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LAMINAR NATURAL CONVECTION HEAT TRANSFER AND AIR FLOW IN CUBIC ENCLOSURES WITH PIN FINS ATTACHED TO THE HOT WALL

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ABSTRACT

In this study, laminar natural convection heat transfer and air flow in cubic enclosure is studied numerically. The enclosure is heated from one lateral wall while the enclosure is cooled from the opposite lateral wall; other walls are adiabatic. Cylindrical pins are attached to the heated wall to enhance heat transfer from the air-filled enclosure. Three-dimensional steady-state continuity, Navier-Stokes and energy equations along with the Boussinesq approximation are solved using FLUENT®. The validity of the numerical solution is carried out in comparison to other 3D natural convection benchmarks. The Finite Volume Method (FVM) with SIMPLE algorithm was used. The Rayleigh numbers considered in this study range from 10^5 to 10^7. The dimensionless pin diameters and the lengths considered in this study are D=0.025, 0.050 ve 0.1, and B=0.1, 0.2 and 0.3, respectively. In a pin array, the number of pins of 2, 4, 6, 8 and 10 are arranged in in-line horizontal and vertical configurations. The air flow pattern and the temperature fields are obtained for various Rayleigh numbers, pin length, pin diameter and the number of pins; and for each case, the mean Nusselt numbers for the cold surface are computed.

INTRODUCTION

Natural convection in enclosures is one of the topics concerning thermal and mass transport processes involved in various types of engineering systems such as thermal insulation in buildings, fire hazards, solar energy collectors, cooling chips in electronic or computer equipment etc. For the case of a simple 3D rectangular enclosure without any partition, obstruction, or local heating, only very limited studies have been conducted during the past few decades in comparison to 2D studies mainly due to the limitations of the computer memory and/or cpu speed.

NOMENCLATURE

b	[m]	Pin length
B	[-]	Dimensionless pin length
d	[m]	Pin diameter
D	[-]	Dimensionless pin diameter
g	[m/s^2]	Gravitational acceleration
h	[W/m^{2}K]	Heat transfer coefficient
H	[m]	Height of the enclosure
k	[W/mK]	Thermal conductivity
k	[-]	Unit vector of z-direction
ℓ	[m]	Side length of enclosure
L	[-]	Dimensionless length of the cubic enclosure
Nu	[-]	Nusselt number
p	[N/m^2]	Pressure
P	[-]	Dimensionless pressure
Pr	[-]	Prandtl number
Ra	[-]	Rayleigh number
t	[K]	Temperature
T	[-]	Dimensionless temperature
u,v,w	[m/s]	Velocity components
U,V,W	[-]	Dimensionless velocity components
V	[-]	Dimensionless velocity
x,y,z	[m]	Cartesian coordinate system
X,Y,Z	[-]	Dimensionless coordinate system

Special characters

α	[m^2/s]	Thermal diffusivity
β	[K^{-1}]	Thermal expansion coefficient
μ	[N.s/m^2]	Dynamic viscosity
υ	[m^2/s]	Kinematic viscosity
ρ	[kg/m^3]	Density of air

Subscripts

b	Bare
c	Cold
h	Hot
p	With pins

Natural convection studies in 3D enclosures are becoming more common in the last decade. Ravnik et al. [1] studied a natural convection phenomenon in cubic and parallelepipedal inclined enclosures. The simulation of coupled laminar viscous flow and heat transfer is performed using an algorithm based on a combination of single domain Boundary Element Method (BEM). Lo et al. [2] proposed differential quadrature method to simulate natural convection in an inclined cubic cavity using velocity–vorticity form of the Navier–Stokes equations. Test results obtained for an inclined cubic cavity with different angle of inclinations for Rayleigh number equal to 10^3, 10^4, 10^5 and 10^6. Frederick and Quiroz [3] numerically studied the natural convection in cubical enclosures with vertical, square, frontally opposed sources of different sizes and temperatures. He et al. [4] numerically studied laminar natural convection in a cubic enclosure with a cold vertical wall and two hot square heaters with constant temperature on the opposite wall. The effect Rayleigh number, aspect ratio of cubical enclosure and Prandtl number were examined. Ravnik et al [5] presented acceleration and computer memory reduction of an algorithm for the simulation of laminar viscous flows and heat transfer.

Tric et al. [6] presented accurate solutions to the differentially heated cubic cavity problem for values of Rayleigh number in the range 10^3—10^7. From mesh refinements and extrapolations, the spatial resolution of the data is claimed to be better than 0.02% in relative spatial error at the highest Rayleigh number. Lee et al. [7] analyzed numerically natural convection in three-dimensional rectangular enclosures. The effect of the Rayleigh number and two dimensional approximation limit were investigated. The results showed that the temperature disturbance imposed on the end wall reinforces the axial flow and magnifies the three-dimensional effect. Bairi [8] worked the problem both as a numerical and experimental. Working one side of the square edges was kept in hot constant temperature; the opposite side was kept in cold constant temperature. Rayleigh number and inclination angle of the square box with different values of the Nusselt number of changes were observed.

Nasa et al. [9] investigated experimentally natural convection heat transfer and fluid flow in horizontal and vertical narrow enclosures with heated rectangular finned base plate, for different fin spacing and fin lengths. They found that increasing the fin length increases Nusselt number and finned surface effectiveness. Mobedi et al. [10] numerically investigated steady state natural convection heat transfer in a longitudinally short rectangular fin array on a horizontal base. The fin length and fin height were varied from 2 to 20 and 0.25 to 7-fin spacing, respectively. Frederick et al. [11] numerically investigated 3D natural convection of air in a cubical enclosure with a fin on the hot wall for Rayleigh numbers of 10^3–10^6. The fin was placed horizontally on the hot wall. Bilgen et al. [12] numerically studied the natural convection heat transfer in inclined rectangular enclosures with perfectly conducting fins attached to the heated wall. A numerical study of 3D natural convection in a differentially heated cubical enclosure was performed by Fugesi et al [13]. A numerical study was carried out by Lakhal et al [14] in differentially heated square cavities, which were formed by horizontal adiabatic walls and vertical isothermal walls. A thin fin was attached on the hot wall. A parametric study is investigated using following parameters: Rayleigh number from 10^4 to 10^9, dimensionless thin fin length from 0.10 to 0.90, dimensionless thin fin position from 0 to 0.90, dimensionless conductivity ratio of thin fin from 0 (perfectly insulating) to 60. It was found that there was always an optimum fin position which natural convection heat transfer was minimized. Silva et al. [15] examined the effects of heat transfer with rectangular fin attached to the hot wall of a 3D cube in laminar natural convection. The numerical results showed that for an enclosure assisted by a large volume fraction fin, the fin aspect ratio does not play an important role, and the average heat flux transferred to the fluid increases monotonically with the fin horizontal length. Chuang et al [16] analyzed the three-dimensional laminar natural convection flow with three chips at various positions by employing the PHOENICS code. Frederick [17] numerically examined natural convection of air in a cubical enclosure with a thick partition fitted vertically on the hot wall for Rayleigh numbers of 10^3–10^6. Dialameh et. al. [18] examined numerically natural convection from an array of aluminum horizontal rectangular thick fins attached on a horizontal base plate. Effect of various fin geometries and temperature differences on the convection heat transfer from the array was determined. Yüncü and Anbar [19] and Güvenç and Yüncü [20] reported experimental studies on natural convection heat transfer of rectangular fins attached to horizontal or vertical surfaces. The separate roles of fin height, fin spacing and fin base to ambient temperature difference were investigated. Fujii [21] studied the natural convection heat transfer from a vertical heated plate with inclined fins, attached on the vertical heated plate to isolate a hot air flow from a cold air flow. Experiments were performed in air for various inclination angles. Tou et al [22] studied a 3D computational domain for natural convection cooling on a 3-by-3 array of discrete heat sources flush-mounted on one vertical wall of a rectangular enclosure filled with various liquids (Pr = 5, 9, 25 and 130) and cooled by the opposite wall. Terekhov [23] numerically examined the heat transfer in a high vertical narrow enclosure in the presence of fins on one of the isothermal side walls. Liu et al [24] reported a numerical analysis of optimum spacing problems for three heated elements mounted on a vertical substrate using an operator-splitting time stepping finite element method. Giri et al [25] studied numerically natural convection heat and mass transfer over a shrouded vertical fin array for various fin spacings. Wang and Mayinger [26] presented the results of an experimental study of natural convective air cooling in one type of electronic equipments which contain several printed circuit boards (PCB's) set up vertically and parallel to one another in an impervious casing.

In the present study, laminar natural convection heat transfer and air flow in a cubic enclosure are numerically studied. The enclosure is heated from one lateral wall while the enclosure is cooled from opposite lateral wall. Cylindrical pin fins are attached to the active heated wall to enhance heat transfer. This study differs from previously studies in that air-filled cubical enclosure with in-line horizontally and vertically arranged circular fins are attached to heated lateral wall. The temperature field is obtained for various Rayleigh numbers, pin length, pin diameter and the number of pins. The mean Nusselt numbers for the cold surface are computed for each case. The effect of pin number, pin length, pin diameter and pin configuration is investigated.

MATHEMATICAL MODEL

The three-dimensional cubic enclosure which is considered is shown in Figure 1. The left and right lateral walls are kept at uniform temperatures equal to t_h and t_c respectively. The top, bottom, front and rear walls are assumed to be perfectly adiabatic. The temperature difference between the t_h and t_c walls promotes buoyancy driven flows inside the enclosure. The temperature difference is also assumed to be small enough so that the Boussinesq approximation is valid.

The steady-state Equations for a Newtonian fluid are given as, for the continuity

$$\nabla \cdot \mathbf{V} = 0 \tag{1}$$

for the momentum equations

$$(\mathbf{V} \cdot \nabla)\mathbf{V} = (\nabla^2 \mathbf{V} - \nabla P)(\mathrm{Ra}/\mathrm{Pr})^{-1/2} + T\,\mathbf{k} \tag{2}$$

and for the energy

$$(\mathbf{V} \cdot \nabla)T = (\mathrm{Ra}\,\mathrm{Pr})^{-1/2}\nabla^2 T \tag{3}$$

where $\mathrm{Pr} = \upsilon/\alpha$ and $\mathrm{Ra} = g\beta(T_h - T_c)H^3/\upsilon\alpha$ are the Prandtl and Rayleigh numbers, respectively.

The following dimensionless quantities are introduced to obtain the non-dimensioned Equations (1)-(4):

$$(X,Y,Z,L) = \frac{(x,y,z,\ell)}{H}, \quad (U,V,W) = \frac{(u,v,w)H}{\alpha\sqrt{Ra}\,\mathrm{Pr}},$$
$$P = \frac{pH^2}{\mu\alpha\sqrt{Ra}\,\mathrm{Pr}}, \quad T = \frac{t - t_c}{t_h - t_c} \tag{4}$$

Boundary conditions are **V**=0 for all solid walls, including pin walls, and $\partial T/\partial n$=0 for the adiabatic walls where n is the perpendicular direction to pertinent wall. Hot and cold wall temperatures are shown on Figure 1.

The mean Nusselt number is computed over the cold wall as

$$\mathrm{Nu} = \frac{hH}{k} = \int_{X=0}^{1}\int_{Z=0}^{1} \frac{\partial T}{\partial Y} dX dZ \tag{5}$$

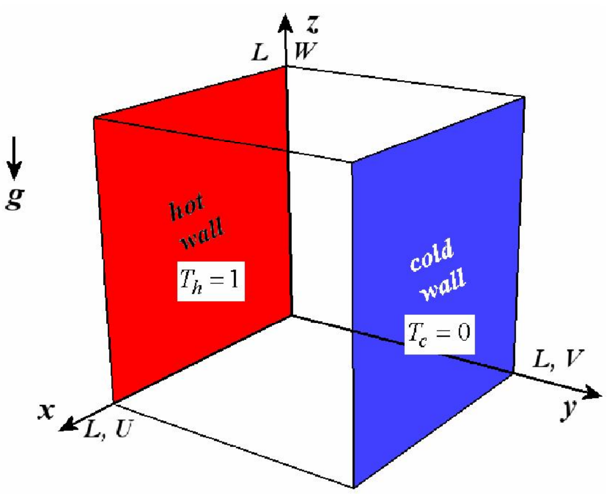


Figure 1 The geometry and the coordinate system of cubic enclosure

NUMERICAL SOLUTION

The differential system composed of Eqs. (1)–(3) was solved in the cubical domain of volume L×L×L=1 using Finite Volume Method present in FLUENT® code [27]. The differencing scheme of the transport terms was chosen to be "Second Order Upwind", and the SIMPLE algorithm was used.

Grid Sensitivity

A cubic enclosure without pins (bare cubic enclosure) was first solved primarily to determine the grid sensitivity and to compare to the present problem tackled. In Table 1, using equispaced homogeneous grids, the computed mean Nusselt numbers for bare cubic enclosure for Ra=10^5 and 10^6 are given. The convergence criterion for the pertinent equations was chosen as 10^{-5}. In this study, grid configuration of 80^3 was determined to be an optimum configuration, however, in the present problem with attached pins, mesh refinement near the pins were applied as well.

Table 1 Grid Sensitivity table for bare cube

Rayleigh Number	Mean Nusselt Number			
	N_x=N_y=N_z			
	10^3	20^3	40^3	80^3
10^5	5.2202	4.7164	4.4363	4.3598
10^6	7.8281	10.1230	9.2441	8.7945

A comparison of the mean Nusselt numbers for a bare enclosure (see Figure 1) and for various Rayleigh numbers is given in Table 2. It is observed that the agreement of the present solutions is excellent with those of provided in the literature.

Table 2 Comparison of the Nusselt number for various Rayleigh numbers

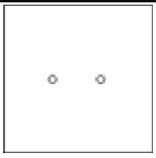
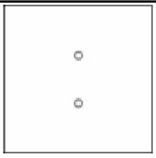
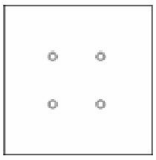
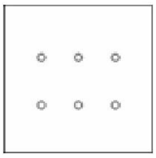
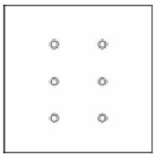
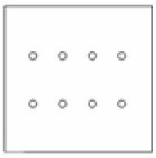
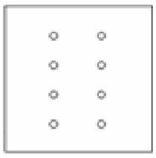
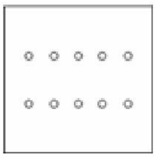
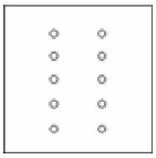
Ra	Present Study	Ref. [1]	Ref. [4]	Ref.[6]	Ref.[11]
10^3	1.0706	1.0713	1.064	1.0700	1.0712
10^4	2.0575	2.0591	2.060	2.0542	2.0570
10^5	4.3598	4.3570	4.400	4.3370	4.3534
10^6	8.7945		8.912	8.6407	8.7400
10^7	17.267			16.343	

Heat Transfer Characteristics

In the case of 3D cubic enclosure with pins, the in-line cylindrical pin arrangements that are considered in this study are illustrated in Table 3. The 4-pin arrangement possess a single case due to symmetry with respect to horizontal and vertical arrangement. The pins of length b and diameter d are arranged by equispaced two-rows or two-columns. The dimensionless pin length and pin diameter are defined as B=b/H and D=d/H. In order to study the effect of the number and the arrangement of the pins on the heat transfer rate from the cold wall, a parameter called Nusselt Number Ratio (NNR) defined by Eq. (6) is introduced. In view of this parameter, we can state that the heat transfer is enhanced if the value of this parameter is greater than 1, or a reduction of heat transfer is indicated when NNR is less than 1. Thus the mean Nusselt number for the bare cubic and the pinned-cubic enclosures are obtained for the Rayleigh number range studied.

$$\mathrm{NNR} = \overline{\mathrm{Nu}}\Big|_{\mathrm{pinned}} / \overline{\mathrm{Nu}}\Big|_{\mathrm{bare}} \tag{6}$$

Table 3 The in-line pin array arrangements considered

Configuration	In-line arrangements Horizontal	Vertical
2-pin		
4-pin		
6-pin		
8-pin		
10-pin		

In Figure 2, for B=0.1 and D=0.1, the variation of the NNR with the Rayleigh number and horizontally arranged pinned enclosures are depicted. As the Rayleigh number increases, the NNR increases. For Ra>2×10^5, the NNR also increases with increasing number of pins. One of the main reasons for this increase in heat transfer is, of course, the increase in the hot surface area due to the introduction of pins. But the increase in heat transfer is not exactly proportional to the surface area, air flow around the pins play an important role as well.

In Figure 3, for B=0.2 and 6-pin horizontally arrangement, the variation of the NNR with the Rayleigh number and the dimensionless pin diameter D are depicted. For D=0.1, the NNR slightly increases up to Ra=5×10^5 and then decreases. As the Rayleigh number increases, the NNR decreases due to a loss in heated surface area.

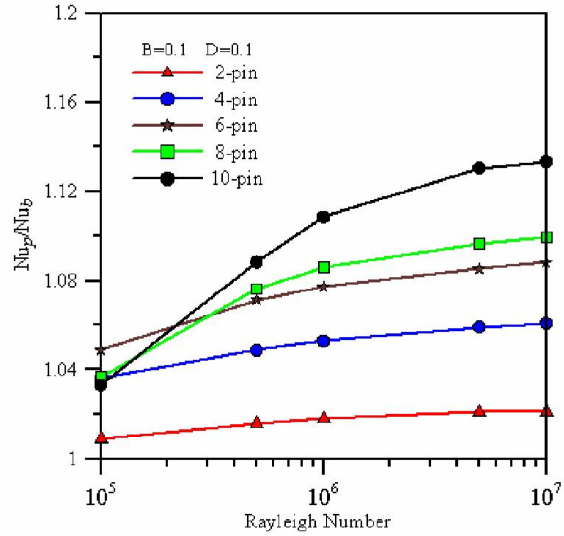


Figure 2 Variation of the NNR with Rayleigh number and horizontally-arranged pinned enclosure for B=0.1 and D=0.1

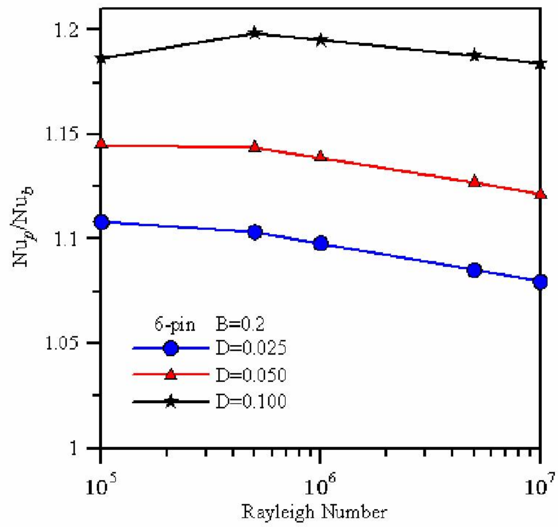


Figure 3 Variation of the NNR with the Rayleigh number and the pin diameter for 6-pin horizontal configuration

In Figure 4, for Ra=10^5, 10^6 and 10^7, the temperature field for 6-pin horizontal and vertical arrangements (pins with dimensions of D=0.025 and B=0.2) are depicted along *X*=0, *Y*=0.1 and *Y*=0.8 planes. Y=0.1 plane is the plane that slices the pins lengthwise from the middle which illustrates the temperature changes around the pins. Horizontal arrangement of the pins allows more heat dissipation in comparison to vertical arrangement. As the heat is dissipated from the bottom pin row, it heats the air around the pin which rises vertically. Air flows around the second-row pins above providing it hotter air; this reduces the temperature gradient which in turn yields less heat dissipation. The third-row pins will receive hotter air from the pins below and thus the heat losses from this row will be less than the second row.

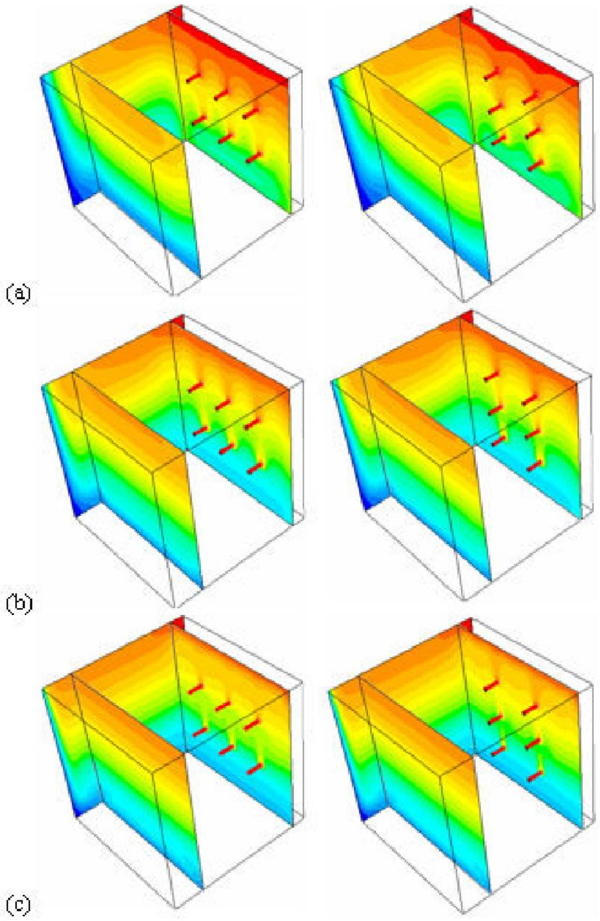


Figure 4 Temperature field for 6-pin arrangements for (a) Ra=10^5 and (b) Ra=10^6 and (c) Ra=10^7 of D=0.025 and B=0.2 case (along *X*=0, *Y*=0.1 and *Y*=0.8 planes)

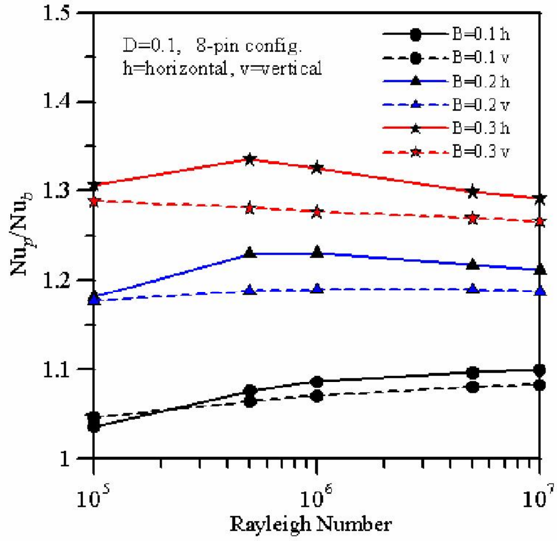


Figure 5 Variation of the mean Nusselt number ratio with Rayleigh number and the pin length for 8-pin horizontal and vertical configurations

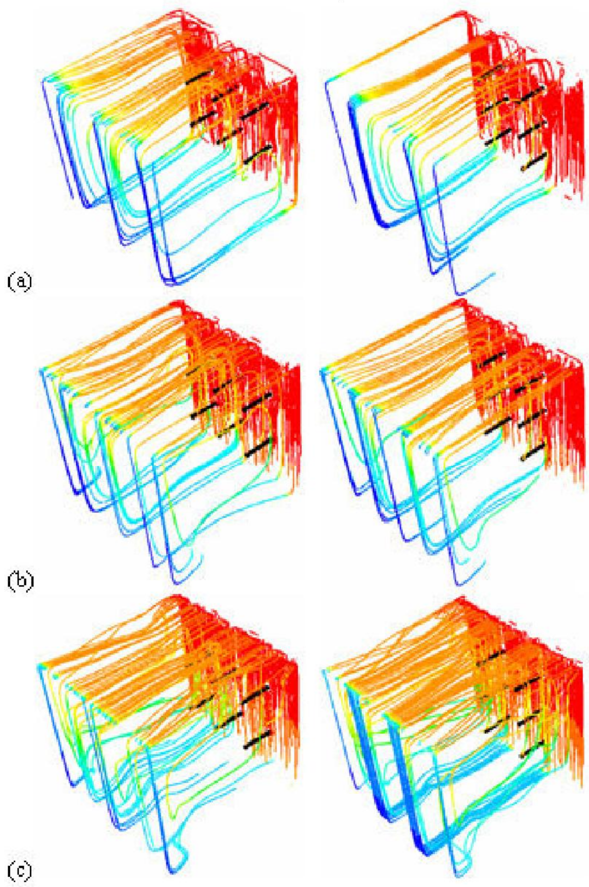


Figure 6 Path lines (colored by temperature) for 6-pin arrangements for (a) Ra=10^5 and (b) Ra=10^6 and (c) Ra=10^7 of D=0.025 and B=0.2 case

In Figure 5, for D=0.1 and 8-pin horizontal and vertical arrangements, the variation of the NNR with the Rayleigh number and the dimensionless pin length is given. The relative increase in heat dissipation is about 7-8% for short pins, and it increases to about 26-34% for longer pins. On the other hand, the heat transfer from the pins arranged horizontally are higher for Ra>2×10^5 irrespective of the pin diameter; that is, the heat transfer from vertically arranged pins is about 4-7% less than that of the horizontal arrangement. In view of Figure 4, this outcome is not surprising.

In Figure 6, for Ra=10^5, 10^6 and 10^7, the path lines for 6-pin horizontal and vertical arrangements (pins with dimensions of D=0.025 and B=0.2) are depicted. The air flow from the pins rises vertically preferably between the street of two pin columns or pin-side walls. The air for low Rayleigh numbers rotates between the cold and hot walls following a rectangular path bounded by the enclosures walls (Figure 6a). With increasing Rayleigh number, the strength of the flow increases and some disturbances in the flow patterns at the bottom wall are encountered (Figures 6b and 6c).

CONCLUSION

In this study, laminar natural convection heat transfer and air flow in cubic enclosure is studied numerically using FLUENT®. The enclosure is heated from one lateral wall while it is cooled from opposite lateral wall. Cylindrical pins of 2, 4, 6, 8 and 10 are arranged in in-line horizontal and vertical configurations are attached to the hot wall to enhance heat transfer from the enclosure. Rayleigh numbers considered in the study range from 10^5 to 10^7. The dimensionless pin diameters and the lengths were varied, and, for each case, the mean Nusselt numbers for the cold surface are computed. The study concludes the following: (a) the Nusselt number increases with increasing Rayleigh number for a fixed case, (b) the NNR increases with increasing number of pins, (c) the NNR increases with increasing pin length and decreases with increasing Rayleigh number; the relative increase in heat dissipation is about 7-8% for short pins and 26-34% for longer pins. (d) the heat transfer from vertically arranged pins is about 4-7% less than that of the horizontal arrangement

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