of lid motion direction on the optimal length and position of rectangular thin fin

Consider the square cavity depicted in Figure (1), where the left and right walls are hot and cold respectively, the horizontal walls are both insulated, and the top and bottom walls can move in two directions moving. The goal is to determine the impact of velocity and direction of the motion of the top lid on the length and position of a fin (on the hot wall) that optimize (minimize or maximize) the heat discharged by the cold wall. Like before, the cavity is filled with air with Prandtl number of 0.71, all analyses are conducted with Grashof number of 10^5, and the analyses of velocity are conducted using the Reynolds numbers of 100,362.32, and 1000. Also as before, the problem is solved for Richardson numbers of 0.1, 1.0 and 10.0. For the comparison of average Nusselt numbers in two cases with attached fin and without fin relative error as Eq.(20) is used:

$$\mathrm{Err} = \left| \frac{\overline{\mathrm{Nu}}_{\mathrm{fin}} - \overline{\mathrm{Nu}}_{\mathrm{no\text{-}fin}}}{\overline{\mathrm{Nu}}_{\mathrm{no\text{-}fin}}} \right| \times 100 \qquad (20)$$

In the cavity with the top lid moving in the negative (positive) direction, the resulting Nusselt numbers should be equal to those obtained for the cavity with the bottom lid moving in the positive (negative) direction. However, there are some disparities in the results regarding the optimal fin length and optimal fin position in the aforementioned cavities.

The cause of these disparities is the start of the free convection boundary layer over the hot wall exposed to forced convection with moving lid at the bottom and the end of the boundary layer of the hot wall exposed to forced convection with moving lid at the top.

As suggested by numerous papers, there are two general mechanisms that oppose flow motion and increase surface on the hot wall of a cavity. Perfect understanding of the flow and heat mechanism in lid-driven cavities can facilitate the prediction of optimal fin position for maximizing or minimizing the heat transfer.

Therefore, the main point of interest is that when the bottom lid moves in the positive direction or the when the top lid moves in the negative direction, free and forced convections act in the opposite direction, and the heat transfer in these cavities is lower than the cavities where free and forced convections act in the same direction.

8.1 Impact of lid motion on the optimization of rectangular fin for heat transfer minimization

For heat transfer reduction in Richardson number of 0.1, forced convection is more important than free convection. The optimal lengths and positions of the rectangular fin and the percent of the depression of heat transfer in down lid driven cavity for two x-directions are shown in Table (5), best reduction related to Ri=0.1 in negative direction of down lid. Also The optimal lengths and positions of the rectangular fin and the percent of the depression of heat transfer in up lid driven cavity for two directions are shown in Table (6), best reduction related to Ri=0.1 in negative direction of down lid.

The presence of a block against the lid velocity (i.e. positioning the fin near the moving lid) reduces the heat transfer by weakening the effect of lid motion on the main flow (see Figures (11) in right picture). When the fin length completely blocks the moving lid, the mode of heat transfer changes from the mixed convection to the free convection. Thus, since we use the Grashof number of 10^5 for all Richardson numbers, in this state, the flow contour lines (Figures (11) and (12) in the right of picture for the cavities with the bottom and top moving lid, respectively) are similar to those observed for free convection.

Table 5 optimization Nusselt numbers in a cavity and optimal places and sizes of an attached rectangular thin fin on the hot wall for decrease of heat transfer with down lid driven cavity in two directions in N_t =60

Ri	direction	$\overline{Nu}_{30,60}$	(x,y)	$Err\%$
0.1	+	4.03	(0.99,0.017)	39.004
1.0	+	3.42	(0.99,0.607)	30.20
10	+	2.67	(0.99,0.519)	32.06
0.1	-	2.64	(0.442,0.048)	63.711
1.0	-	3.81	(0.366,0.110)	26.73
10.0	-	3.41	(0.865,0.610)	24.64

Table 6 Optimization Nusselt numbers in a cavity and optimal places and sizes of an attached rectangular thin fin on the hot wall for decrease of heat transfer with up lid driven cavity in two directions in N_t =60

Ri	direction	$\overline{Nu}_{30,60}$	(x,y)	$Err\%$
0.1	+	4.03	(0.98,0.981)	44.41
1.0	+	3.92	(0.99,0.917)	24.61
10	+	3.43	(0.949,0.388)	24.28
0.1	-	3.54	(0.499,0.986)	46.11
1.0	-	3.42	(0.99,0.398)	30.06
10.0	-	2.66	(0.922,0.480)	32.48

But we need to also check the possibility of the cavity with mixed convection and an optimally sized and positioned fin achieving a lower heat transfer than the cavity with free convection. The results obtained for the cavity with the bottom lid moving in the negative direction showed that, despite the free and forced convection acting in the same direction, installing a fin with a length of Y/2 (where Y is the total length of cavity) in a position slightly away from the moving lid can make nearly 50% greater reduction in heat transfer than what is achievable with free convection with Grashof number of 10^5.

According to the results of flow function in Figure (11) (in the left of the picture), the vertical eddies have emerged perpendicular to the main flow direction and reduce the heat transfer. As is clear from the flow pattern, the movement and collision of cool fluid have created a large return flow between the two main vertical flows, of which one is spinning near the cold wall and the other is spinning near the hot wall. This intermediary flow influences the area above the cold wall where free convection velocities are higher and forced convection also acts in the same direction. In this area, this intermediary flow runs upward and opposes the direction of heat transfer aiding motion, therefore reducing the heat transfer.

Thus, in Richardson number of 0.1(for a bottom lid driven in negative direction), the presence of a half block near the moving lid facilitate the heat transfer reduction. Thus, to reduce the heat transfer in Richardson number of 0.1, where the lid motion effect is dominant, a fin should be placed near the moving lid.

For the cavity with the top lid moving in the negative direction, the lid motion moves the nearby cold flow toward the hot wall, but as it moves, this flow hits the block and returns without passing by the hot wall (see Figure (12) in the left of the picture). Thus, the hot flow created before the block is not affected by the lid velocity and moves upwards until it hits the cold flow and then returns downwards and contacts the cold wall in its lower section (the flow rennin downstream is because of diminishing impact of the lid velocity).

Using a shorted fin also results is increased heat transfer by expanding the right side eddy.

As shown in Figure (13a) in the left of the picture, for the cavity with the bottom lid moving in the negative direction, in Richardson number of 1.0, much like in Richardson number of 0.1, the optimal fin is positioned near the moving lid and has a length of more than 1/3Y (where Y is the total length of cavity).

We can see that as Richardson number increases and the effect of free and forced convection heat transfer become more important, optimal fin becomes shorter and its distance from the moving lid increases. By using a higher than optimum fin length causes a denser flow near the cold wall and increases the average Nusselt number. Conversely, using a less than optimum fin length prevents the formation of return flow and increases the boundary layer on the hot surface. The flow contour lines illustrated in Figure (13) in the left of the picture show the described behavior in this case.

As shown in Figure (14a) in the right of the picture, for the cavity with the up lid moving in the positive direction, in Richardson number of 1.0, much like in Richardson number of 0.1, the optimal fin is positioned near the moving lid but its length completely obstructs the lid. In the cavity with the bottom lid moving in the positive direction and the top lid moving in the negative direction, in Richardson number of 1.0, free and forced convection act in opposite direction. Here, the effect of forced convection is limited to downward motion in the bottom third and upward motion in the top third of the cavity. In most of the cavity (the top two-thirds when the lid drivenis at the bottom and the bottom two-thirds when the lid driven is at the top), lid velocity has a reduced effect, so formation of any block that would result in many eddies will cause flow dissipation and reduced heat transfer.

As shown in Figure (13a) in the right of the picture, for the cavity with the moving lid at the bottom, a block is positioned at Y=0.6, and as seen in Figure (14a) in the left of the picture for the cavity with moving lid on the top, a block with maximum length is positioned at Y=0.4.

As can be seen in the flow contour lines of these cavities (Figures (14) in the left picture and 13 in the right picture), three eddies have been formed, resulting in reduced heat transfer.
In Richardson number of 10, free convection is more important. In this case the vertical high velocities expand some far from the hot and cold walls, therefore for negative direction of motion of down wall and positive direction of motion of up wall a fin is attached in 2/3Y and 1/3Y respectively.
Also, in cavity with bottom lid moving in positive direction and top lid moving in the negative direction (see figures (13b) in the left of the picture and (14b) in the left of the picture) for Ri=10.0, much like in Ri=1.0 three sub-cavities are existed.
For clarity of above explanations, Nusselt numbers and non-dimensional shear stresses ($\tau^* = \frac{\partial V}{\partial X} = \mu \frac{\partial v}{\partial x} \Big/ \frac{U_0 \mu}{L}$) on the cold and hot wall are plotted in Figure (15) at Ri=1.0 for 4 cases. As seen, the local Nusselt number in each case with attached fin on the hot wall is lower than same case (same boundary condition) without attached fin.
These results show that fin attached to the hot wall in optimal location with optimal length is caused to heat transfer decrease.

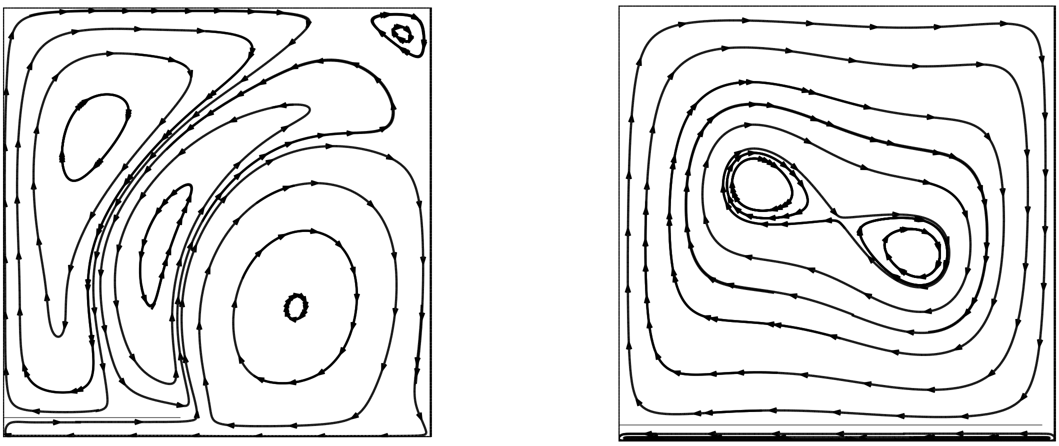


Figure 11 Flow contour lines in the cavity with the down lid moving in the negative direction (left) and in the positive direction (right) for Richardson numbers of 0.1, for decrease of heat transfer

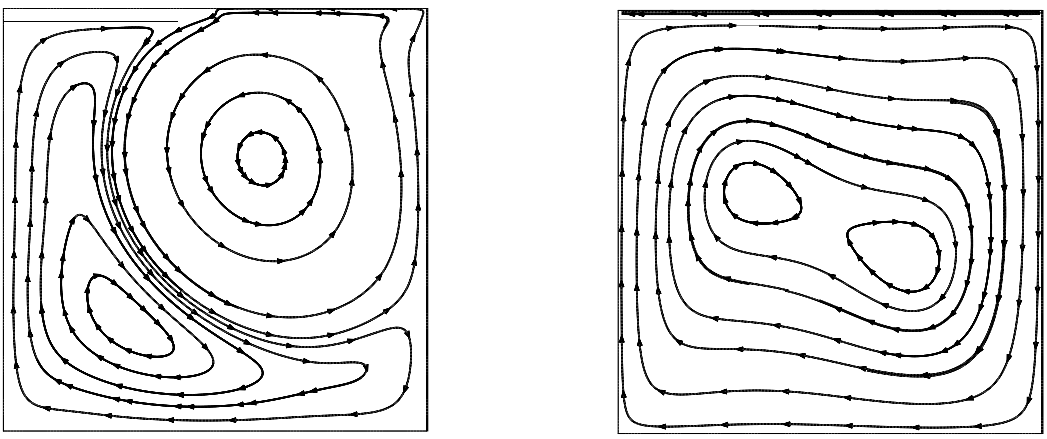
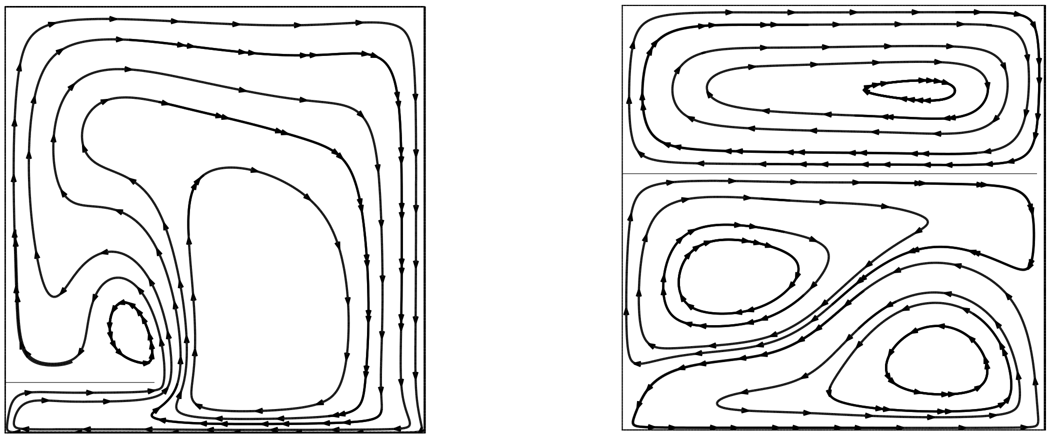
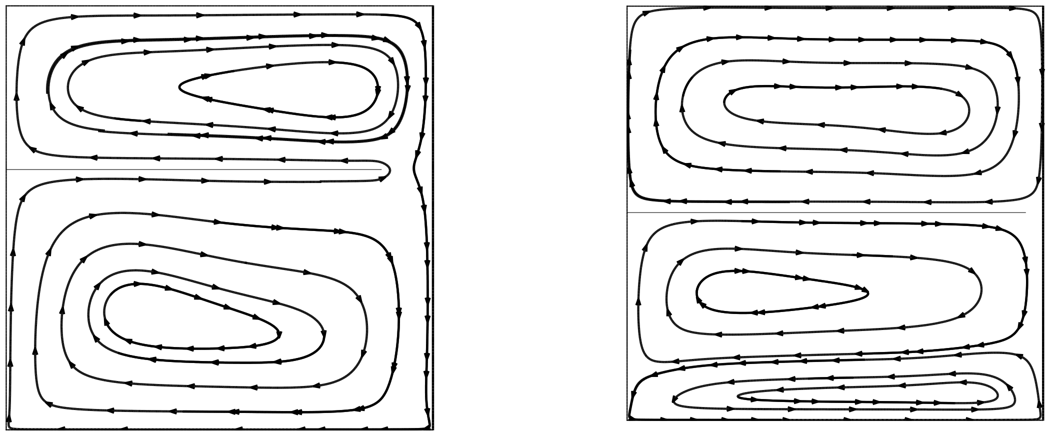


Figure 12 Flow contour lines in the cavity with the up lid moving in the negative direction (left) and the cavity with the up lid moving in the positive direction (right) for Richardson numbers of 0.1, for decrease of heat transfer



a)Ri=1.0



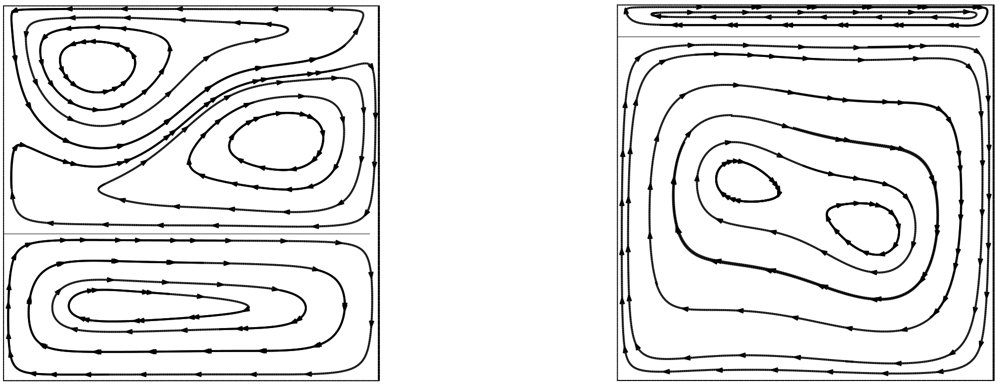
b)Ri=10.0

Figure 13 Flow contour lines in the cavity with the down lid moving in the negative direction (left) and in the positive direction (right) a)Ri=1.0,b)Ri=10.0 for decrease of heat transfer

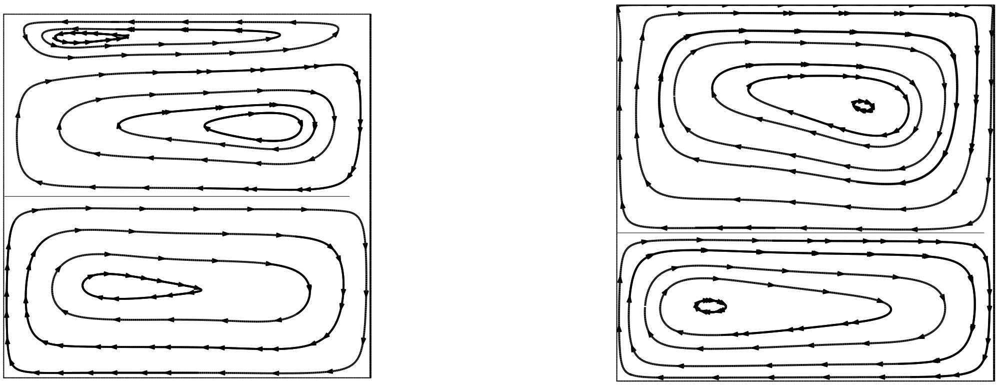
Also non-dimensional shear stresses show decreased on the cold wall for all cases compared with no fin attachment case because the fin cause that the velocities decrease near the cold and hot wall and blocking of flow is existed. The plotted charts are accordance to figures (13a) and (14a).

8.2 Impact of lid motion on the optimization of rectangular fin characteristics for heat transfer maximization

To maximize the heat transfer, we use a rectangular fin with a thermal conductivity ratio of 1000. The optimal lengths and positions of the rectangular fin and the percent of the enhancement of heat transfer in down lid driven cavity for two directions moving are shown in Table (7), best enhancement related to Ri=10.0 in positive direction of down lid. Also, the optimal lengths and positions of the rectangular fin and the percent of the enhancement of the heat transfer in up lid driven cavity for two directions moving are shown in Table (8), best enhancement related to Ri=10 in negative direction of up lid. Naturally, to increase the heat transfer, fin should be positioned at a location with low vertical flow velocity, so that the effect of increased surface area on heat transfer is maximized.



a)Ri=1.0



b)Ri=10.0

Figure 14 Flow contour lines in the cavity with the up lid moving in the negative direction (left) and in the positive direction (right) a)Ri=1.0,b)Ri=10.0 for decrease of heat transfer

Also, the impact of the fin on the cold wall should reduce its boundary layer and thereby increase the heat transfer. It is therefore clear that in Richardson number of 0.1, the optimal fin position is on the top of the cavity, where the effect of bottom lid velocity (both positive and negative) is minimized (see Figure (16a)).

In such cavity, the fluid that is exposed to the hot wall moves along this fin, becomes warmer, and allows a higher heat transfer ratio. The stream lines and non-dimensional shear stress plots on the hot wall shown in Figures (17) for these cavities illustrate the formed flow.

Naturally, the optimal fin length is the maximum length allowed by problem geometry.

In Richardson numbers of 1.0 and 10.0, for the cavity with bottom lid moving in the negative direction, the best fin position is still the place where, the effect of lid velocity, from both free convection and forced convection perspectives, is minimized (see figure (16b,c)). Since free and forced convections act in the same direction, this position is above the hot wall (near the top of the cavity). Also, the shear stress on the hot wall show zero value on the fin location for all cases (see figures (17) in left picture) as seen for negative direction different between fin attachment cases and no fin cases are negligible therefore less percent of enhancement (see Table (7)) are seen in these cases.

For accordance of shear stress plots and stream lines, one case from stream lines in each Richardson number are shown in Figure (17) in right picture. For example from the comparison of stream lines with shear stress for case of down lid driven cavity in positive direction specified that before the fin location, shear stress has negative value and after the fin location, it has positive value that it is accordance to two eddies with opposite directions that those existed bottom and up of the attached fin.

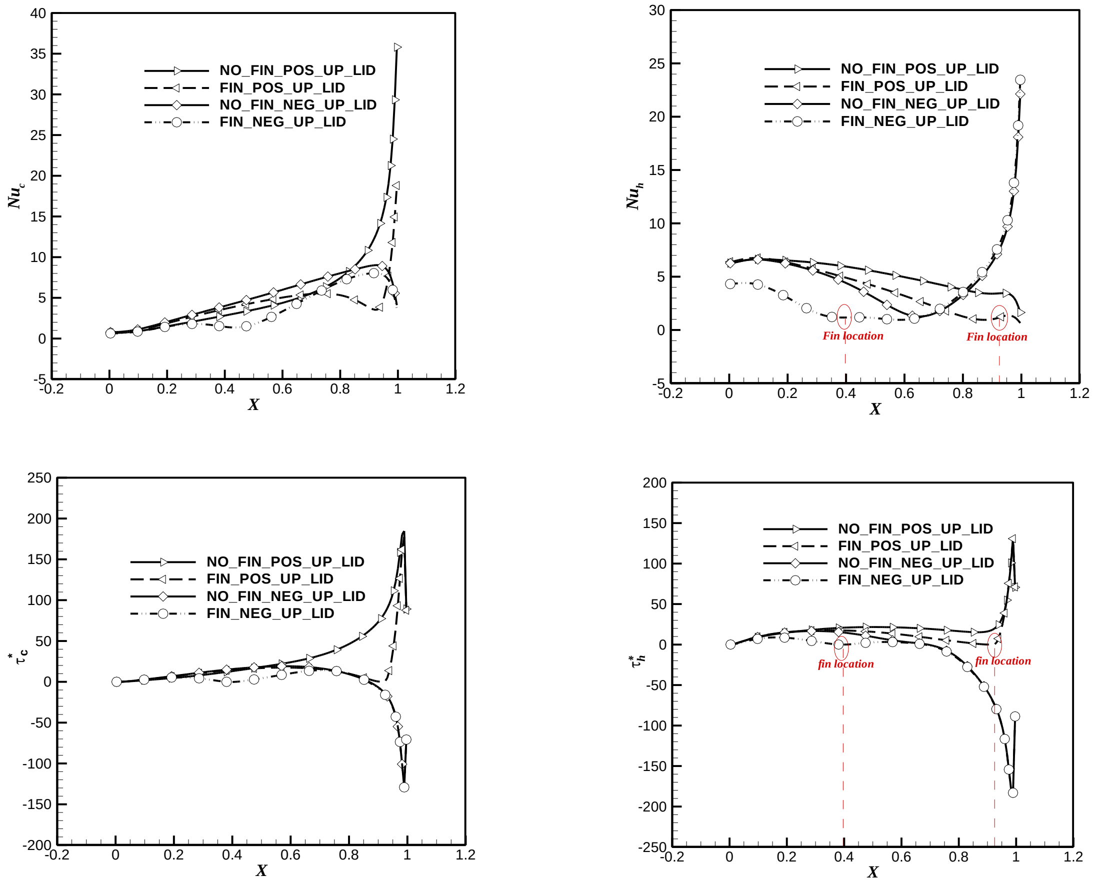


Figure 15 local Nusselt numbers and non-dimension shear stress for up lid motion cavity at Ri=1.0 in positive and negative directions a) Nusselt number on the cold wall b)Nusselt number on the hot wall c) shear stress on the cold wall d) shear stress on the hot wall

Table 7 Optimization Nusselt numbers in a cavity and optimal places and sizes of an attached rectangular thin fin on the hot wall for increase of heat transfer with down lid driven cavity in two directions in N_t =60

Ri	direction	$\overline{Nu}_{30,60}$	(x,y)	$\overline{Nu}_{30,60} / \overline{Nu}_{no-fin}$
0.1	+	9.67	(0.99,0.949)	44.99
1.0	+	7.77	(0.99,0.273)	57.95
10	+	6.77	(0.99,0.069)	71.50
0.1	-	9.68	(0.99,0.961)	33.05
1.0	-	7.68	(0.99,0.941)	47.03
10.0	-	6.97	(0.99,0.941)	54.03

Table 8 optimization Nusselt numbers in a cavity and optimal places and sizes of an attached rectangular thin fin on the hot wall for increase of heat transfer with up lid driven cavity in two directions in N_t =60

Ri	direction	$\overline{Nu}_{30,60}$	(x,y)	$\overline{Nu}_{30,60} / \overline{Nu}_{no-fin}$
0.1	+	10.34	(0.99,0.039)	42.62
1.0	+	8.06	(0.99,0.057)	55.00
10	+	7.18	(0.99,0.057)	58.50
0.1	-	9.13	(0.99,0.037)	38.96
1.0	-	7.62	(0.99,0.057)	55.82
10.0	-	6.56	(0.99,0.926)	66.49

For the cavity with the bottom lid moving in the positive direction, in Richardson number of 1.0, since free and forced convections act in the opposite directions, the optimization algorithm needs to check three positions: 1) The bottom of the cavity, where free convection has the lowest velocity. 2) The top of the cavity, where the effect of velocity due to forced convection and the velocity due to free convection are low. 3) the position where the collision of opposing free and forced convections reduce the vertical velocity values. The optimal fin position found by the optimization algorithm is located at the bottom third of the cavity at y=0.27 for Ri=1.0(see Figure (16b) in right of the picture (16)). Also, for Richardson number of 10, where free convection is dominant, the effect of bottom lid motion is negligible and the best position is located near the moving lid at y=0.06. In this case, for both Richardson (Ri=1.0,Ri=10.0) numbers, the flow contour lines show a large eddy in the upper section of the cavity (see figure (17 b,c) in right picture). As can be deduced from Table (8), when Richardson number is 0.1, for the cavity with the up lid moving in the positive and negative directions the best fin position is in the lowest part of the wall as expected. Also, when Richardson number is 1.0, much like Ri=0.1 the fin with highest length is attached on the hot wall in lowest part of this wall.

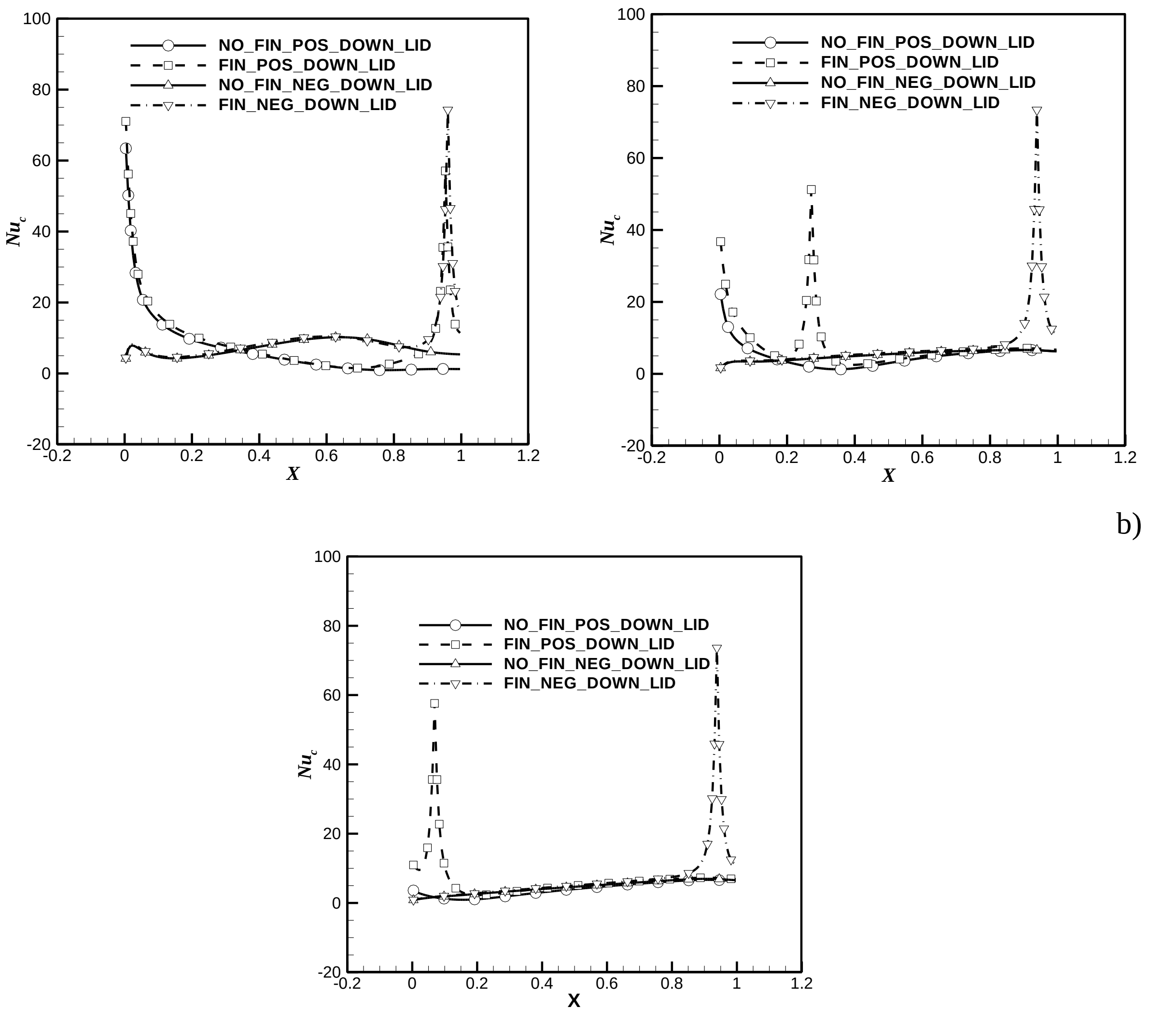
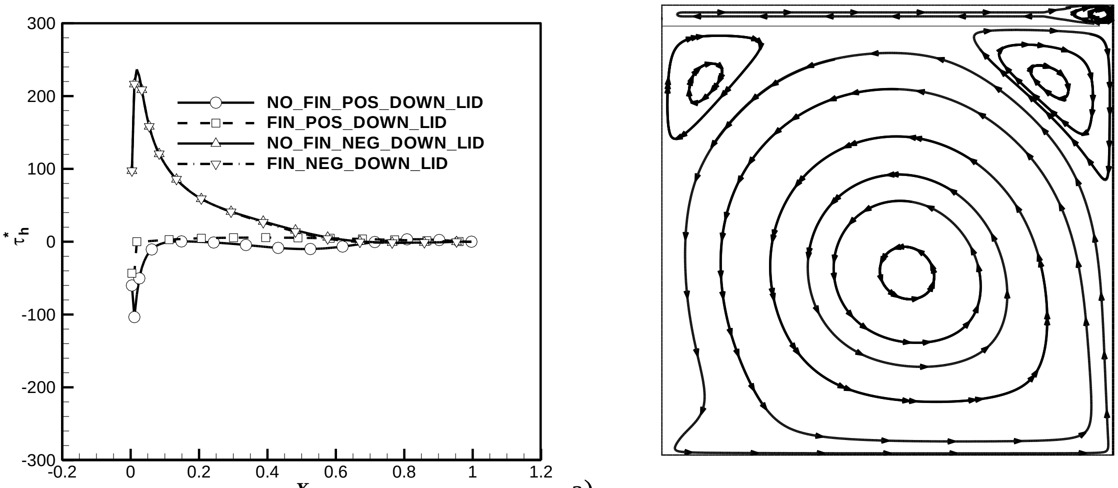
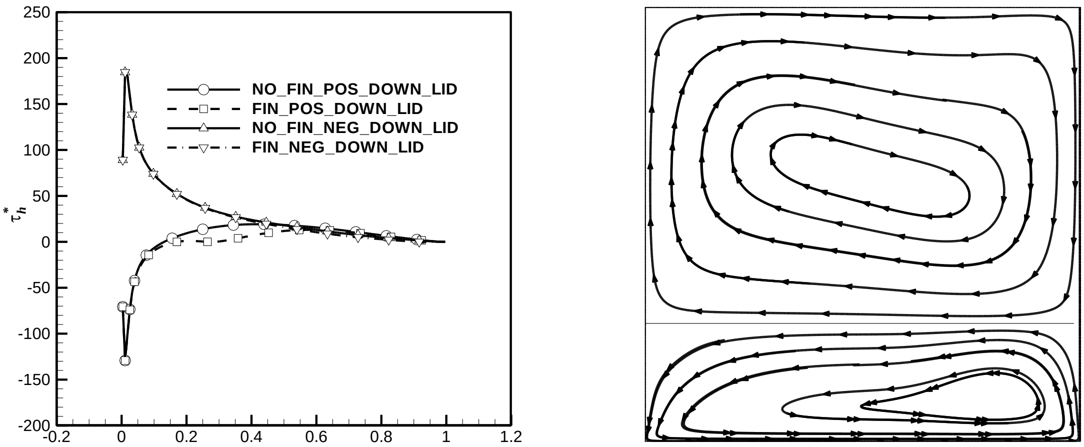


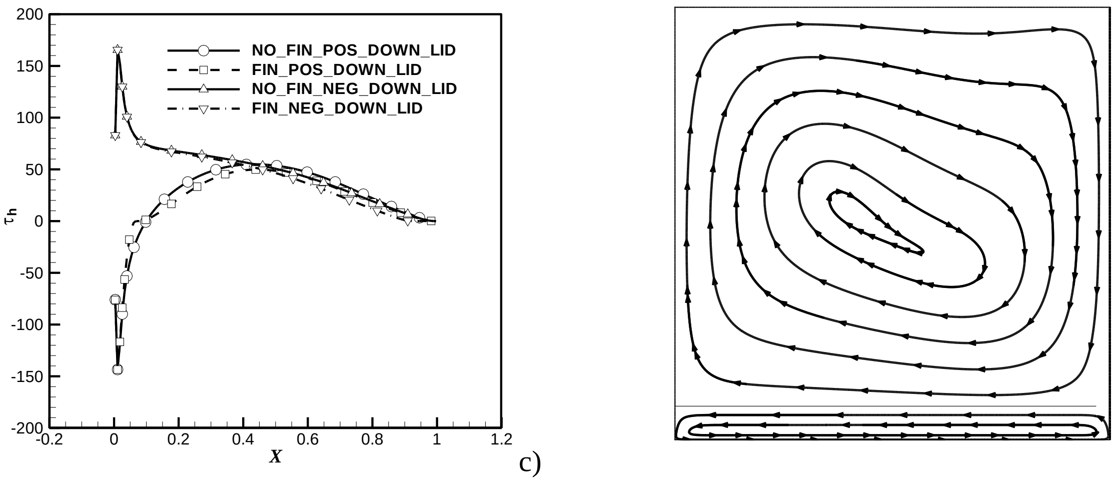
Figure 16 local Nusselt number plots with the down lid moving in the negative and positive directions a,)Ri=0.1, b)Ri=1.0 , and c)Ri=10 for increase of heat transfer.



a)



b)



c)

Figure 17 Stream lines with the down lid moving in the positive direction (right) and non-dimensional shear stress on the hot wall for 4 cases (left) a,)Ri=0.1, b)Ri=1.0 , and c)Ri=10 for increase of heat transfer.

However, for Ri=10.0 with the opposite of the forced flow and free flow, the best fin position in the same cavity relocates to the up part of the wall (Figure (18b)) but for aiding force and free convection flows best position of thin fin is near the down insolation wall (Figure (18b)). The shear stress plots on the hot and cold walls for these cases do not show large changes (see Figure (18c,d). It shows effect of blocking is negligible and heat transfer effect is dominant.

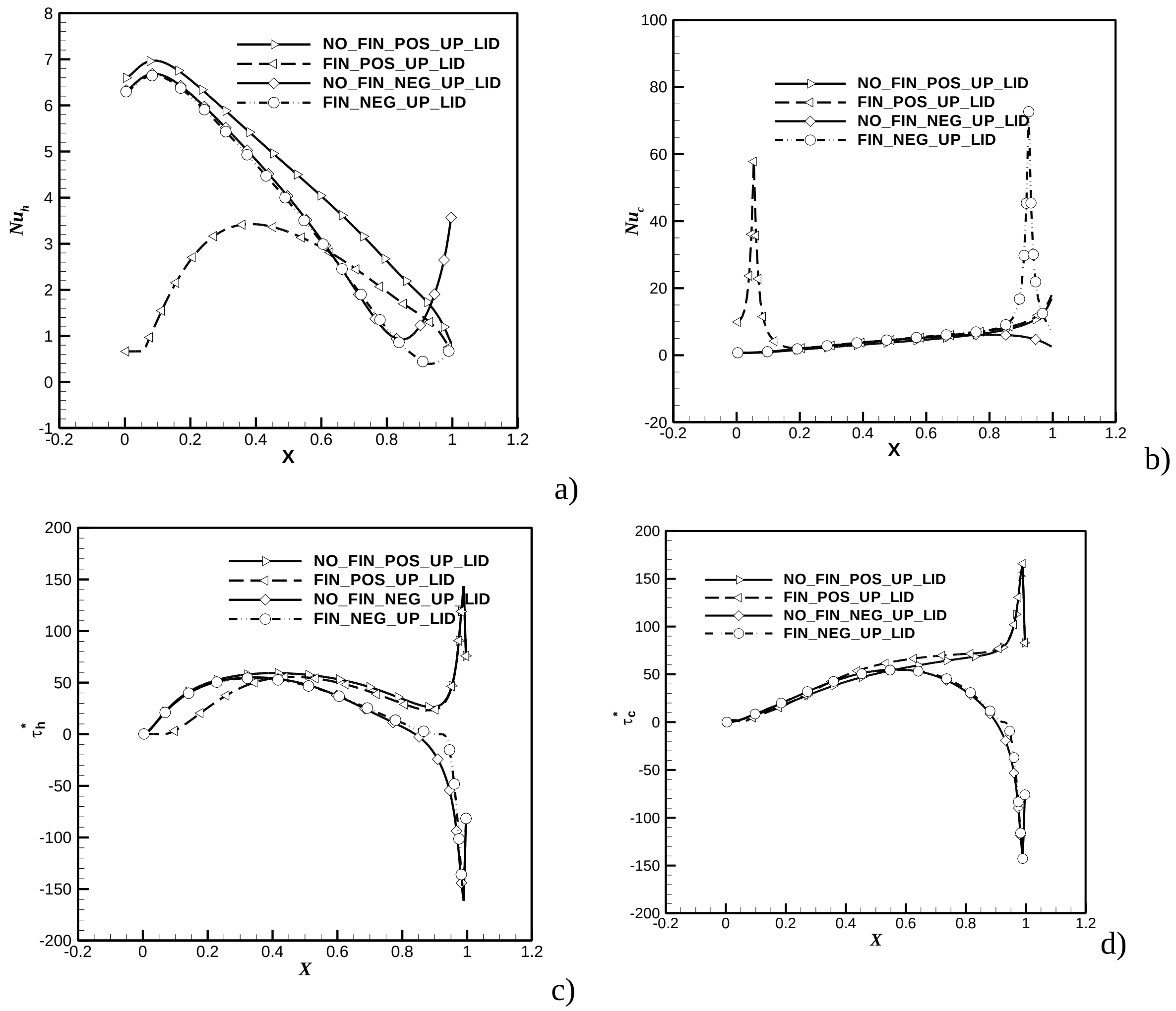


Figure 18 local Nusselt number and non-dimensional shear stress plots for negative and positive of up lid driven cavities at Ri=10.0 a,c) on the hot wall b, d) on the cold wall

The comparison of hot and cold Nusselt numbers (figure (18a,b)) show a fin with high conductivity cause to large value of energy enter to the medium and exit from the cold wall.

9 Conclusion

In this study, we investigated the effect of optimal fin position on the control over heat transfer through a square lid-driven cavity with active vertical walls and inactive (insulated) horizontal walls. Considering the dominance of mixed convection because of the presence of an insulated moving lid on the top, mixed convection equations were solved using the volume control method and with the help of the SIMPLER algorithm.

The particle swarm optimization algorithm was used to determine the fin position (on the hot wall) that minimizes or maximizes the heat transfer to the cold wall. This paper consisted of three main parts. In the first part, we examined for a triangular fin with specific dimensions, the fin position that minimizes the heat transfer through the cavities where top lid moves in the positive or negative direction. For this analysis, fin thickness and length were considered constant, but fin position could vary from 0.1Y to 0.99Y, where Y is the total length of the cavity. The results obtained for three Richardson numbers of 0.1, 1.0, and 10.0 and for top lid motion in both positive and negative directions showed that, for all Richardson numbers,

when the top lid is moving in the positive direction, the optimized position of the fin with a thermal conductivity ratio of 1 is not different from the unoptimized position.
But when the lid moves in the opposite direction, with the conflict of free and forced convection and the dominance of forced convection in Richardson numbers of 0.1 and 1.0, the top of the hot wall becomes the optimal fin position. The best heat reduction achieved with this fin was 9% in the Richardson number of 1.0. In the second part of the paper, we investigated the effect of fin shape on heat transfer reduction. For this purpose, we repeated the analyses with a rectangular fin with the same area as the triangular fin analyzed in the previous section. This comparison revealed the higher heat transfer reduction performance of the rectangular fin as compared to triangular one, but the optimal fin positions remained unchanged. In the last part of the study, we investigated the effect of the top lid and bottom lid direction of motion on the optimal length and position of a rectangular thin fin mounted on the hot wall to control the heat transfer through the cavity. The part of the study showed unpredictable results, which demonstrated the need for the use of optimization algorithm. Of course some important results are summarized as follows:
1. The effect of top and bottom lid motion velocity are different on the optimal position and length of the fin.
2. The Effect of direction of up or down insulated lids were caused high different in shape of flow counters with a fin optimal characteristics.
3. The most decrease of average Nusselt number related to Ri=0.1 that it was less than pure free convection with $Gr=10^5$, Pr=0.71.
4. Some optimal position and size of fin were not predictable and did not detected to now.

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Nomenclature

C_1	Cognitive Parameter
C_2	Social Parameters
GO	objective function
g	gravitational acceleration, m/s^2
H	height of cavity, m
L	Length of cavity,m
N	Number of variables
N_t	Number of particle
Nu	Nusselt number
P	Gas pressure (pa)
PI_g	Best position of all particle at iteration t
PI_i	Best position of particle i at iteration t
Pr	Prandtl number
Q	Non-dimensional heal flux
R	Number of grids over the cold surface
r_1,r_2	Random numbers
Ra	Rayleigh Number
t	Temperature, K
u,v	Velocity components in x,y direction respectively, m/s
VI_i	Velocity of *i*-th particle
x,y	Cartesian coordinates
XI_i	Position of *i*-th particle

Greek Symbol

α	Thermal diffusivity, m^2/s
β	Thermal expansion coefficient, K^{-1}
υ	Kinematic viscosity, m^2/s
ρ	Density (kg m^{-3})
ω	Inertia weight

Subscripts

0	Reference
c	Cold
d	Desired
e	Estimated
end	Ending value
f	Fin
h	Hot
Max	Maximum value
Min	Minimum value
New	New value
Old	Old value
Start	Starting value

چکیده

در این مقاله اثر حرکت صفحه متحرک بر روی بهترین مکان و اندازه پره متصل به دیوار گرم یک محفظه مربعی با دیوارهای عمودی فعال مورد مطالعه قرار می‌گیرد. تاثیر موقعیت و مکان پره در اعداد ریچاردسون ۰/۱، ۱ و ۱۰ بر روی میدان جریان و حرارت مورد تحلیل قرار می‌گیرد. وجود یک پره مستطیلی با ضرایب هدایت ۱ و ۱۰۰۰ در محفظه با سطح متحرک، بر روی کاهش و افزایش انتقال حرارت از چنین محفظه هایی بررسی می شود. به این منظور، معادلات جابجایی ترکیبی با استفاده از روش حجم کنترل و به کمک الگوریتم سیمپلر حل می‌شوند. همچنین از الگوریتم کوچ پرندگان به منظور یافتن مکان بهینه پره متصل بر روی دیوار گرم استفاده می‌شود، طوری که میزان انتقال حرارت به دیوار سرد کاهش و یا افزایش یابد.
نتایج به دست آمده، نشان میدهد که علاوه بر آنکه جهت سرعت در بهترین مکان و طول پره تاثیر گذار است وجود سرعت صفحه عایق بالا یا پایین نیز در بهترین مکان و طول آن بسیار موثر است. بیشترین تاثیر کاهنده پره بر روی عدد ناسلت محفظه در ریچاردسون ۰/۱ در حالت درب متحرک پایین در جهت منفی است که این کاهش به ۶۳٪ نسبت به حالت بدون پره میرسد و بیشترین افزایش ناسلت مربوط به عدد ریچاردسون ۱۰ در جهت مثبت حرکت درب پایین با افزایش ۷۱٪ می شود.