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Natural Convection Heat Transfer in a Baffled Triangular Enclosure

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Abstract

To reduce the natural convection heat loss from enclosures many researchers used convection suppression devices in the past. In this study a single baffle is used under the top tip to investigate numerically the natural convection heat loss in an attic shaped enclosure which is a cost effective approach. The case considered here is one inclined wall of the enclosure is uniformly heated while the other inclined wall is uniformly cooled with adiabatic bottom wall. The finite volume method has been used to discretize the governing equations, with the QUICK scheme approximating the advection term. The diffusion terms are discretized using central-differencing with second order accuracy. A wide range of governing parameters are studied (Rayleigh number, aspect ratio, baffle length etc.). It is observed that the heat transfer due to natural convection in the enclosure reduces when the baffle length is increased. Effects of other parameters on heat transfer and flow field are described in this study.

Introduction

One of the most important forms of heat transfer and fluid flow in an enclosure is natural convection where the fluid motion is simply induced by density gradients due to temperature differences. Natural convection in a triangular enclosure has received extensive attention (see [1]) since it has many engineering applications in energy transfer in rooms and buildings, convective motion in solar stills, nuclear reactor cooling, and electronic equipment cooling.

One of the earliest studies on natural convection in triangular cross sectional enclosure was conducted by Probert and Thirst [2, 3]. In their experiment studies the authors found the optimal pitch angle leading to the minimum rate of heat transfer in the attic of a modeled pitched roof with specified boundary conditions. They showed that the contribution by convection heat transfer is increased rather than conduction with the increase of aspect ratio up to a critical value.

Flack et al [4] used laser velocity meter and an interferometer to measure convection velocities and heat transfer rate of a bottom heated and cold inclined walls of a triangular enclosure filled with air. Single roll circulation in each half of triangular cross section was reported and transition to turbulence occurred for larger Rayleigh number. Gradual dependency of mean Nusselt number on Rayleigh number, regardless of geometry, was observed for the mentioned case. Latter, natural convection in a right angled triangular enclosure with bottom heating and cold side walls for air and distilled water was studied by Poulikakos and Bejan [5, 6] where Rayleigh - Benard type convection occurred. The authors established a new relationship for mean Nusselt number when air filled the enclosure.

Holtzman et al. [7] performed series of experiments using smoke injected into the triangular enclosure with bottom heated and cold side walls. Symmetry assumption was reported for lower Rayleigh number, however, with the increase of Rayleigh number asymmetry and multicellular flow progressed inside the enclosure denoting the appearance of pitchfork bifurcation. Number of counter rotating cells and asymmetricity were increased with increased of Rayleigh number until a critical value is achieved. Ghassemi and Roux [8] reported the reduction of mean Nusselt number when the pitch angle is increased due to the increment of distance between cold and hot wall. The appearence of pitchfork bifurcation is also investigated for diurnal thermal forcing condition [10] where the temperature on inclined surfaces sinusoidally changes with time.

Since one of the objectives of designing a residential house is to reduce the heat transfer inside the attic space, several researchers have conducted research by adding adiabatic fin on the walls [11–15]. Varol [11] incorporated single thin adiabatic plate onto a triangular cavity in order to disrupt the heat flow. It was found that for a low Rayleigh number, the main mode of heat transfer is by conduction which is in line with the definition of Rayleigh number. From this research, the maximum and minimum mass flow of the stream function can be decreased by increasing the distance of the plate from the origin. For better understanding on the way of affecting heat transfer in a triangular enclosure, more works have been conducted by adding adiabatic fins on several places of the walls [12]. This is a similar study conducted by by Chamkha [13] with the addition of a single fin to the base of the triangle.

In this study steady natural convection in a triangular enclosure with a baffle attached to the top tip is investigated numerically. The finite volume method has been used to solve the governing equations. Effects of Rayleigh number, aspect ratio and baffle length on heat transfer and flow field are described in this study. It is found that the effect of the length of the baffle or interrupter has a great influence on the heat transfer. The results are presented as a form of temperature contours and stream functions. The heat transfer is represented as a form of Nusselt number.

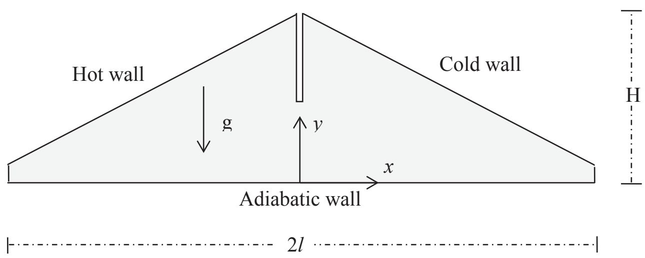

Figure 1: Schematic of the geometry and the boundary conditions

Problem Description

Under consideration is the steady flow behaviour resulting from heating/cooling a quiescent, isothermal Newtonian fluid of air in a two-dimensional triangular enclosure of height H and horizontal length $2l$. There is a flow interrupter attached with the top tip of the enclosure. The left inclined surface is heated and the right inclined surface is cooled whereas the bottom base is adiabatic as shown in figure 1. All boundaries are kept nonslip. It is also assumed that the flow is laminar. In order to avoid the

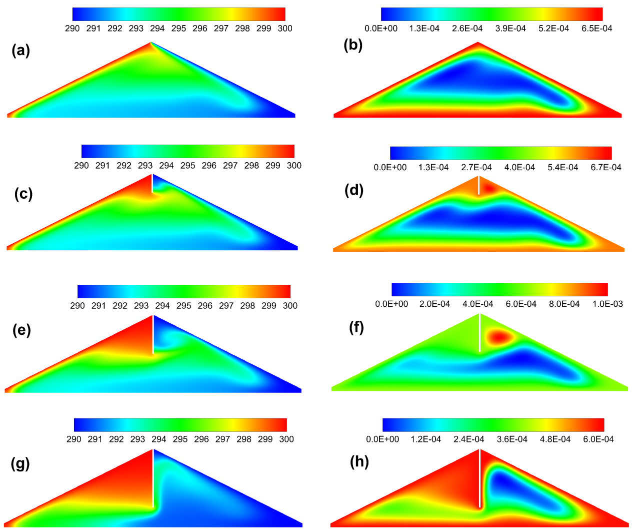

Figure 2: Temperature contours (left) and stream functions (right) for different length of the interrupter; (a,b): no interrupter, (c,d): 25% of H, (e,f): 50% of H, (g,h): 75% of H when Ra = 10^6 and $A = 0.5$.

singularities at the tips in the numerical simulation, the tips are cut off by 5% and at the cutting points (refer to figure 1) rigid non-slip and adiabatic vertical walls are assumed. We anticipate that this modification of the geometry will not alter the overall flow development significantly.

The evolution of the temperature and flow fields in the triangular enclosure is described by the following set of governing equations, for which the Boussinesq assumption has been made:

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

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{\partial p}{\partial x} + Pr\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \tag{2}$$

$$u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} = -\frac{\partial p}{\partial y} + Pr\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) + RaPrT \tag{3}$$

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

where u and v are the velocity components along x and y directions respectively, p is the presure and T is the temperature. The Rayleigh number, Ra and the Prandtl number Pr are defined by

$$Ra = \frac{g\beta\Delta T H^3}{\kappa\nu}, \text{ and } Pr = \frac{\nu}{\kappa} \tag{5}$$

where ν, β and κ are kinematic viscosity, coefficient of thermal expansion and thermal diffusivity respectively.

The aspect ratio and the Nusselt number are defined as

$$A = \frac{H}{l}, \text{ and } Nu = \frac{qH}{\Delta T k} \tag{6}$$

where q is the convective heat flux through a boundary and k is the thermal conductivity.

The initial and boundary conditions are defined as follows:

- initially the fluid is quiescent and isothermal.
- on the sloping walls, a rigid non-slip and uniform heating temperature conditions are applied on left wall and uniform cooling temperature conditions are applied on right wall(see figure 1).
- the bottom horizontal wall is maintained with an adiabatic and rigid non-slip.
- at the cutting points of the bottom tips, rigid non-slip and adiabatic vertical walls are assumed.

The governing equations (1)-(4) along with the specified initial and boundary conditions are solved using control volume method. The finite volume scheme has been chosen to discretize the governing equations, with the QUICK scheme approximating the advection term. The diffusion terms are discretized using central-differencing with second order accuracy. An extensive grid independence test is carried out for this study. The results are not showed here due to brevity. The suitable number of grid nodes are adopted for three different aspect ratios of $A = 1.0, 0.5$ and 0.2 are 169848, 166744 and 160532 respectively.

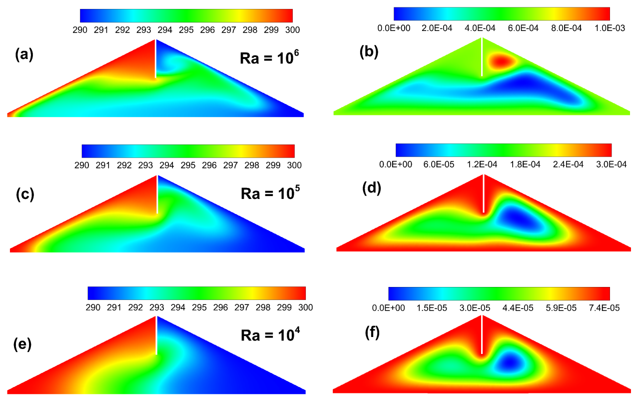

Figure 3: Temperature contours (left) and stream functions (right) for different Ra while $A = 0.5$ and length of the interrupter is 50% of height.

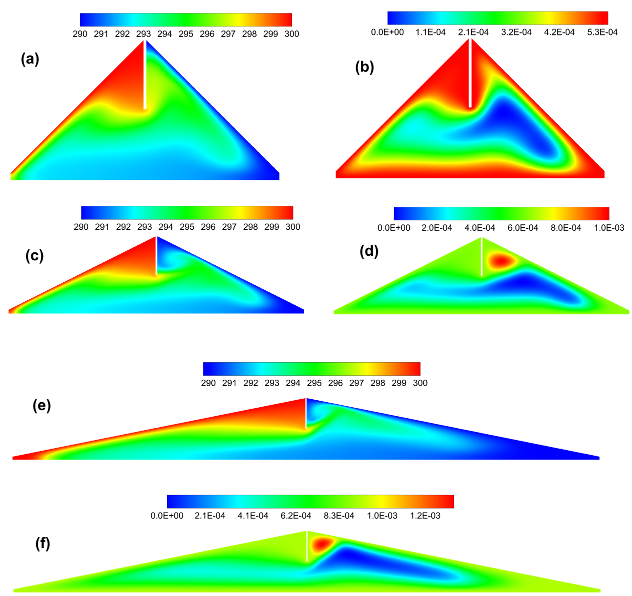

Figure 4: Temperature contours (left) and stream functions (right) for different A when Ra = 10^6 and the length of the interrupter is 50% of height

Results and Discussions

Effects of Length of the Interrupter:

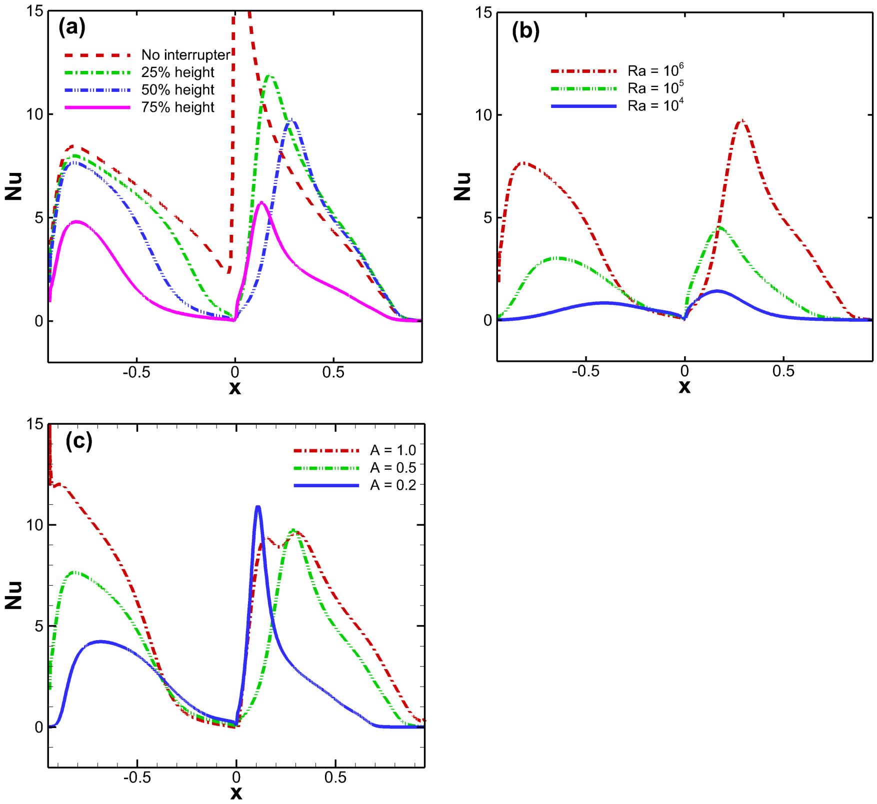

Figure 5: Nusselt number calculated on the inclined surfaces: (a) effect of interruption for Ra = 10^6 and $A = 0.5$ (b) effect of Ra when $A = 0.5$ and length of interrupter is 50% of H and (c) effect of A while Ra = 10^6 and length of interrupter is 50% of H.

It is clearly observed from figure 2 that with the addition of an interrupter on the top tip of the enclosure, the heat transfer within the enclosure changes drastically. As the length of the interrupter increases, the heat trapped on the hot side of the wall increases. This is due to the nature of heated air, as hot air does not flow downwards easily due to buoyancy. This is clearly shown in the enclosure for the case when the length of the intrrupter is 75% of the cavity height. The build-up of heat layers occur before the heat begins to flow to the cold wall. This modification of the geometry would be very beneficial during winter conditions as the trapped layer of heated air would play a vital part in heating the house and have a significant saving in heating application.

Effects of Rayleigh Number:

Figure 3 represents the temperature contours and stream functions for various Ra. It is observed from this figure that for low Rayleigh number single cell is visible, however, as Rayleigh number increases, the solution becomes a two-vortex solution. As the Rayleigh number increases further, a multiple vortex solution can be observed. It is also observed from the temperature contours that the thermal boundary layer for low Rayleigh number is thicker. However, with increase of Rayleigh number the thermal boundary layer becomes thinner (concentrated near the inclined walls). That means, the flow is dominated by convection.

Effects of Aspect Ratio:

From the temperature contour, it can be observed from figure 4 that the heat distribution is quickest when the aspect ratio is the lowest whereas heat distribution is lowest when the aspect ratio is highest. This can be verified by the stream function (right side of figure 4). For the aspect ratio of 0.2, there are areas of concentrated lines indicating a low pressure, high velocity zone. With this low pressure, it would act as a vacuum to ensure air distribution from the left side of the enclosure to the right side. However, for the aspect ratio of 1.0, there are larger pockets of cooler air compare to the 0.2 ratio. This is due to the heat being trapped in the upper areas of the enclosure.

Heat Transfer:

Figure 5 shows the Nusselt number calculated on the inclined surfaces of the enclosure. Figure 5a represents variation of Nu for variation of the length of the interrupter. It is found that when there is no interrupter the heat transfer through the inclined walls is the highest. However, the heat transfer becomes lower when the length of the interrupter increases. The lowest Nusselt number calculated is for 75% of the height of the enclosure which is the highest length of the interrupter. In figure 5b the heat transfer is higher for higher Rayleigh number which is expected due to the fact that convection deminates the heat transfer for higher Rayleigh number. Variation of Nu is shown for different aspect ratios is shown in figure 5c. It is found that heat transfer is lower for lower aspect ratio. Since the base of the enclosure is much longer than its height, close to two tips (left and right tips) the heat transfer is dominated by conduction. The mixing of hot and aold air happens quicker.

Conclusions

This study shows the enhancement/suppression of natural convection when there is a modification of the geometry, i.e. the aspect ratio, or addition of an interrupter hanging vertically from the apex of the triangular enclosure. From the simulations obtained, it is found that with the addition of an interrupter, the heat transfer is significantly reduced. It is also found that the heat transfer reduces as the height of the interrupter increases. This would be a useful application on houses with winter climates. The trapped heat would be able to heat the interior of the house reducing the power consumption for heating devices. Here, it has also been demonstrated that with a higher aspect ratio, the mass transfer of the air decreases. It is found that with a lower aspect ratio the mass transfer increases thereby increases the flow and mixes of hot air with the cold air.

For further studies on the effects of an interrupter on the heat flow in an attic space, the geometry of the interrupter could be modified. To ensure that the interrupter would be viable for both summer and winter conditions, it would be recommended to introduce movable covered slots onto the interrupter itself. The purpose of these slots is to allow heat flow through the interrupter during summer conditions when opened and to act as a solid boundary layer during winter conditions when closed. Even though this is a good design to keep the heat from reaching the house, it is impractical as the cost of materials needed would increase.

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