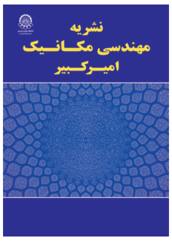

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Natural Convection Heat Transfer Inside a Square Enclosure with a Flexible Fin

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ABSTRACT: The present study aims to address the effect of the presence of a flexible fin on the natural convection heat transfer inside a square cavity. A flexible fin is placed on the left vertical wall by initial tilted angle 30o from the horizontal direction. An Arbitrary Lagrangian-Eulerian method for fluid-structure (fluid-flexible fin) interaction is utilized. Based on this method, the governing system of equations for laminar fluid and heat transfer is formulated into a non- dimensional form and then solved using the finite element method and then results accuracy evaluated against previous valid studies. The results are plotted for an enclosure containing a flexible fin as well as a solid fin in the non-dimensional time interval of 0 to 0.07 and in the Rayleigh number range of 10^6 to 2×10^7 and the fin tilted angle of -10° to +40°. The results show that the presence of a flexible fin deteriorates the heat transfer compared to a solid fin. In other words, using an insulated fin instead of a conductive fin makes different patterns for average Nusselt number curve in a range time and causes a reduction of the rate of heat transfer. Also, the presence of a flexible fin mounted on the hot wall especially affects the average Nusselt number in the areas above the fin location and induces oscillating heat transfer patterns.

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1- Introduction

The design of systems based on natural convection heat transfer is interesting and challenging due to its application in several engineering fields such as solar collectors [1], insulating systems [2], air conditions of buildings [3], cooling of electrical systems [4] and piezoelectric [5]. A piezoelectric generates electricity by means of vibration. In other words, the vibration of a piezoelectric can lead to the voltage difference. Nevertheless, Fluid Structure Interaction (FSI) in terms of natural convection heat transfer inside an enclosure can explain behavioral aspects of a piezoelectric. Hence, the present study investigates the effects of natural convection heat transfer on a flexible fin placed on the left vertical wall inside of a square cavity.

2- Mathematical Modeling

Fig. 1 shows an $L\times L$ enclosure accompanied by related boundary conditions. In the enclosure, a flexible fin (or rigid one) with l in length, t_f in thickness and inclination angle ϕ has been mounted on the left hot wall (T_h). The right wall is at lower temperature ($T_h>T_c$). In addition, the horizontal walls are adiabatic. It is assumed that the fluid flow is in laminar, incompressible and the fluid acts as a Newtonian fluid.

According to the mentioned assumptions, governing equations for solid phase (fin) and fluid phase have been defined and then transformed to their dimensionless forms as follows:

For solid phase (flexible fin):

$$\kappa\left(\frac{\partial^2\theta_s}{\partial x^2}+\frac{\partial^2\theta_s}{\partial y^2}\right)=\frac{\partial\theta_s}{\partial\tau} \tag{1}$$

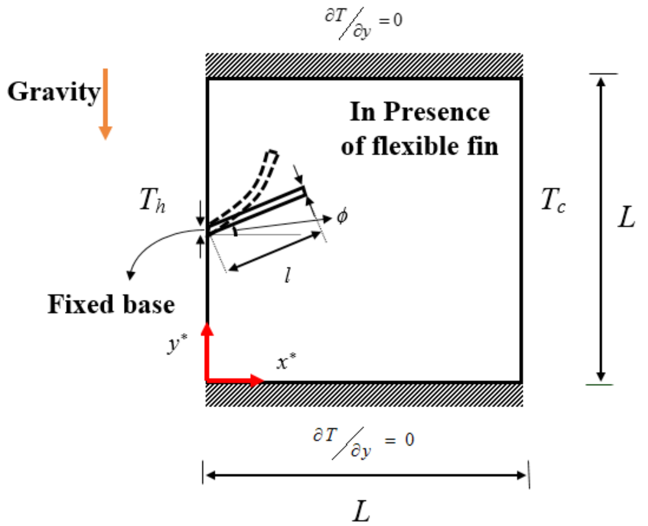

Figure 1. Schematic diagram of the square enclosure with a flexible fin

$$\frac{1}{\rho_R}\frac{d^2 d_s}{d\tau^2}-E_\tau\left(\frac{\partial\sigma}{\partial x}+\frac{\partial\sigma}{\partial y}\right)=E_\tau F_V \tag{2}$$

And for fluid phase

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \tag{3}$$

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$$\frac{\partial v}{\partial \tau}+(u-w_x)\frac{\partial v}{\partial x}+(v-w_y)\frac{\partial v}{\partial y}=-\frac{\partial P}{\partial x}+\Pr\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right) \quad (4)$$

$$\frac{\partial v}{\partial \tau}+(u-w_x)\frac{\partial v}{\partial x}+(v-w_y)\frac{\partial v}{\partial y}=-\frac{\partial P}{\partial y}+\Pr\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+Ra\Pr\theta \quad (5)$$

$$\frac{\partial \theta_f}{\partial \tau}+(u-w_x)\frac{\partial \theta_f}{\partial x}+(v-w_y)\frac{\partial \theta_f}{\partial y}=\left(\frac{\partial^2 \theta_f}{\partial x^2}+\frac{\partial^2 \theta_f}{\partial y^2}\right) \quad (6)$$

Moreover, the following relation has been defined to show the enhancement percent of the natural convection heat transfer by a flexible fin versus a rigid fin:

$$\text{Enhancement}(\%)=\frac{Nu_{Flexible}(\tau)-Nu_{Rigid}(\tau)}{Nu_{Rigid}(\tau)}\times 100 \quad (7)$$

where $Nu_{Flexible}$ indicates the Nusselt number for a flexible fin and Nu_{Rigid} shows the heat transfer for a rigid fin.

3- Numerical Solution Method

In the modeling of systems, in which moving of boundary causes a change in the system geometry, the Arbitrary Lagrangian-Eulerian (ALE) method can be utilized. This method relates to two suitable analytical points of views, the domain moving (presented by Lagrange) and fixed domain (presented by Euler) [6]. In the present study, the computational grid can go through arbitrary movements. The grid movements depend on the displacement of the flexible fin. The displacement of the fin is under the direct influence of the fluid motion.

The set of the non-linear governing equations subject to the related boundary conditions were solved utilizing the finite element method in ALE system. To use finite element method, governing equations were rewritten in the weak form. The computations of the residuals were continued using Newtonian iterative method until the accuracy becomes less than 10^{-5}.

4- Results and Discussion

First, to determine independency of the solution from the grid size, several samples, and grid sizes have been investigated for fin (solid phase) and fluid phase. Then, to be certain of the utilized computational code, the obtained results have been validated against prior works. Finally, the main results are shown based on the defined default values.

Fig. 2 indicates patterns of the average Nusselt number in cases of the flexible fin and rigid fin on the hot left wall in the dimensionless time range of zero to 0.07. As seen, the curves pattern of flexible and rigid fins are different. In addition, it is clear that heat transfer rate for a flexible fin is lower than that of the rigid one.

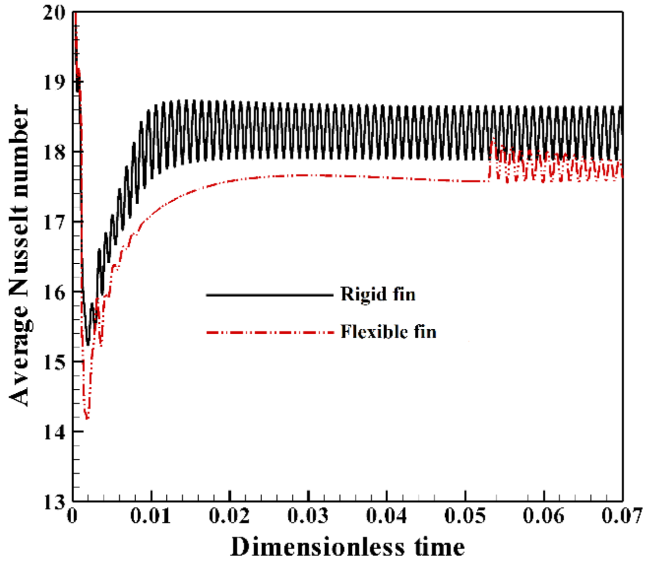

Figure 2. Average Nusselt number as a function of dimensionless time

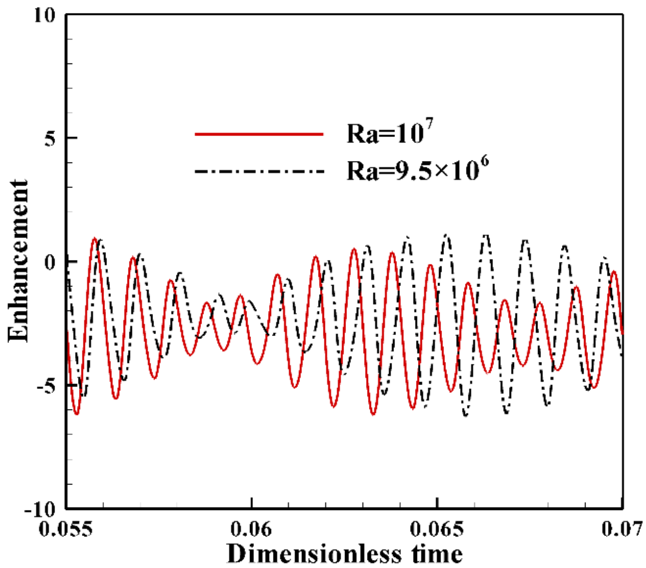

Figure 3. The heat transfer enhancement for $Ra=10^7$ and 9.5×10^6.

Enhancement percent of heat transfer (i.e. Eq. (7)), has been evaluated for two Rayleigh numbers $9.5\times10^{+6}$ and 10^{+7} and the results are depicted in Fig. 3 The obtained values reveal that for both of the cases of $Ra=9.5\times10^{+6}$ and $Ra=10^{+7}$ the heat transfer rate has been decreased by using a flexible fin. Also, it can be found that average deterioration intensities of the heat transfer (in range of $\tau=0.055$ to 0.07) by $Ra=10^{+7}$ and $9.5\times10^{+6}$ are 3.70% and 3.22%, respectively.

5- Conclusions

Natural convection heat transfer, as well as impressions of FSI in the square enclosure with a flexible (or rigid) fin mounted on the left vertical wall, was evaluated.

Results of the present study show that the presence of a flexible fin in comparison with a rigid fin causes deterioration of heat transfer rate. On the other hand the oscillating behavior of the fin can become an applicable element in the piezoelectric field.

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