Unsteady Natural Convection within a Differentially Heated Porous Enclosure

Ali Hooshyar Faghiri

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Prof. Dr. Elvan Yılmaz
Director

I certify that this thesis satisfies the requirements as a thesis for the degree of Master of Science in Mechanical Engineering.

Prof. Dr. Uğur Atikol
Chair, Department of Mechanical Engineering

We certify that we have read this thesis and that in our opinion it is fully adequate in scope and quality as a thesis for the degree of Master of Science in Mechanical Engineering.

Prof. Dr. Fuat Egelioğlu *Co-supervisor*	*Prof. Dr. Hikmet Ş.Aybar* *Supervisor*

Examining Committee

1. Prof. Dr. Fuat Egelioğlu
2. Prof. Dr. Hikmet Ş. Aybar
3. Prof. Dr. Uğur Atikol
4. Assoc. Prof. Dr. Hasan Hacışevki
5. Asst. Prof. Dr. Neriman Özada

ABSTRACT

Investigation on heat transfer performance of porous enclosure is numerically performed. The working fluid is air and the square cavity is heated from left side wall and the right edge is cold. Two other walls including top and bottom edge are kept as adiabatic. The mentioned above feature is used in several practical and industrial fields of engineering such as solar collectors. FLUENT 6.3 were used for simulating some specified case studies to control the amount of heat transferred through the media. Validating the work, we employed a simple porous enclosure of air to see how well we predict the previous results of researchers for square configuration. Then, there are distinct cases which we considered: using an insulated block on bottom/ top edge, couple insulated blocks at top and bottom and block within the media. All designed cases let the thermal system manufacturers know and produce a device much more economical. To have better insight to the rate of heat transfer, all studied cases are investigated from the starting time of the heating up to a steady state condition.

Keywords: *Unsteady natural convection, Porous enclosure, Computational fluid dynamics.*

ÖZ

Nusselt numarası ortalaması ile gözenekli muhafazanın ısı transferini araştırılması sayısal olarak gerçekleştirilmektedir. Çalışma sıvısı havadır ve kare muhafaza sol dikey duvardan ısıtılmaktadır ve sağ köşesi soğuktur. İki üst ve alt duvarlar izole edilmiştir. Yukarda sözü geçen özellik birçok Pratik ve endüstriyel alanlarda kullanılmakta örneği kolektörlerinde, ısı transfer miktarın kontra ıcın farklı durumlar dıkkate alındı : izolasyon, üst ve altta cıfte yalıtımlı blok ve ortam ıcınde blok . Tüm tasralanmişdurumlar,durumlar,termal sistemlerin daha ekonomik uretimi ıçındır. Isı transferı hakkında daha ıyı fikır sahıbı olmak ıçın çalışılan bütün konfigürasyonlarda ısıtmaya başlanılan andan kurarlı ortama ulaşılıncaya arştırma gerçekleştırıdı

Anahtar Kelimeler: *Kararsız doğal konveksiyon, Gözenekli muhafaza, Hesaplamalı akışkanlar dinamiği*

To My Family

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LIST OF SYMBOLS

C	*Matrices for inertial loss [m/s]^2*
C_2	*The inertial resistance factor [-]*
C_0 and C_1	*User-defined empirical coefficients [-]*
D	*Matrices for viscous loss [m/s]^2*
E_f	*Total energy of fluid [J]*
E_s	*Total energy of solid medium [J]*
H	*Height[m]*
h	*Convection coefficient [W/m^2.K]*
k_{eff}	*Effective medium thermal conductivity[W/m.K]*
Nu	*Nusselt Number [-]*
P	*Pressure[Pa]*
Pr	*Prandtl Number [-]*
Ra	*Rayleigh Number [-]*
T	*Temperature [K]*
X, Y	*Dimensionless coordinates[-]*
x,y	*Coordinates of dimension[m]*

LIST OF SYMBOLS (CONT.)

Greek letters

α	*Permeability[-]*
β	*Thermal expansion coefficient[1/K]*
φ	*scalar quantity[-]*
γ	*Porosity of the medium[-]*
μ	*Viscosity[Pa.s]*
θ	*Dimensionless temperature[-]*
τ	*Dimensionless time[-]*
σ	*Ration of porous material specific heat capacity [-]*
ν	*Kinematics viscosity[m^2/s]*
$\boldsymbol{\Gamma}$	*Accuracy of convergence[-]*

Subscripts

c	*Cold*
H	*Hot*

Chapter 1

INTRODUCTION

During recent decades scientists performed many studies on heat removal or controlling the amount of heat transferred though thermal system and devices such as electronic devices, components and equipments, solar collectors. Since force convection heat transfer requires an external force and extra cost, natural convection heat transfer receives much attention and study.

The main mechanism for the natural convection is the density gradients of upper and lower parts of the fluids which induce it to flow to the positions contain lower value of density. The generated velocity removes the produced heat. Note that the velocity values are of the lower order with respect to the velocity value in force convection heat transfer. Solving this problem, different additional equipments or strategies is used, such as using Nanofluids, installed fines, employing porous media and many other techniques. In the current study, porous media used to increase the amount of heat transferred and some blocks to control that value. It is important for thermal system designers to prepare economical and appropriate devices. Porous media is substance with solid matrix while there exist many voids and the induced air flows among those pores. The way we treat to predict fluids flow within such these media will be discussed in next parts.

In many porous matrials, there exist plenty of irrigular voids and free spaces relative to its size and shape. For instance, we can call special kinds of sands such as sandstone, limestone or rye bread, wood, and the human organic components as natural porous media

The most important characteristics of the porous medium are porosity. Other properties such as electrical properties, mechanical properties or metallurgical can sometimes be extracted from the related properties of its rigid matrix or fluid flowing inside, but this kind of derivation is usually complex. Often both the solid matrix and the pore space are related to each other in which they create the same feature as we see in a sponge. Also there is a definition of closed porosity and effective porosity, i.e., the free space related to the working fluid to flow. Many other materials including rocks, zeolites, biological tissues, and many substances created by human such as cements and ceramics can be treated as porous media. Note that these materials should be considered as porous media to truly evaluate the amount of their distinct properties.

Different usages are considered for porous media in various fields of study. Filtration, mechanics (different area of this science such as acoustics, geomechanics, soil mechanics, rock mechanics, etc), engineering (many cases in real world engineering are following the rules considered in these media, for example petroleum engineering, construction engineering, etc), geosciences (hydrogeology, petroleum geology, geophysics, etc), biology and biophysics, material science, etc. Fluid transport phenomena has received an extensive attention by the side of most researchers. Also deformation of these material is investigated in a science so called poromechanics.

As mentioned above, liquids and gases flow within porous substances is a subject of most common interest and has emerged a separate field of study. Due to the above applications, convective heat transfer in the rectangular / square saturated porous enclosures have received extensive attention in past decades

1.1 Objective of Study

The aim of this study is to investigate unsteady natural convection within porous enclosure of a square. But the main difference of this study with those simple and initial studies of for a square is using insulated block within the enclosure. This block is employed for the time reducing or increasing the amount of heat transferred thorough the medium.

1.2 Thesis Organization

In Chapter 2, the background of natural convection in porous media is introduced generally. These works include numerical and experimental techniques. In Chapter 3, mathematical modeling of enclosure is explained and related equations and numerical procedure is defined. In Chapter 4 results and discussion are expressed and plots related to the Nusselt number which represents mainly the amount of heat transfer are given. Validation of the work is presented in this chapter. In Chapter 5 conclusion of this study is presented

Chapter 2

BACKGROUND INFORMATION AND LITERATURE SURVEY

Natural convective heat transfer in porous enclosures where the flow is induced mainly by the communication between variation in density and the gravitational field has attracted extensive attention of researchers as the transport phenomenon in a fluid.

Water desalination devices, extracting energy stored within the earth, oil purification and recovery, processes related to food, thermal insulation of buildings, air conditioners filters, cosmetic applicators, dispersion of chemical contaminants in different processes in the industry and many others are huge practical applications of porous media. Due to huge amount of applications, it demands detailed analysis of convective heat transfer in various industrial operations.

Free convection appears inherently in several fields of study, where the annoying and extra heat to be dissipated is not too high and thermal system due to its cheapness, reliability and simplicity of employment.

Due to the above applications, convective heat transfer in the rectangular / square saturated porous enclosures have received extensive attention in the past decades. A thorough review and extensive bibliography on natural convection in porous cavities can be found in [1, 2] and Ingham and Pop [3] chipped into an complete review of *this important field for heat transfer in porous media. There are many previous works related to natural convection in rectangular porous cavities [4-11] available in the literature.*

Baytas and Pop[4] performed numerical calculations for the steady-state natural convection within a sloped cavity filled with a fluid-saturated porous medium. The inclined walls were kept at constant hot and cold temperature, while the horizontal top and bottom walls were insulated. They reported their reults for momentum and heat transport characteristics within an extensive range of the Ra, inclined angle and aspect ratio of the enclosure. Their results can be considered as a valuable reference for other solutions to be compared with, even for the current problem.

Bejan [6] described boundary layer behavior in a vertical cavity occupied with porous medium. He used a new approach to predict Nusselt number of the heating wall as well as previous results. Manole and Lage also [9] performed a benchmark study for porous filled cavity and reported results related to various Rayleigh numbers.

According to the above literature, thermal boundary layer related to the hot side wall was shown to increase gradually with time, which results in decrease of heat transferred to the media. To solve the current problem, new boundary conditions, modifications on side/ bottom walls and many other considerations were employed to enhance the amount of heat transferred among the two vertical walls.

Saeid [7] reported fluid flow and heat transfer characteristics for cavity of differentially heated from bottom and cooled from top where the side walls are kept *as insulated. A sinusoidal temperature variation was assumed for the bottom heated wall which has the higher mean value with respect to the cold wall temperature. Investigating parameters contained amplitude of the sinusoidal temperature and length of the heat source where the natural convection in the cavity were studied for* $20<Ra<500$*. It was shown that the average Nu for higher length of the heat source or higher the amplitude of the temperature variation increases.*

Varol et al [10] performed numerical studies about free convection heat transfer occurred in a porous rectangular enclosure where the temperature variation for the heated wall is assumed to be sinusoidally varying temperature. Other edges assumed to be as adiabatic and only the mentioned edge is being heated and cooled sinusoidally. Heat transfer rate increased for higher amplitude and decreased for enhancement of aspect ratio. Cellular flows (different number of cells within the flow) were reported for the range of studied parameters.

Bejan [12] has outlined a complete review of scale analysis for convection heat transfer in porous media and revealed that the average Nuesselt number for the natural convection boundary layer besides a vertical impermeable wall employed in a porous medium at a constant temperature is related to second order root of wall Darcy Rayleigh number. He could anticipate the exact results within 23 percent of those obtained using the scale up analysis. Magyari et al. [13] derived analytic solutions for two of the similarity cases identified by Johnson and Cheng [14] for the unsteady free convection boundary layer flow over an impermeable vertical flat plate adjacent to a fluid saturated porous medium. They are the solutions corresponding to an exponential and a power law variation of the surface temperature, respectively.

For engineering and industrial applications, it is vital to evaluate the natural convection heat transfer that is time dependent i.e. unsteady problem. As an earlier study, Patterson and Imberger [15] reported the features of unsteady free convection in a air filled cavity subjected to sudden heating and cooling based on the scaling analysis. A thermal boundary layer besides the sidewall, a horizontal movement of fluid next to the top and bottom walls and the flow in the middle region were the transient phenomena and stages analyzed by them. The transient characteristics of natural convection [16 - 20] has been studied in a cavity widely using different methods.

When sudden heating or cooling is applied to the side walls, two important flow regimes are generated for the initially quiescent fluid. The two different flow regimes diffuse into the middle of the cavity. When the flow reaches a steady /quasi-steady state, the fluid flowing at the core region is turned to be stratified, as mentioned by Eckertf and Carlson [21], and a special kind of thermal boundary layer is created, Xu et al.[22]. Studies of Schöpf and Patterson [19] and Xu et al. [22] showed the existence of a convective instability in the boundary layer flow. From the results of Xu et al.[22 - 24], during sudden imposing the thermal boundary condition to a steady state, the transition of the boundary layer next to the sidewall was divided into three stages: 1) initial growth 2) an entrainment development and 3) steady stage. Boundary layer grows at the early stage. At the entrainment development stage horizontal movement is reported near the vertical boundary layer which took for long time. At last, when the steady stage reaches, a special kind of boundary layer structure (approximately double layer structure) of the vertical boundary layer is created with stratified interior fluid at the core. This is due to the transient behavior of sudden heating in natural convection of a cavity fill with air.

The available literature on the both steady and unsteady natural convection within air filled and porous saturated enclosures is performed within simpler geometry such as the rectangular, square, triangular etc. Some extensive investigations have been documented in [25 - 32]. For past decades, natural convection of air in modified enclosures with complex configurations has also been researched. The most distinctive works of Morsi and Das [33], Saha et al. [34], Noorshahi et al. [35], Yao [36], Mahmud et al. [37] and Hasan et al. [38, 39] on air natural convection in complex enclosures are available in the literature which describe the flow pattern within various geometries.

Saeid and pop [40] investigated unsteady natural convection in a 2D square cavity filled with a porous medium where side walls are suddenly heated and cooled. It is reported that the average Nu indicating an descending during the unsteady stage and that the time needed to reach the steady state stage is seemed to be longer for low Ra and shorter for high Ra. The main objective of this work is to extend the work done in [40] to modified the horizontal walls where the walls are adiabatic. Modifying the mentioned walls, we employ insulated blocks within the enclosure to control the Nusselt number related to the hot side wall. There are different cases for employing this method:

- *Using insulated block at the bottom edge*
- *Using insulated block at the top edge*
- *Using couple blocks for the both top and bottom edge*
- *Using insulated block within the media*

Displacement of the block for the mentioned case and reporting the Nusselt number variation with time is the main objective of the present study. Almost no information exists about this kind of modification in the literature which appears applicable in thermal systems.

Chapter 3

MATHEMATICAL MODELLING OF ENCLOSURE

Porous media model is applicable to an extensive variety of fields having to single phase and multi phase problems such as packed bed reactors and etc. Researchers usually define a cell zone in which the porous media assumptions and pressure loss related to the flow are determined via our inputs for the characteristics of the media. Heat transfer through this media is also predictable according to the thermal equilibrium of the media and the working fluid. As we are simulating the problem via the commercial software FLUENT, there are some special limitations and assumptions of porous media

3.1 System Description

Consider the flow of air within porous enclosure with height H as shown in Fig 3.1. In the current study, unsteady natural convection within a square porous saturated enclosure has been considered. The left and right walls are considered to be differentially heated and the flat top and bottom walls are assumed as adiabatic. The schematic diagram of the investigate enclosure has been shown in the Fig 3.1.

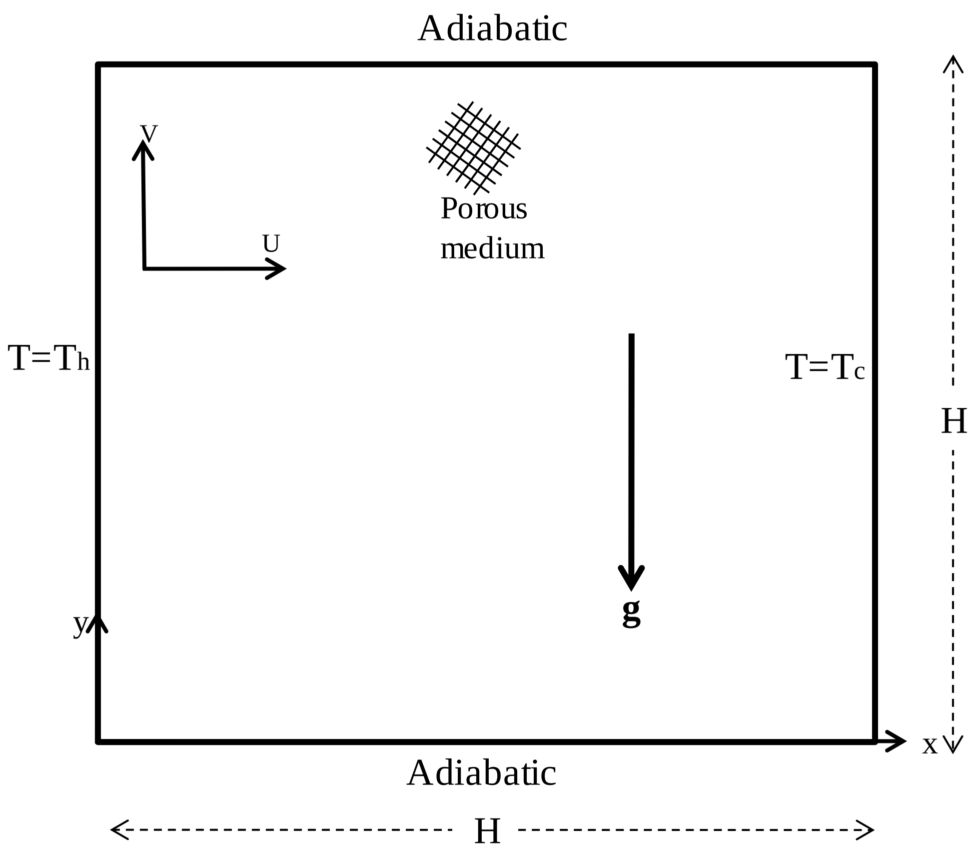

Figure 3.1. Schematic of the problem

Investigating parameters for thermal and flow behavior and heat transfer features such as Rayleigh number and porosity are kept constant for all cases. Ra of 1000 and porosity of 0.25 were considered as we are investigating presence of one/two insulated block within the enclosure. The most famous governing equations used by researchers are obtained considering following assumptions: (a) The entire enclosure is filled with porous material completely which is isotropic in thermal conductivity, (b) Darcy's law is applicable, (c) Normal Boussinesq incompressible fluid assumption is considered, (d) Inertia effects are negligible.

3.2 Governing Equations

3.2.1 Energy Equation in Porous Media

There are some modifications for classic energy equation for porous media which are related to the transient terms and conduction flux. As there is solid and fluid phase, a mean or an effective conductivity should be defined. Also the thermal inertia of solid phase has to be regarded as time dependent on the medium:

$$\frac{\partial}{\partial t}(\gamma\rho E_f + (1-\gamma)\rho_s E_s) + \nabla.(\vec{v}(\rho_f E_f + p)) = \nabla.\left[k_{eff}\nabla T - \left(\sum h_i J_i\right) + (\bar{\bar{\tau}}.\vec{v})\right] + S_f^h \qquad (3.1)$$

Where

E_f *= total energy of fluid*

E_f *= total energy of solid medium*

Υ *= porosity of the medium*

E_{eff} *= efficient medium thermal conductivity*

S_f^h *= source term related to fluid enthalpy*

3.2.2 Limitations and Assumptions of Porous Media Model

As the flow passing through this media encounters resistance, there is empirical relations for the related quantity of the loses. In fact, the model describing porous media applies momentum sink in the governing equations related to momentum. To represent a valuable model, following assumptions:

- *The porosity used in the model is assumed to be isotropic for both single phase and multi-phase flow.*
- *The resistances of porous media momentum and heat source terms are*

evaluated individually on each phase.

- *The interplay between a porous medium and shock waves are not deliberated.*
- *Thermal equilibrium is assumed between the porous media solids and multiphase fluid flows. The solids temperature is thus calculated by phase temperatures.*

3.2.3 Momentum Equation for Porous Media

A source term is added to the classic fluid flow equations for modeling porous media. This source term consists of two terms: a viscous loss term, and an inertial loss term.

$$S_i = -\left(\sum_{j=1}^{3} D_{ij}\mu v_j + \sum_{j=1}^{3} C_{ij}\frac{1}{2}\rho|v|v_j\right) \tag{3.2}$$

Where S_i is the source term for the momentum equation, $|v|$ is the magnitude of the velocity and D and C are matrices for viscous loss and inertial loss, respectively. Preparing a momentum sink in the porous cell, it generates pressure loss that is related to the fluid velocity (or velocity squared) in the computational cell. According to the homogeneous porous media assumption:

$$S_i = -\left(\frac{\mu}{\alpha}v_i + C_2\frac{1}{2}\rho|v|v_i\right) \tag{3.3}$$

α, C_2 are permeability and inertial resistance multiplier, simply specify D and C as matrices with $1/\alpha$ and C_2, respectively, on the diagonals. Power law assumption of velocity value is also applicable in FLUENT for the source term:

$$S_i = -C_0|v|^{C_1} = -C_0|v|^{(C_1-1)}v_i \tag{3.4}$$

Where C_0 and C_1 are user-defined empirical coefficients.

3.2.4 Darcy's Law in Porous Media

In general, laminar flow of porous media, pressure loss which is related to the velocity magnitude and C_2 may be regarded as zero. Darcy's law is defined in which *convective acceleration and diffusion is neglected, the porous media model change into:*

$$\nabla p = -\frac{\mu}{\alpha}\vec{v} \tag{3.5}$$

The software predicts pressure loss in each of the three (x, y, z) coordinate directions through the according relationship:

$$\Delta p_x = \sum_{j=1}^{3}\frac{\mu}{\alpha_{xj}} v_j \Delta n_x \tag{3.6}$$

$$\Delta p_y = \sum_{j=1}^{3}\frac{\mu}{\alpha_{yj}} v_j \Delta n_y \tag{3.7}$$

$$\Delta p_z = \sum_{j=1}^{3}\frac{\mu}{\alpha_{zj}} v_j \Delta n_z \tag{3.8}$$

Δn_x, Δn_y *and* Δn_z *are the thicknesses of the medium in the x, y, and z directions. Here, the thickness of the medium (*Δn_x, Δn_y *or* Δn_z*) is the real thickness of the porous region in our model.*

3.2.5 Inertial Losses in Porous Media

For high values of velocity, a modification is required for inertial losses within the media which is provided by the constant C_2 *in relation (3.1).* C_2 *is a coefficient per unit length along the flow direction. This constant admits the pressure loss to be determined as a function of dynamic head.*

3.2.6 Effective Conductivity in the Porous Medium

Volume average of the fluid conductivity and solid conductivity is used for average conductivity of the media:

$$k_{eff} = \gamma k_f + (1-\gamma)k_s \tag{3.9}$$

Where

Υ = *porosity of the medium*

k_f = *fluid phase thermal conductivity*

k_s = *solid medium thermal conductivity*

3.2.7 Effect of Porosity on Transient Scalar Equations

Evaluating unsteady parameters for porous region, the effect of porosity on the time-derivative terms is calculated for in all scalar transport equations and the continuity equation. Time-derivatives evolve into when the effect of porosity is taken into account, where φ is the scalar quantity and γ is the porosity.

3.2.8 Numerical Procedure and Boundary Conditions

Under the above assumption, the non-dimensional governing equations in terms of the stream function (ψ) and temperature (θ) are:

$$\frac{\partial^2 \psi}{\partial X^2} + \frac{\partial^2 \psi}{\partial Y^2} = -Ra \frac{\partial \theta}{\partial X} \tag{3.10}$$

$$\frac{\partial \theta}{\partial \tau} + \frac{\partial \psi}{\partial Y}\frac{\partial \theta}{\partial X} - \frac{\partial \psi}{\partial X}\frac{\partial \theta}{\partial Y} = \frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} \tag{3.11}$$

Where dimensionless variables are defined by:

$$X = \frac{x}{H}, \quad Y = \frac{y}{H}, \quad \tau = \frac{t}{\sigma H^2/\alpha}, \quad U = \frac{u}{\alpha/H}$$
$$V = \frac{v}{\alpha/H}, \quad \psi = \frac{\Psi}{\alpha}, \quad \theta = \frac{T - T_C}{T_H - T_C}, \quad Ra = \frac{g\beta \Delta T K}{\alpha \upsilon} \tag{3.12}$$

The non-dimensional stream function, ψ, satisfies the following equations

$$U = \frac{\partial \psi}{\partial Y}, \quad V = -\frac{\partial \psi}{\partial X} \tag{3.13}$$

Equation (1) and (2) are subject to the following boundary conditions:

Bottom surface $$U = 0, \quad V = 0, \quad \frac{\partial \theta}{\partial Y} = 0 \tag{3.14}$$

Top surface $$U = 0, \quad V = 0, \quad \frac{\partial \theta}{\partial Y} = 0 \tag{3.15}$$

Left surface $$U = 0, \quad V = 0, \quad \theta = 1 \tag{3.16}$$

Right surface $$U = 0, \quad V = 0, \quad \theta = 0 \tag{3.17}$$

The working fluid has been chosen as Pr = 0.71. The physical quantities of interest in this problem are the average Nusselt number along the hot wall, defined by

$$Nu = -\int_0^H \frac{hy}{k} dy = -\int_0^1 \frac{y}{H}\left(\frac{\partial \theta_s(Y)}{\partial X}\right)_{X=0.1} dY = -\int_0^1 Y\left(\frac{\partial \theta_s(Y)}{\partial X}\right)_{X=0.1} dY \tag{3.18}$$

3.3 Numerical Procedure

3.3.1 Numerical Procedure

The governing normalized coupled equations of (3.1) and (3.2) with the boundary conditions of Eq. 6 were solved using Finite Volume Method employing commercial software FLUENT. The Poisson like momentum equation (3.1) and the energy equation (3.2) are discretised using the central difference but the time derivative is discretised employing the three points backward difference relationship to ensure the second order accuracy in both time and space. The computational domain has been discretized with Nonuniform grids of rectangular elements with smaller grids in regions near the heated and cooled walls (Fig 3.2).

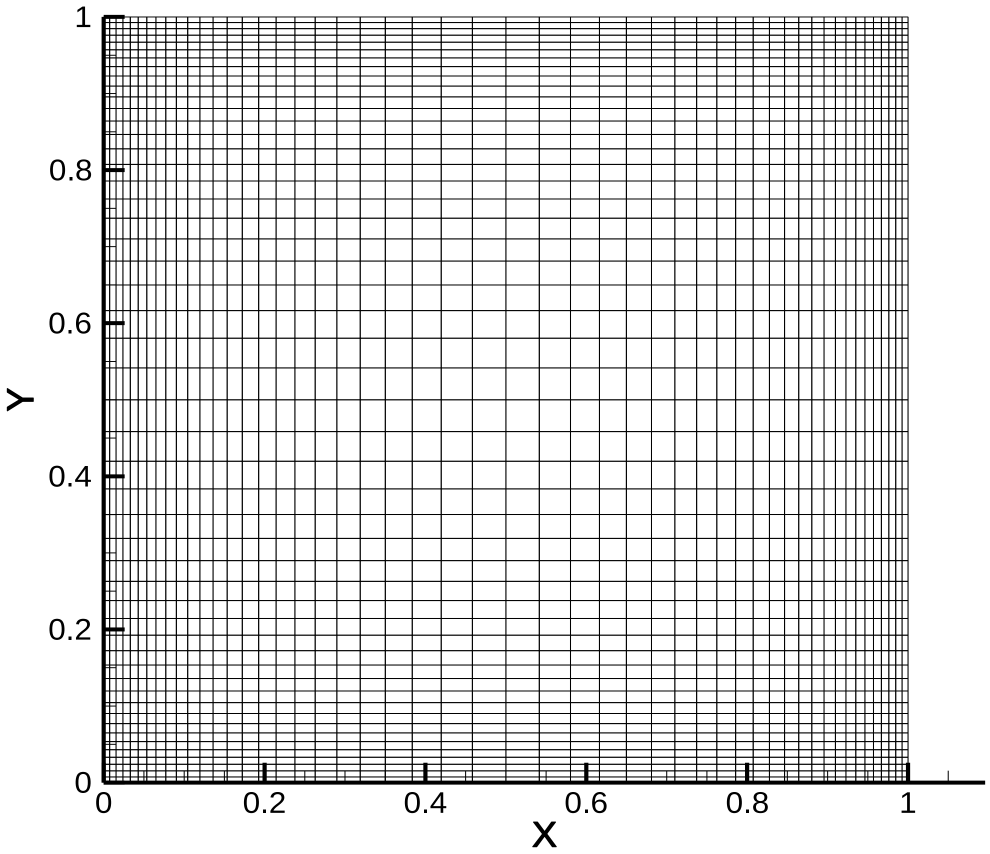

Figure 3.2. Grids used for the simulations

The convergence of the solution is assumed when the relative error for each variable between the iterations is reported through the below convergence criterion:

$$\left| \frac{\Gamma^{n+1} - \Gamma^{n}}{\Gamma^{n}} \right| \leq 10^{-6} \tag{3.19}$$

3.3.2 Grid Refinement Check

To evaluate grid independence of the current solution, many numerical runs are carried out. These experiments suggest that a structured spaced grid of 40000 elements as shown in Fig 3.2 is adequate to capture correctly the flow and heat transfer process inside the enclosure.

Chapter 4

RESULTS AND DISCUSSION

4.1 Validation

In this section, we investigate truthfulness of the method we use for main part of study. In order to obtain insurance for the results, we simulate a benchmark problem of a simple square heating from left and cooling from right side and working fluid is air. This problem has been studied by many researchers. Table 4.1 shows that the mean Nu for the hot wall obtained from our simulations and those results numerically or experimentally obtained by other scientists. It is clear that for the three studied Ra number, the mean Nu are close to each other and a negligible difference with other works exists.

Table 4.1. Comparison of Nu number at steady state for various Ra number

Ref.	*Nu*		
	Ra=100	*Ra=1000*	*Ra=10,000*
Bejan [6]	*4.200*	*15.800*	*50.800*
Manole and Lage [9]	*3.118*	*13.637*	*48.117*
Baytas [4]	*3.160*	*14.060*	*48.330*
Saeid and Pop [46]	*3.002*	*13.762*	*43.953*
Present study	*3.105*	*13.019*	*43.948*

Temperature rising near the hot wall is studied here to show how non dimensional temperature enhances after differentially heating the wall. Figure 4.1 illustrates that the temperature rising consists of three main parts. This first initial rising indicates a pure conduction for the initial time of heating. Then it passes through a transitional stage. Previous studies have revealed that this part is oscillating for air filled cavities, but for the present cavity, porous media and the resistance along the flow ruin the oscillating behavior and a smooth curve is created for transitional stage. Finally, the flow assuages to a steady state condition.

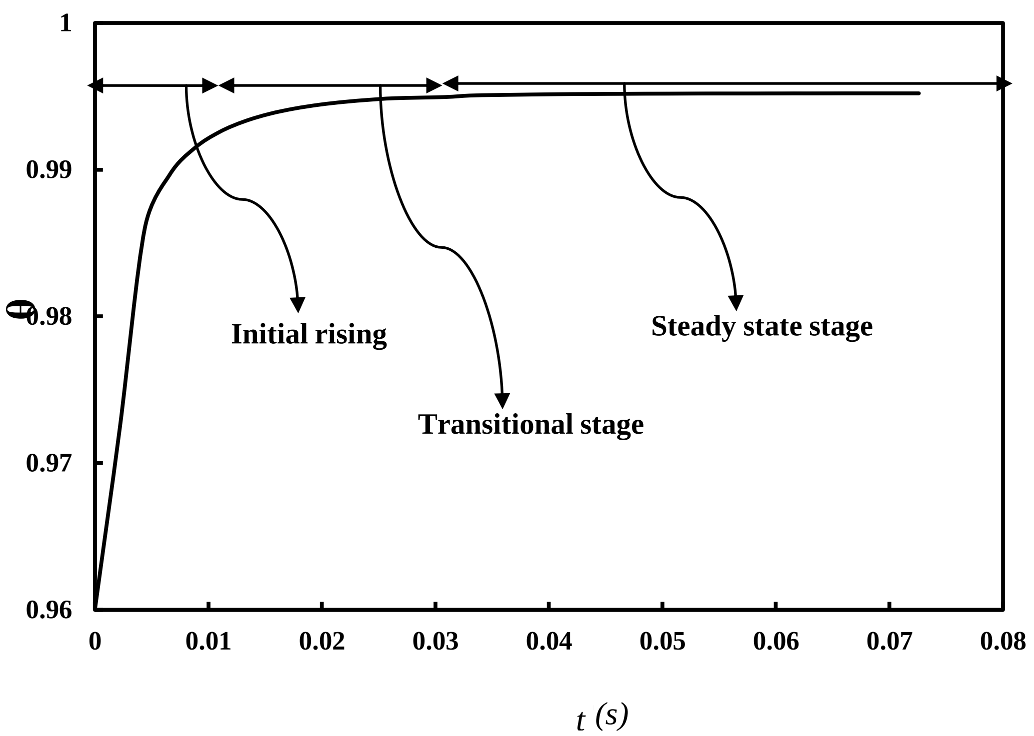

Figure 4.1. Dimensionless temperature change near the hot wall

4.2 Presence of Insulated Block

4.2.1 Presence of Insulated Block at the Bottom Edge

This part is devoted to study the effect of insulated block presence at the bottom edge on mean Nu of hot side. Firstly we move the block at the bottom edge, then to see the *influence of block height, we change its height when it stands at the middle edge. Figure 4 shows the schematic of the physical description of the problem.*

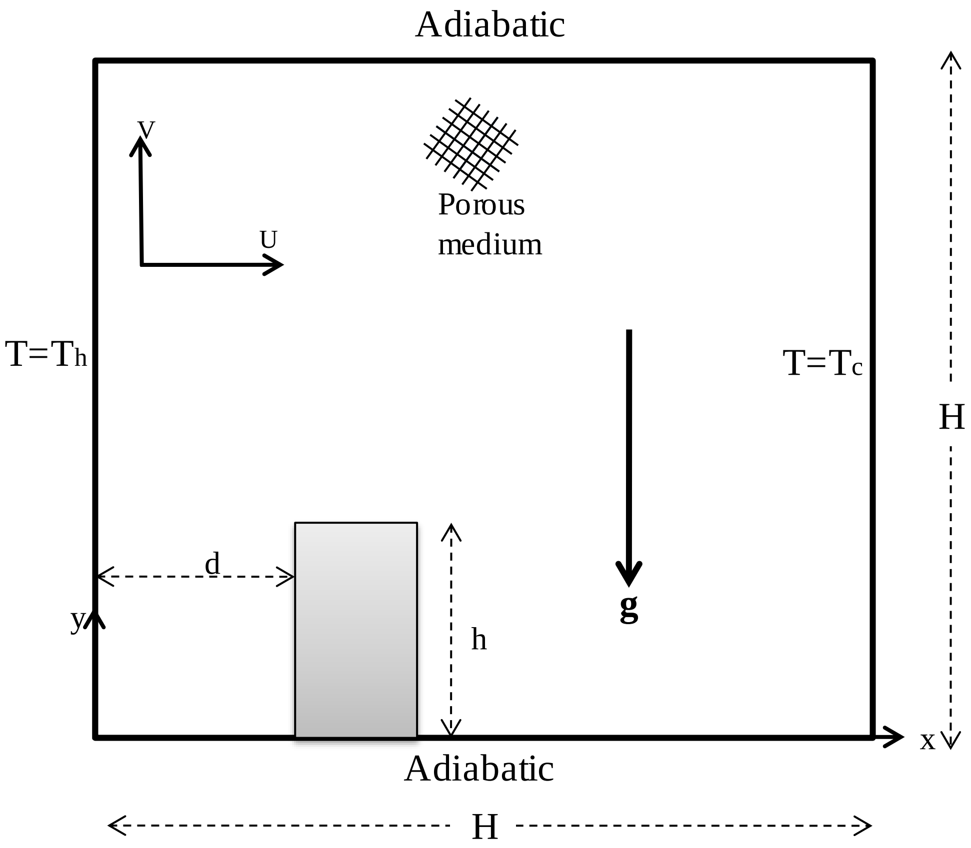

Figure 3.2. Block at the bottom edge

As a matter of fact, when a thermal system like this cavity starts heating inside, the working fluid is induced to move upward adjacent to the hot wall. Then a hydrodynamic boundary layer grows and resists thermal communication of hot side and the fluid moving upward. So we expect that the mean Nu which demonstrate rate of heat transfer coefficient in the problem reduce gradually as the time progresses until it assuages to a steady state status. But the investigating parameters will affect on the value of mean Nu and the time it takes to reach the steady state condition.

All the mean Nu are divided by the mean Nu of a simple square with no block. Almost all cases will have less value of Nu with respect to the base case due the decreased distance of hot and cold walls on behalf of insulated block presence. This is what we are looking for; to control and manage amount of heat transferred through the media.

Figure 4.3 shows the mean Nu of hot wall for three conditions of the block position. As seen from the figure, the mean Nusselt number decreases when the block gets closer to the hot wall showing an undershoot. This can be attributed to the gap between the block and hot wall when the distance is low. This gap can be a place of vortex motion like of the flow which reduces the thermal communication of the hot and cold walls. It also is clear that this phenomenon needs much time to assuage steady condition due to the mentioned reason.

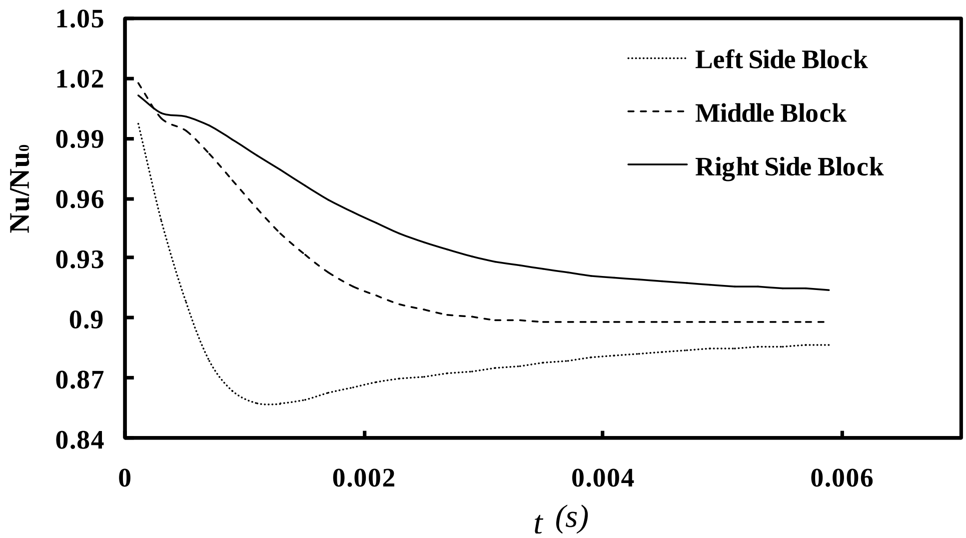

Figure 4.3. Mean Nu of hot wall when the block is displaced at the bottom edge

Now, block is installed firmly to the bottom middle edge with variable height. Height of the block is varied from 0.25 of the side length to 0.75H. Fig.4.4 indicates the results of this case. It can be understood that the mean Nu of hot wall decreases with the increase of block height. Increasing the height of block not only reduces the porous conductive and convective media for mass and heat transport, but also reduces the thermal communication of hot and left walls. It is clear that the increment of the height is also effective in time needed to get a steady status. This can be explained as the reason of previous cases due to the generation of a large vortex besides the block which leads to reach a steady state later.

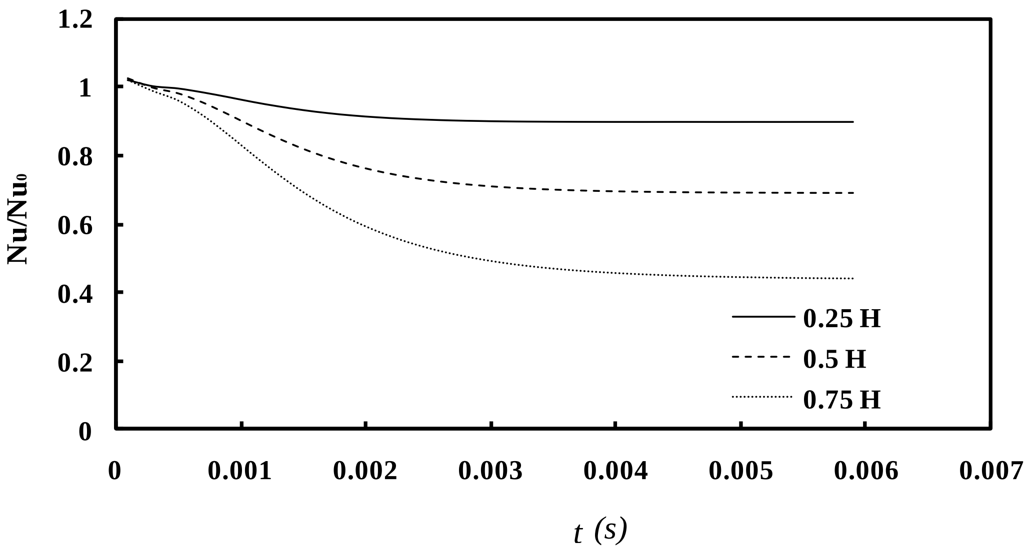

Figure 4.4 Mean Nu of hot wall when the height of block is changed

4.2.2 Presence of Insulated Block at the Top Edge

In this part, the insulted block is positioned at the top edge (Fig 4.5). The place and the height of the block is investigated and results are illustrated as following.

Figure 4.6 shows the unsteady results of mean Nu of hot wall for different place of block at top edge. Similar pattern for Nu is seen for the top installed insulated block.

The right block scenario has the highest mean Nu (Fig 4.7). For height variation of the block, we report the results likewise the bottom installed block.

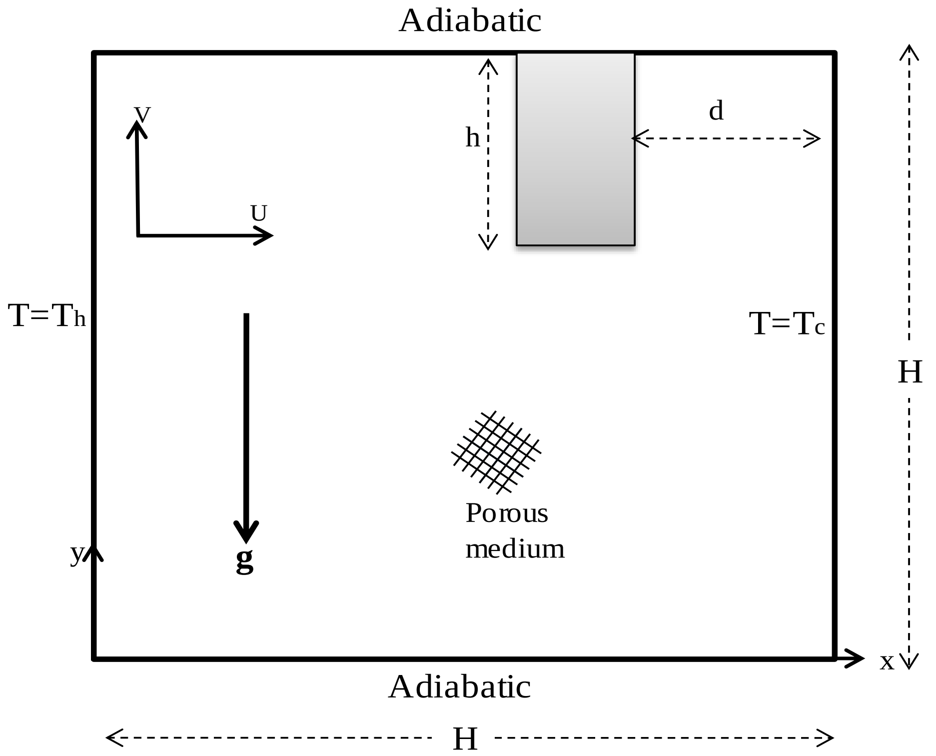

Figure 4.5. Block at the top edge

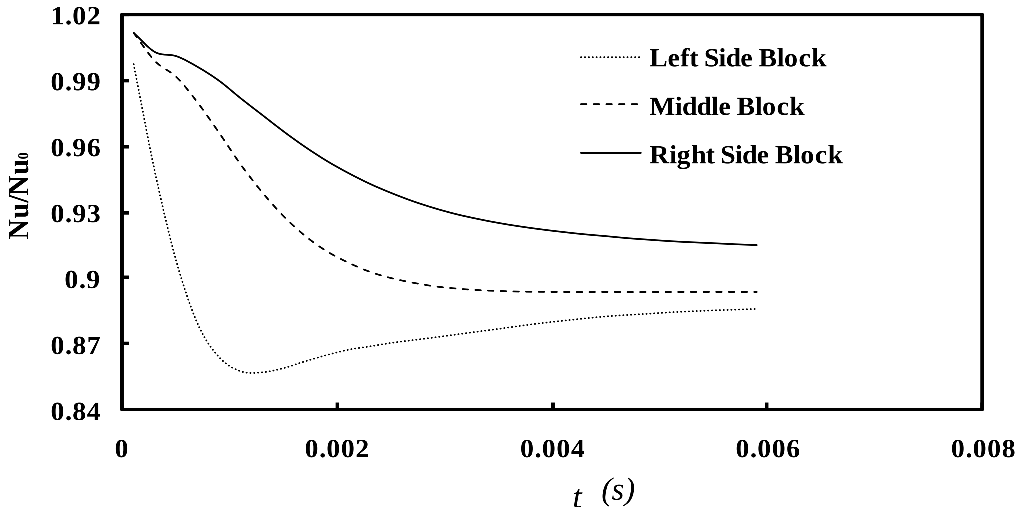

Figure 4.6. Mean Nu of hot wall when the block is displaced at the top edge

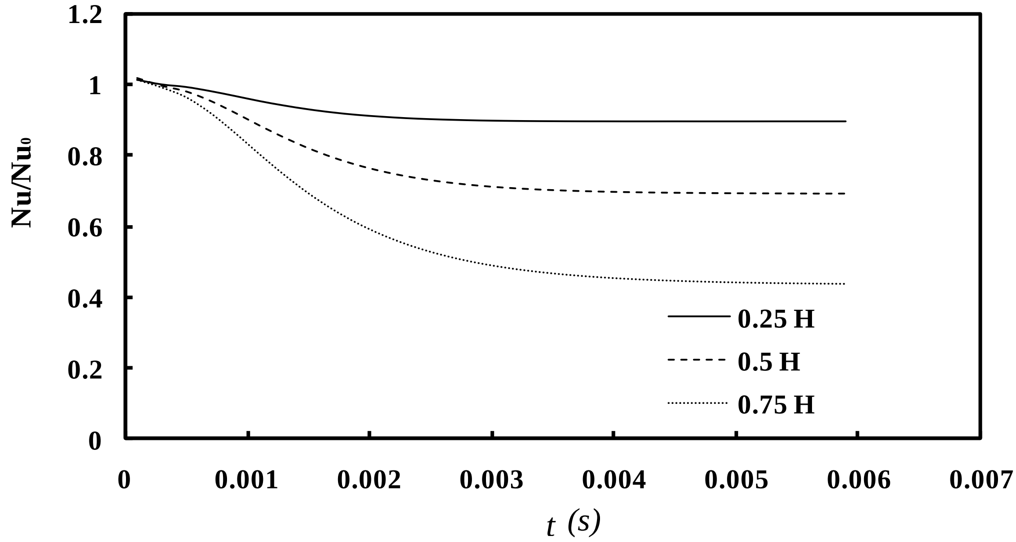

Figure 4.7. Mean Nu of hot wall when the height of block is changed

4.2.3 Presence of two Insulated Blocks at the Top and Bottom Edges

A couple of insulated blocks are used to see the related effect on mean Nu of hot wall (Fig 4.8). The both blocks are installed in front of each other and their position from cold wall and their height are the varying cases.

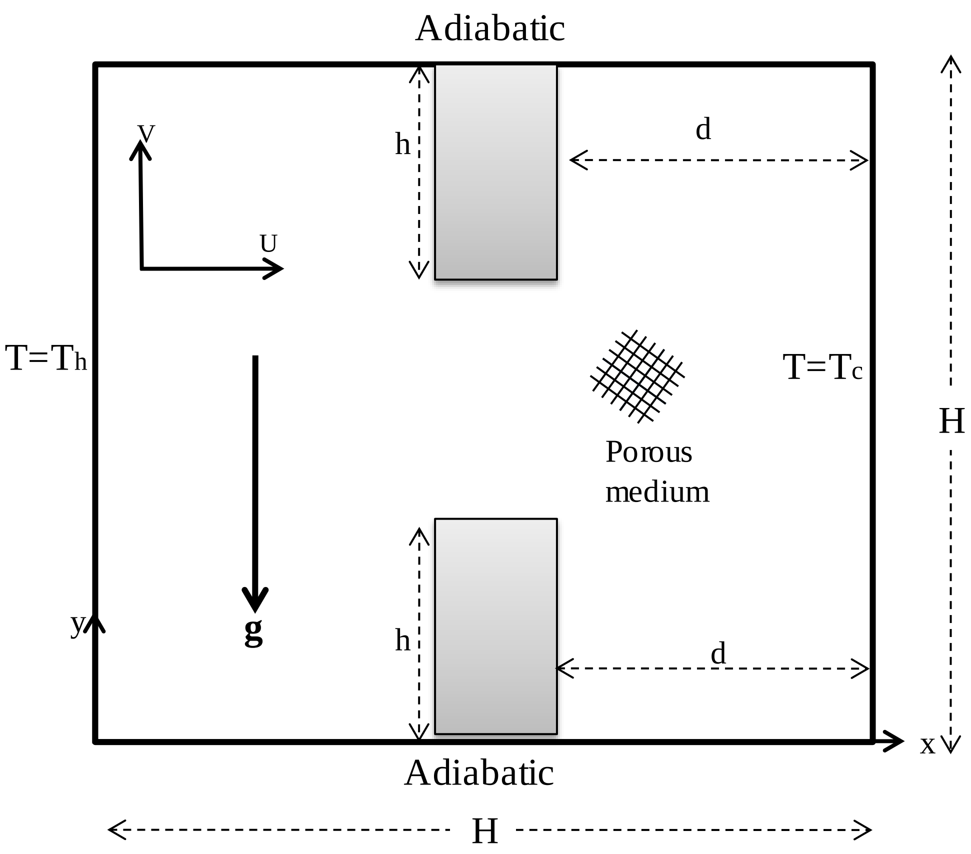

Figure.4.8. Couple blocks in front of each other

As seen from Fig 4.9 mean Nu value is almost near each other and for the right installed blocks, there is an undershoot for the initial time of heating but then it reaches the steady state status. It is also clear that increasing the blocks height, much more porous media is occupied and thereby less mean Nu is concluded (Fig4.10).

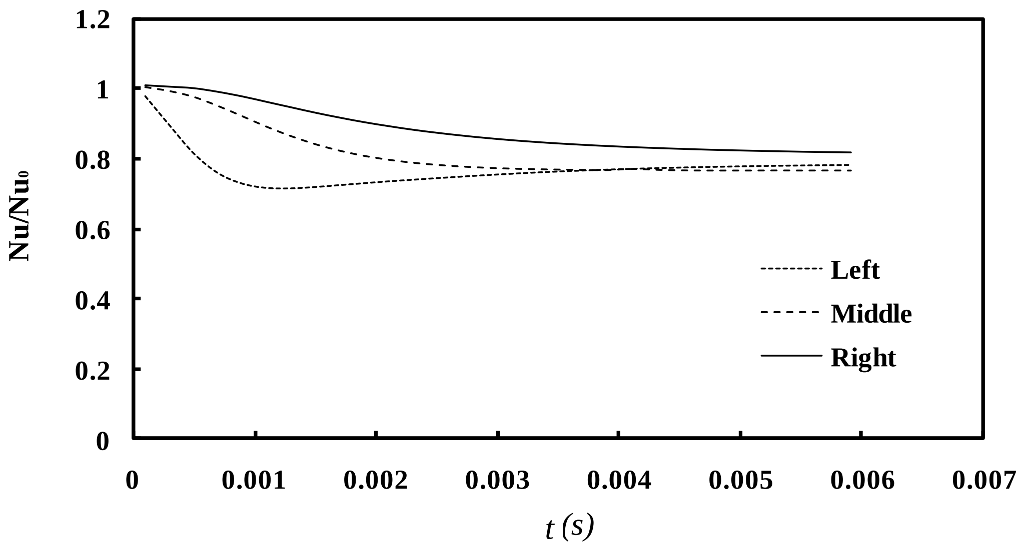

Figure 4.9. Displacement of couple blocks

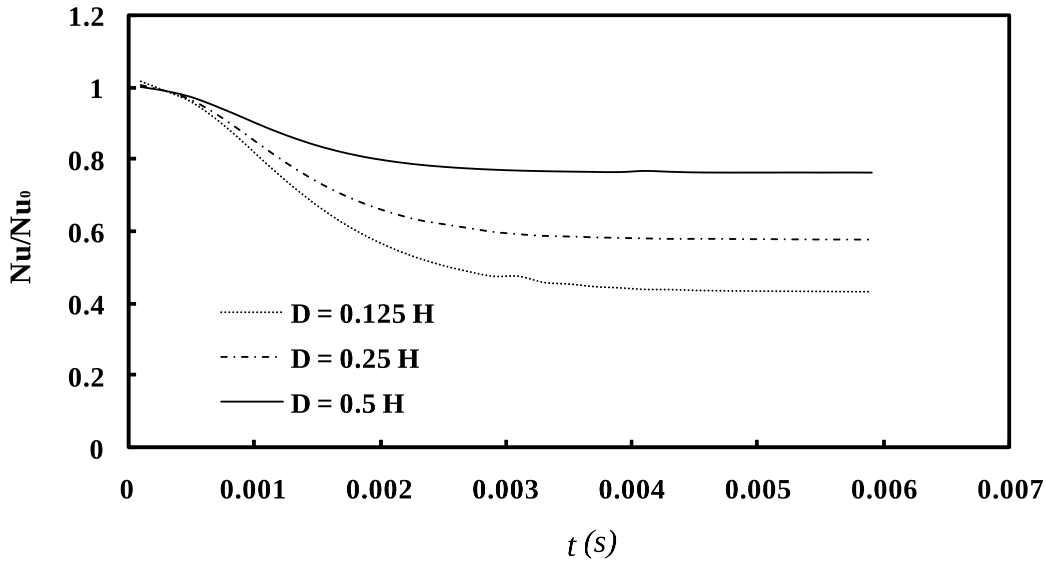

Figure 4.10. Distance of couple blocks

4.2.4 Presence of Block within the Media

Now the last part is devoted to study the effect of block while is within the media (Fig 4.11). Firstly the height is varied and then replacing the block vertically and horizontally is considered. With the increase of height within the media, mean Nu is decreased and the reason is similar to reason of cases which occupy the media more (Fig 4.12). But the values of all cases are comparable.

Figure 4.13 shows that moving the block to the upside or downside (No attachment to the edges) results in a similar mean Nu of a simple square with no blocks. Although the media is occupied, decreasing the fluid's flow area besides the top and bottom edges results in higher value of velocity and this phenomenon can repair the decreased Nu value. Similar pattern is seen when the block is moved horizontally (Fig 4.14).

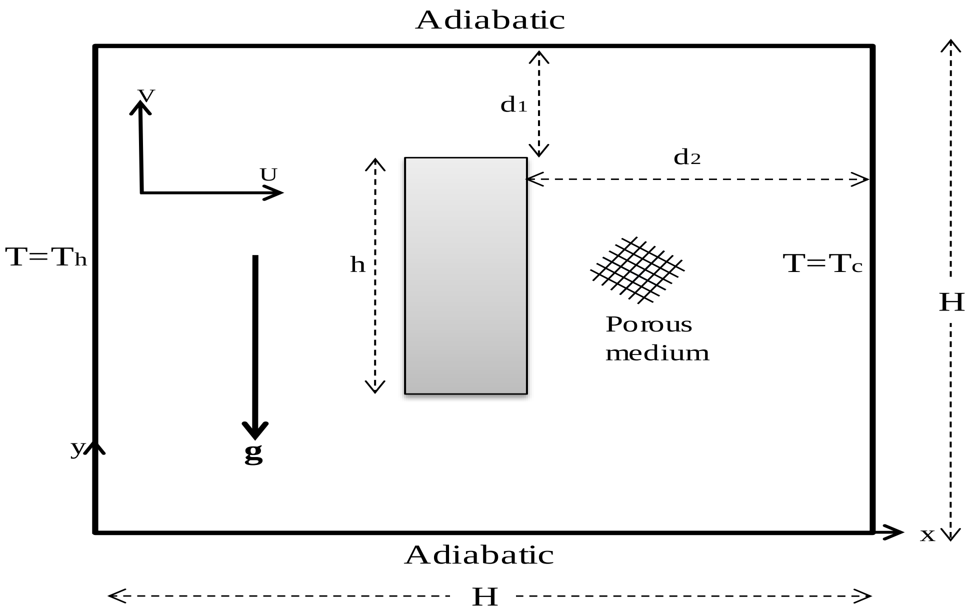

Figure 4.11. The block within the media

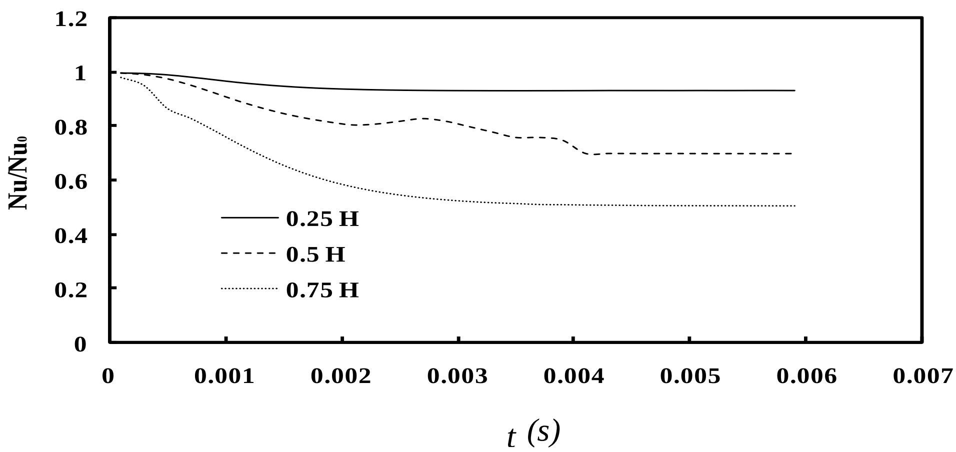

Figure 5.12. Height varıatıon within the media

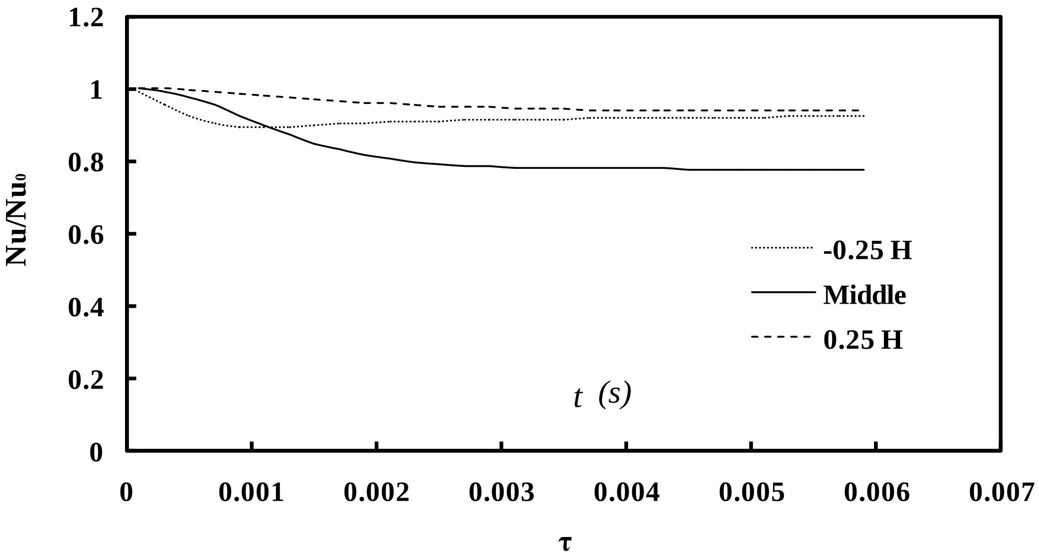

Figure 4.14 Horizontal variation of the block within the media

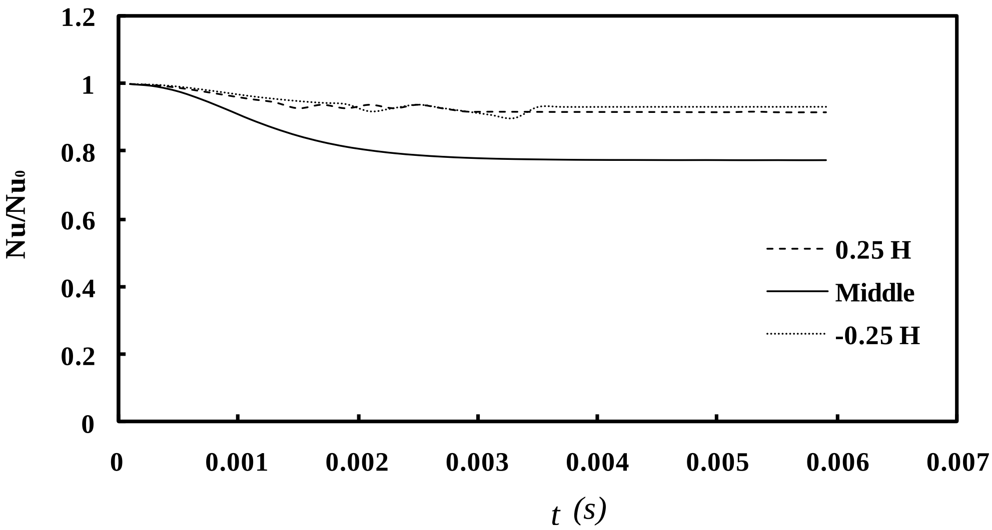

Figure 4.13. Vertical variation of block within the media

Chapter 5

CONCLUSION

Present study deals with the problem of free convection within porous filled enclosure. A simple Square with hot left and cold right sides including two top and bottom insulated supposed for the current study. Numerical simulations were performed to investigate some specified ways to control the amount of heat transferred through the media which has many industrial applications such as packed bed reactors. Firstly results obtained for a benchmark problem of Simple Square of porous filled using FLUENT were validated with previous studies. Then the effect of presence of insulated block / blocks within the media at various positions were studied. Transient Nusselt number of hot wall investigated for all cases where the energy is entered the media. Results showed that almost all cases Nu is reduced from initial time to a steady state time due to the growth of thermal and hydro dynamical boundary layer. Also employing block/blocks within the media reduces the mean distance of hot and cold walls, so Nu is conveniently controlled at the required amount.

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