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Effect of Forced Convection on Temperature Distribution and Velocity Profile in a Rectangular Enclosure with Varying Fan Speed

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Abstract:
Forced convection in a three dimensional rectangular enclosure was considered with heaters placed on the opposite wall on the y-z planes, two windows on the adjacent opposite walls on x-z planes and one fan centrally fixed at the top (ceiling) x-y plane. The fan was set to rotate, the speed was varied. Results showed that temperature increases with increase in room depth. As the fan speed increases, temperature increases with increase in room depth at a lower rate. However the rate of increase in temperature is higher with increase in the room's depth. At any particular room depth, temperature is higher at lower Reynolds number and lower at high Reynolds number. Temperatures within the room are generally lower when the fan speed is increased. With respect to velocity profile, velocity of air within the room decreases with decrease in rooms' depth. The rate at which velocity decreases is higher at lower Reynolds number. As the fan speed decreases the rate at which velocity decreases lowers. Results also indicate that velocity is lower directly beneath the fan. The lowest velocity is registered when Reynolds number is high and highest at low Reynolds number.

Keywords: *Central difference scheme, finite difference method, momentum equation and energy equation*

1. Introduction

1.1. Background of Study

Heat transfer is the exchange of thermal energy between physical systems. The rate of heat transfer is dependent on the temperatures of the systems and the properties of the intervening medium through which the heat is transferred. Three fundamental modes of heat transfer are conduction, radiation and convection. Conduction is the transfer of heat through an intervening matter without bulk motion of the matter. Radiation is the transmission of heat energy through space without the necessary presence of matter.

Convection is the mechanism of heat transfer through a fluid in the presence of bulk fluid motion. It can be classified as natural (free) or forced convection depending on how the fluid motion is initiated. Irrespective of the classification, the fluid motion through convection enhances heat transfer with the higher the velocity the faster the heat transfer**.** The high velocity fluid results in a decreased thermal resistance across the boundary layer from the fluid to the heated surface. This in turn increases the amount of heat that is carried away by the fluid.

To obtain the best possible rate of heat transfer we require a fluid flow which is turbulent and this condition can only be guaranteed with a forced convection system. Forced convection is faster than natural free convection and therefore more heat is transferred. In the free convection, there is no mixing of the fluid and heat must be transferred within this layer by conduction. Forced convection on the other hand creates turbulent air motion, this causes fluid particles to have a circular, eddying motion. The eddies sweep the edge of the laminar layer and probably penetrate it taking with them into the turbulent core, fluid which is at a higher temperature than the fluid in the core. The hotter fluid, therefore, is rapidly mixed with the cooler fluid in the turbulent core and heat is quickly transferred from the laminar layer to the turbulent core/region. Turbulent region can be considered of three regions: laminar sublayer (where viscous effects are dominant), buffer layer (where both laminar and turbulent effects exist), and turbulent layer. The intense mixing of the fluid in turbulent flow enhances heat and momentum transfer between fluid particles, which in turn increases the friction force and the convection heat transfer coefficient. In heat exchanger design it is desirable to design for turbulent flow for this reason. Investigating velocity profile and temperature distribution in a room with varying fan speed enables one to determine the velocity pressure. The velocity pressure depends upon the velocity of air and always acts in the direction of the airflow. The higher the air flow rate within a room the faster the cooling process. Also, temperature distribution determines the pressure gradient which dictates the rate of air flow within the enclosure. Varying fan speed alters pressure difference within the room due to the differential cooling and by extension modifies the rate of heat transfer.

1.2 Statement of the Problem

Forced convection in a room is a mechanism in which air motion is generated by an external source like pump or fan. Due to its unusual high speed, heat is transferred quickly and efficiently. Past study have been conducted on forced convectional flow in a rectangular enclosure in form of three dimensional rectangular enclosure with heaters placed on two opposite vertical wall, windows on the other two walls and a fan at the top (ceiling). In this study the effects of forced convection on temperature distribution and velocity profile with varying fan speed is the main focus.

1.3. Justification of the Study

In many heat transfer applications forced convection of air is used to effect heat transfer and facilitate cooling process. This study makes an in-depth analysis based on varied fan speed in a forced convection. The results will help improve convection heat transfer more so in electronic gadgets, improve heat transfer in other relevant areas such as the cooling systems of refrigerators, radiators and also help to vary design in civil engineering with particular reference to the thermal effects in building and other structures.

1.4. Specific Objectives

The specific objectives of this study are to;

i. To determine the temperature distribution in a room caused by forced convection with varying fan speed.
ii. To determine velocity profile in the room caused by forced convection with the varied fan speed.
iii. To determine the effect of varying Reynolds on temperature and velocity with varying fan speed.

1.5. Significance of the Study

Designs for heating and cooling have a wide range of application in engineering field including air fans for electronics devices. This numerical study seeks to simulate forced convection in a rectangular enclosure with two heaters placed on the opposite vertical wall, windows on other two opposite walls and cooling fan centrally placed at the top. The focus shall be placed on the effect of convection on temperature distribution and velocity profile with varying fan speed. This is related to cooling air fans which has wide application in electronic equipment.

1.6. Assumptions of the Study

a. Fluid density only varies on the vertical component
b. Viscosity dissipation effects are negligible.
c. Body forces due to gravity are negligible.
d. There is no pressure variation on the horizontal direction.

1.7. Geometry of the Problem

The problem under consideration is a three dimensional rectangular enclosure with two heaters placed on the opposite walls, two windows on the other adjacent opposite walls and one fan centrally fixed at the top (ceiling).

2. Literature Review

Owing to its wide range of application, Heat transfer through convection in an enclosed surface has been studied adversely. Eckert and Carson (1961) studied natural convection in an air layer enclosed between two vertical plates with different temperatures, the result showed that heat transfer occurs by conduction only in major portion of air space at low Rayleigh numbers. Convection currents deformed the temperature field at increasing Rayleigh numbers creating boundary layers along the vertical walls within which the mean temperature drop was concentrated. Bhattacharyya (1965) conducted a study on the effect of forced convection heat transfer in vertical channels. Curves were presented to determine whether free convection or forced convection mode of transfer is predominant for a particular Reynolds number and Rayleigh number. At Reynold $>10^5$ free convection effects are negligible. Also downward flow through heated channel at low Reynolds number is unstable. Under similar condition the overall heat transfer coefficient for downward flow trend tend to be higher than that for upward flow.

Ali and Hussein (1993) studied the effect of corrugation frequencies on natural convective heat transfer and flow characteristics in a square enclosure of veer-corrugated vertical walls. The result showed that the overall heat transfers through the enclosure increased with increase of corrugation for low Gash of number, but the effect was reversed for high Grash of number. Ganzaroli and Millanez (1995) investigated the natural convection in rectangular enclosure heated from below and symmetrically cooled from the sides. The result showed that for the square cavity, the flow and thermal field are not strongly affected by the isothermal or constant heat flux boundary condition at the bottom heat source. Aydin and Young (2000) investigated numerically the natural convection of air in vertical square cavity with localized isothermal heating from below and symmetrical cooling from the side walls. The top wall as well as the non-heated parts of the bottom was considered adiabatic. The length of the symmetrically placed isothermal heat source at the bottom was varied. The result showed that two counter rotating vertices were formed in the flow domain due to natural convection. Manca (2003) conducted a study on the effects of heated wall position on mixed convection in a channel with an open cavity and the result showed that forH/D=1.0 and Reynolds numbers of100 and 1000 recirculation cells developed within the cavity which improved the heat removal from heat source for opposing case and opposing forced flow configuration had the highest average Nusselt number among other configuration for various H/D. Sigey *et al* (2004) did a study of free convection turbulent heat transfer in an enclosed cavity or box. The enclosure contained a convectional heater built in one wall and having a window in the same wall. The results were that the enclosure is stratified into three regions: a cold upper region, a hot region in the area between the heater and the window and a warm lower region. Summon S. (2006) conducted a study on the combined free and forced convection inside a two-dimensional multiple ventilated rectangular enclosure. The study revealed that heat transfer coefficient is strongly affected by Reynolds number and Richardson number. Hussain S.H (2010) studied combined convection flow through inclined rectangular enclosure with a sliding wavy hot top surface the results were presented as stream lines and temperature contours graphs and the variations in local and average Nusselt number variation at the top and bottom were explained. The result indicated that streamlines and isothermal contours are affected by the wavy shape of the top surface. Studies were also undertaken by Sigey *et al.* (2011), who studied buoyancy driven free convection turbulent heat transfer in an enclosure. They investigated a three dimensional enclosure containing a convectional heater built into one wall having a window in same wall. The heater is located below the window and the other remaining wall insulated. The results were that three regions a cold upper region, a hot region in the area between and a warm lower region. Nogueira *et al*. (2011) studied natural convection in rectangular cavities with different aspect ratio. In their setup, natural convection was numerically analyzed in rectangular cavity heated on one side and cooled on the opposite side. Temperatures of the heated wall and of the cooled wall were assumed to be constant. Result showed that Rayleigh number drastically influenced the flow profile and heat transfer inside the cavity as well as thickness of the thermal boundary layer. Also, Nusselt number is strongly dependant on the L/D and that this dimensionless variable increases with increase in W/L. Hamid and Mohammed (2011) carried out an investigation of turbulent mixed convection in air filled enclosures. The result showed that when Reynold's number increases the circulation of flow vortices increases and becomes stronger making the forced convection effective more dominant for different values of Richardson numbers, also in large Richardson numbers the natural convection is a major parameter of heat transfer in a cavity. Salleh (2011) studied numerical solution of forced convection boundary layer flow on a horizontal circular cylinder with Newtonian heating and the result showed that an increase in the value of Prandtl number lead to a decrease in temperature profiles. Hassan *et al* (2011) investigated the effects of corrugation frequency and aspect ratio on natural convection within the enclosure having sinusoidal corrugation over heated top surface. A constant flux heat source was flush mounted on the top sinusoidal wall; modeling a wavy sheet shaded room exposed to sunlight. The flat bottom surface is considered as adiabatic, while the vertical walls were maintained at constant ambient temperature. The fluid considered inside the enclosure was air having Prandtl number 0.71. Results revealed that the convective phenomenon is greatly influenced by the presence of corrugation and variation of aspect ratio. Ghadhimi *et al.* (2012) studied analysis of free and forced convection in air flow windows using numerical simulation of heat transfer. The results showed air flow influence increases in air flow windows (in both forced and natural convection).Also it shows that air flow is proportional to inlet temperature and flow rate, but the effect of temperature is higher than effect of flow rate. Studies undertaken by Mairura *et al,*(2013) on the natural convection with localized heating and cooling on opposite vertical walls in an enclosure indicates that if the Reynolds number of a system is small, the viscous force is predominant and the effect of viscosity are important. On the other hand if the Reynolds number is large, the inertial force is predominant and the viscous effects are only important in narrow layer near the solid boundary. In general, the convective currents caused by buoyancy forces play a major role in determining the velocity profiles in a room. Gareh (2014) conducted numerical study of forced convection in a rectangular channel and the result, showed that the velocity profiles and calculated temperature has the side effect on the input speed limits for two developing layers extended over a more or less large length according to the value of the Reynolds number. Hsu-Cheng *et al* (2007) did a study on The Measurement of Flow Characteristics of a Ceiling Fan with Varying Rotational Speed. In their experimental set up, they used a ceiling fan equipped with the fan blades in spindle shape. A series of measuring points were taken horizontally, 100cm below the fan. Results revealed that the air distribution of the fan exhibits a unique pattern; that the higher speed occurs somewhere below the center of the blade and the velocity declines as the measuring points are gradually away from the center. Rahman *et al* (2007) conducted a Numerical Study of Opposing Mixed Convection on a Vented Enclosure using finite element method. Ventilations were placed in such a way that external fluid entered the enclosure through an opening in the left vertical wall and exit from another fixed opening in the right vertical wall. Various inlet port configuration was studied with the changing of the governing parameters. Results showed that with the increase in Reynolds and Richardson numbers the convective heat transfer becomes predominant over conduction heat transfer and the rate of heat transfer from the heated wall significantly depends on the position of the inlet port. Rehena *et al* (2013) investigated forced convection heat transfer phenomenon in a two-dimensional horizontal channel having an open cavity with porous medium. A non uniform heat flux was considered to be located on the bottom surface cavity. The three different heating modes at the bottom, the rest of the surfaces were considered to be perfectly adiabatic. The physical domain was filled with water based nano fluid. Fluid entered from left and exits from right. Results showed that increasing Prandtl causes the enhancement of heat transfer rate. Numerical study undertaken by Robins *et al*, (2014) of Mixed Convection Flow Inside Ventilated Enclosure with bottom wall uniformly heated, two vertical walls maintained at constant cold temperature and top wall insulated indicated that the strength of circulation increases with the increase in value of Richardson number irrespective of the

Reynolds number and Prandtl number and as the value of Richardson number increases, there occurs a transition from conduction to convection dominated flow at Richardson number 1.Momanyi *et al*, (2015) recently studied the Effect of Forced Convection on Temperature Distribution and Velocity Profile in a Room and the results showed that temperature decreases with increase in room height also the velocity decreases as fluid particles flow up the room. Momanyi *et al*, (2015) in their model used a fan whose speed was constant. In this study, we consider the effect of forced convection on temperature distribution and velocity profile with the varying fan speed. The subsequent chapter will show the governing equations which in this case encompass continuity equation, momentum equation and energy equation. These equations are non-dimensionalised making it possible to introduce parameters such as Pr and Re numbers which are instrumental to investigate the fore listed objectives. Discretized governing equations together with the appropriate boundary conditions are used to generate tri-diagonalised matrices which are solved to generate results.

3. Methodology

This section highlights the governing equations that govern the flow of fluids with particular reference to air. The three equations under consideration that govern Newtonian fluid experiencing heat and mass transfer include; continuity equation, momentum equation and energy equation. Considering the nature of investigation at stake, the equations are presented in two-dimension.

3.1. Conservation Equations

We considered the equations governing behavior of Newtonian fluids experiencing heat and mass transfer. These fundamental equations of fluid dynamics are based on the following universal laws of conservation; conservation of mass (continuity), momentum and energy.

3.2. Continuity Equation

The law of conservation of mass states that the rate of increase of mass within the controlled volume is equal to the net rate of influx through the controlled surface. According to (Currie 1984) the continuity equation can be written as;

$$\frac{\partial \rho}{\partial t}+\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \qquad (1)$$

For steady state

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \qquad (2)$$

Where x and y are the distances measured along the horizontal and vertical directions respectively, u and v are the velocity components in x and y direction respectively.

3.3. Momentum Equation

The equation is derived from Newton's second law of motion, which states that the sum of the body and surface forces acting on a system is equal to the rate of change of linear momentum of the system. Here under forced convection, the following momentum equation holds;

$$\rho u\frac{\partial u}{\partial x}+\rho v\frac{\partial u}{\partial y}=-\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \qquad (3)$$

$$\rho u\frac{\partial v}{\partial x}+\rho v\frac{\partial v}{\partial y}=-\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+g\beta\left(T-T_0\right) \qquad (4)$$

Where p denotes the thermodynamic pressure; μ, g, β and T are the kinematics viscosity, gravitational acceleration, thermal expansion coefficient and temperature respectively.

3.4. Energy Equation

This is derived from the first law of thermodynamics which states that the rate of energy increase in as a system is equated to the heat added to the system and the work done on the system. From Currie (1974) assuming no external heat source, the energy equation is often written as

$$\rho c_p u\frac{\partial T}{\partial x}+\rho c_p v\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right)+\mu\Phi \qquad (5)$$

where $\Phi=2\left(\frac{\partial u}{\partial x}\right)^2+2\left(\frac{\partial v}{\partial y}\right)^2+\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)^2$ and α is the thermal diffusivity.

3.5. Method of Solution

The continuity, momentum and energy equations are non dimensionalised so as to reduce complex physical problems to the simplest form prior to obtaining quantitative solutions. Dimensional analysis also makes the equations more succinct from the physical relationship: equations that hitherto look formidable can be solved with little effort. This process reduces the number of independent variables that specify the problem.

Using the following dimensionless variables:

$X = \frac{x}{L}, Y = \frac{y}{L}, U = \frac{u}{u_0}, V = \frac{v}{v_0}, P = \frac{p}{pu_0^2}, \theta = \frac{T - T_0}{T_1 - T_0}$

the governing equations are reduced to non-dimensional form as follows. Consider the continuity equation (2)

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

Substituting the non-dimensional variables results in the following form of continuity equation

$$\frac{u_0}{L}\frac{\partial U}{\partial X} + \frac{u_0}{L}\frac{\partial V}{\partial Y} = 0 \quad \text{……} (6)$$

Equation (6) can be written in non-dimensional form as:

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \quad (7)$$

Consider the x-component of Navier-Stokes equation, (3) and substituting the non-dimensional variables we get that|;

$$\frac{\rho u_0^2}{L} U \frac{\partial U}{\partial X} + \frac{\rho u_0^2}{L} V \frac{\partial U}{\partial Y} = -\frac{\rho u_0^2}{L}\frac{\partial P}{\partial X} + \frac{\mu u_0}{L^2}\left(\frac{\partial^2 U}{\partial^2 X} + \frac{\partial^2 U}{\partial^2 Y}\right) \quad \text{……} (8)$$

Multiplying equation (8) by $L / \rho u_0^2$ gives

$$U\frac{\partial U}{\partial X} + V\frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \frac{\mu}{\rho u_0 L}\left(\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2}\right) \quad \text{……} (9)$$

But $\mathrm{Re} = \rho u_0 L / \mu$ therefore, the non-dimensional form of the x- component of the Navier-Stokes equation becomes;

$$U\frac{\partial U}{\partial X} + V\frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \frac{1}{\mathrm{Re}}\left(\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2}\right) \quad \text{……} (10)$$

Similarly, the non-dimensional form of the y- component of the Navier-Stokes equation is;

$$U\frac{\partial V}{\partial X} + V\frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \frac{1}{\mathrm{Re}}\left(\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2}\right) \quad \text{……} (11)$$

From equation (5) the energy equation can be written as,

$$\rho c_p u \frac{\partial T}{\partial x} + \rho c_p v \frac{\partial T}{\partial y} = \alpha\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) + \mu\Phi \text{ where } \Phi = 2\left(\frac{\partial u}{\partial x}\right)^2 + 2\left(\frac{\partial v}{\partial y}\right)^2 + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)^2$$

Substituting the non dimensionless variables into equation (5) gives

$$\frac{\rho c_p u_{0(T_1 - T_0)}}{L}\left(U\frac{\partial \theta}{\partial X} + V\frac{\partial \theta}{\partial Y}\right) = \frac{\alpha(T_1 - T_0)}{L^2}\left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2}\right) + \frac{\mu u_0^2}{L^2}\Theta \quad \text{……} (12)$$

Where $\Theta = 2\left(\frac{\partial U}{\partial X}\right)^2 + 2\left(\frac{\partial V}{\partial Y}\right)^2 + \left(\frac{\partial U}{\partial Y} + \frac{\partial V}{\partial X}\right)^2$

Simplifying equation (12):

$$U\frac{\partial \theta}{\partial X} + V\frac{\partial \theta}{\partial Y} = \frac{\alpha}{\rho c_p u_0 L}\left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2}\right) + \frac{\mu u_0}{\rho c_p L(T_1 - T_0)}\Theta \quad \text{……} (13)$$

For low velocities, viscous dissipation is negligible.

But $\frac{\alpha}{\rho c_p u_0 L} = \frac{\alpha}{\rho c_p u_0 L}\left(\frac{\mu}{\mu}\right) = \left(\frac{\alpha}{\mu c_p}\right)\left(\frac{\mu}{\rho u_0 L}\right) = \frac{1}{\mathrm{Pr}\,\mathrm{Re}}$ Equation (13) becomes:

$$U\frac{\partial \theta}{\partial X} + V\frac{\partial \theta}{\partial Y} = \frac{1}{\mathrm{Pr}\,\mathrm{Re}}\left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2}\right) \quad \text{……} (14)$$

Re. number is used for scaling of similar but different sized flow situation. The relative thickness of velocity and thermal boundary layers is described by Prandtl number. For low Prandtl number heat diffuses much faster than momentum flow and the velocity boundary layer is fully contained within the thermal boundary layer. On the other hand, for high Prandtl number, like in air heat diffuses much slower than the momentum and thermal boundary layer is contained within the velocity boundary layer.

3.6 Computational Procedure

In this study the Central Difference scheme is used to solve the momentum and energy equations. The method obtained a finite system of linear algebraic equations from discretized momentum and energy equations. Using the algebraic equations from discretized momentum and energy equations tri-diagonal matrix were obtained and solved using MATLAB to generate results.

3.7. Discretization of the Governing Equations

In these section numerical results of the horizontal velocity, vertical velocity and temperature in an enclosure are displayed for various Reynolds number Re (100, 200, 400). The air in the room is considered to have fixed Prandtl number 0.71. Fans speed is varied, with the effect reflected in pressure variation and temperatures in the room.

Considering momentum and energy equations,

3.8. Horizontal Fluid Velocity

We investigate both the horizontal and vertical velocities of the fluid in the room. For the Central Difference scheme (CDS), the values u_x, u_y, u_{xx} and u_{yy} are replaced by central difference approximation. When these values are substituted into Equation (7), we get

$$\left[U\frac{U_{i+1,j}-U_{i-1,j}}{2\Delta X}+V\frac{U_{i,j+1}-U_{i,j-1}}{2\Delta Y}\right]=-\left[\frac{P_{i+1,j}-P_{i-1,j}}{2\Delta X}\right]+\frac{1}{\mathrm{Re}}\left[\frac{U_{i+1,j}-2U_{i,j}+U_{i-1,j}}{(\Delta X)^2}+\frac{U_{i,j+1}-2U_{i,j}+U_{i,j-1}}{(\Delta Y)^2}\right] \quad \text{……………………… (15)}$$

We investigate the effect of Re on the fluid horizontal velocity. Taking $\Delta X = 0.25$ and $\Delta Y = 0.01$, for convergence $\frac{\Delta Y}{(\Delta X)^2}$ should be less than 0.5, Re = 100and V = 0,U = 1, and multiplying equation (15) by ΔY while letting $r=\frac{\Delta Y}{(\Delta X)^2}$ we get the scheme.

$$(2-r)U_{i+1,j}+(2r-200)U_{i,j}-rU_{i-1,j}=-2P_{i+1,j}+2P_{i-1,j}+100U_{i,j+1}+100U_{i,j-1} \quad \text{………………………………………………. (16)}$$

Taking and i = 1, 2, 3…….6 and j = 1 and r = 0.16, U(x, 0) = 1and U(x, 2) = 0 ,P(x, 0) = 10^5 (atmospheric pressure of air), we get the matrix-vector equation

$$\begin{bmatrix} -199.68 & 1.84 & 0 & 0 & 0 & 0 \\ -0.16 & -199.68 & 1.84 & 0 & 0 & 0 \\ 0 & -0.16 & -199.68 & 1.84 & 0 & 0 \\ 0 & 0 & -0.16 & -199.68 & 1.84 & 0 \\ 0 & 0 & 0 & -0.16 & -199.68 & 1.84 \\ 0 & 0 & 0 & 0 & -0.16 & -199.68 \end{bmatrix}\begin{bmatrix} U_{1,1} \\ U_{2,1} \\ U_{3,1} \\ U_{4,1} \\ U_{5,1} \\ U_{6,1} \end{bmatrix}=\begin{bmatrix} 100 \\ 100 \\ 100 \\ 100 \\ 100 \\ 100 \end{bmatrix} \quad \text{…………. (17)}$$

If Re is varied to 200 and 400 and solving the above matrix equation (17), we get the solution as shown in Figure 4

According to Bernoulli's principle, when the fan speed is increased the pressure decreases. We therefore expect pressure to drop of air near the fan. The following results for velocity in the room at varying Re are also obtained.

3.9. Vertical Velocity

Discretizing the vertical velocity equation (2) becomes;

$$\left[U\frac{U_{i+1,j}-U_{i,j-1}}{2\Delta X}+V\frac{V_{i,j+1}-V_{i,j-1}}{2\Delta Y}\right]=-\left[\frac{P_{i+1,j}-P_{i-1,j}}{2\Delta X}\right]+\frac{1}{\mathrm{Re}}\left[\frac{V_{i+1,j}-2V_{i,j}+V_{i-1,j}}{(\Delta X)^2}+\frac{V_{i,j+1}-2V_{i,j}+V_{i,j-1}}{(\Delta Y)^2}\right] \quad \text{…………………………… (18)}$$

We investigate the effect of Re on the fluid vertical velocity. Taking $\Delta X = 0.25$ and $\Delta Y = 0.01$,Re= 100andV=1, U = 0, and multiplying equation (9) by ΔY with $r=\frac{\Delta Y}{(\Delta X)^2}$ we get the scheme;

$$rV_{i+1,j}+(2r+2)V_{i,j}-rV_{i-1,j}=-2P_{i+1,j}+2P_{i-1,j}-51V_{i,j+1}+51V_{i,j-1} \quad \text{……………………………………………………. (19)}$$

Taking and i = 1,2,3…….6 and j = 1 and r = 0.16, and taking boundary conditions V(x, 2) =1, V(x, 0) = 0 we get the matrix-vector equation as;

$$\begin{bmatrix} 2.32 & 0.16 & 0 & 0 & 0 & 0 \\ -0.16 & 2.32 & 0.16 & 0 & 0 & 0 \\ 0 & -0.16 & 2.32 & 0.16 & 0 & 0 \\ 0 & 0 & -0.16 & 2.32 & 0.16 & 0 \\ 0 & 0 & 0 & -0.16 & 2.32 & 0.16 \\ 0 & 0 & 0 & 0 & -0.16 & 2.32 \end{bmatrix}\begin{bmatrix} V_{1,1} \\ V_{2,1} \\ V_{3,1} \\ V_{4,1} \\ V_{5,1} \\ V_{6,1} \end{bmatrix}=\begin{bmatrix} -51 \\ -51 \\ -51 \\ -51 \\ -51 \\ -51 \end{bmatrix} \quad \text{………………………………………..(20)}$$

If Re is varied to 200 and 400 and solving the above matrix equation (20), and the solutions obtained. When the Fan speed increases pressure in the room drops in accordance with Bernoulli's principle therefore for this case pressure near the fan is assumed to drop, considering this alteration and using matrix (20) we get the results for varying Re are obtained.

3.10. Fluid Temperature

Discretizing the temperature equation (3) becomes

$$U\frac{\theta_{i+1,j}-\theta_{i-1,j}}{2(\Delta X)}+V\frac{\theta_{i,j+1}-\theta_{i,j-1}}{2(\Delta Y)}=\frac{1}{\Pr\times\mathrm{Re}}\left[\frac{\theta_{i+1,j}-2\theta_{i,j}+\theta_{i-1,j}}{(\Delta X)^2}+\frac{\theta_{i,j+1}-2\theta_{i,j}+\theta_{i,j-1}}{(\Delta Y)^2}\right]$$

…………………………………………… (21)

We investigate the effect of Re, at fixed Pr=0.71on the fluid velocity. Taking $\Delta X = 0.25$ and $\Delta Y = 0.01$,Re=100 and V=1,U=0, and multiplying equation (21) by ΔY with $r=\frac{\Delta Y}{(\Delta X)^2}$ we get the scheme;

$$r\theta_{i+1,j}+(2r+20000)\theta_{i,j}-r\theta_{i-1,j}=10035.5\theta_{i,j-1}-35.5\theta_{i,j+1}$$

…………………………………………………. (22)

Taking and i = 1, 2, 3…….6 and j = 1 and r = 0.16, taking the boundary conditionsθ(x,0)=20and θ(x,2) = e^x ,we get the matrix-vector equation.

$$\begin{bmatrix} -19999.68 & 0.16 & 0 & 0 & 0 & 0 \\ -0.16 & -19999.68 & 0.16 & 0 & 0 & 0 \\ 0 & -0.16 & -19999.68 & 0.16 & 0 & 0 \\ 0 & 0 & -0.16 & -19999.68 & 0.16 & 0 \\ 0 & 0 & 0 & -0.16 & -19999.68 & 0.16 \\ 0 & 0 & 0 & 0 & -0.16 & -19999.68 \end{bmatrix}\begin{bmatrix}\theta_{1,1}\\ \theta_{2,1}\\ \theta_{3,1}\\ \theta_{4,1}\\ \theta_{5,1}\\ \theta_{6,1}\end{bmatrix}=\begin{bmatrix}200613.501\\ 200447.6885\\ 199996.9634\\ 198771.7657\\ 195441.3329\\ 186388.2778\end{bmatrix}$$

………………………………………...(23)

If Re is varied to 200 and 400 and solving the above matrix equation (23), we get the solutions as shown in figure1.

When the fan speed is reduced temperature of air near the fan increases due to reduced cooling effect derived from the fan. We therefore assume boundary conditions $\theta(x,0)=25,\theta(x,2)=e^x$. Using the matrix in equation (23) we obtain the following results for temperature distribution.

4. Results and Discussion

Equation (3) was discretized as indicated in equation (14) and (15). Using varied Re number and Pr number of 0.71 together with respective boundary conditions, a set of linear algebraic equations were formed. Tri-diagonal matrix was formed from algebraic equations and solved using software. The boundary conditions were altered to cater for a reduction in fan speed. The matrix was again solved using a new set of boundary conditions and results shown in figure 2

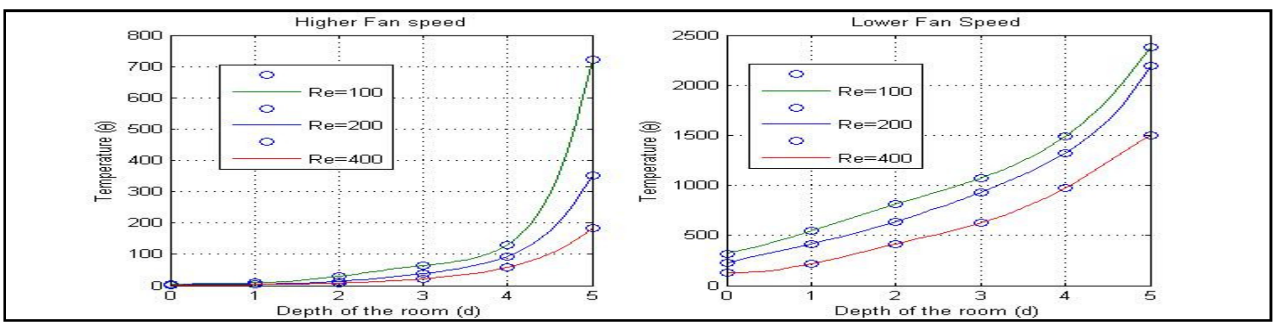

Figure 2: Temperature against Depth of room at varying the Reynolds number

From Figure 2. Temperature increases with increase in room depth due to the cooling at top influenced by the fan. In both cases fluid flow started at low temperature when Re number was high than when it was low, this can be attributed to the fact that at high Re number inertial forces were predominant. Also as the fan speed increases temperature increases with increase in room depth at a lower rate but the rate increases with the increase in the room depth. Figure 2 shows that as the fan speed increases the temperature difference with change in Re number near the fan i.e. (lower room depth) lowers. This is show in Figure 2 where the linear graphs almost coincide. Generally temperature distribution changes with change in fan speed.

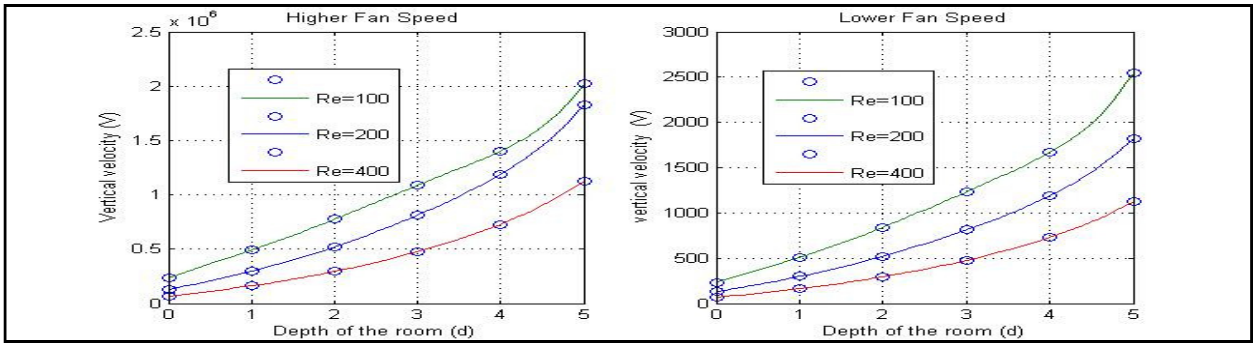

Figure 3: Vertical Velocity against Depth of room at varying the Reynolds number with varying fan speed

At the bottom of the room, velocity of air is higher and it decreases with decrease in rooms' depth. This phenomenon is attributed to the heating effect derived from the heat source at the bottom of the room that heightens the kinetic energy of air upon heating it. The fan at the ceiling on the other hand cools down the fluid particles making them denser thus lowering its velocity.

When fan speed is lowered, pressure within the room increases. Comparing graphs in Figure 3, its evident that the rate at which velocity decreases with decrease in rooms depth is higher at lower fan speed and lower at higher fan speed. From the graphs, initial velocities at higher Re number are higher when the fan speed is higher.

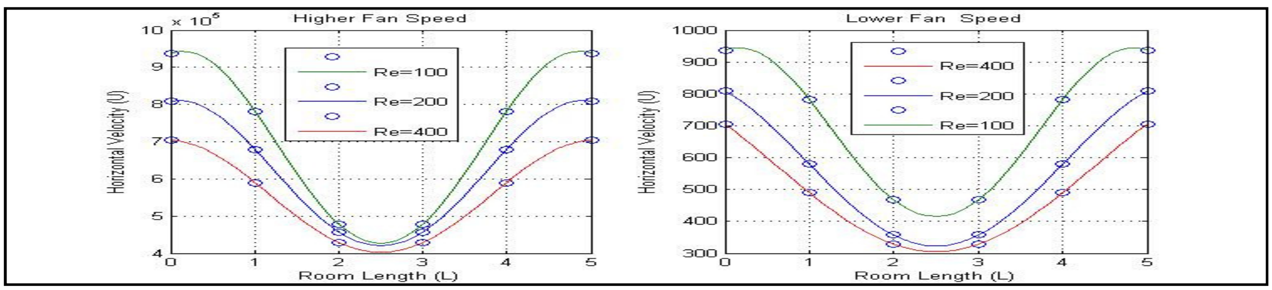

Figure 4: Horizontal Velocity against Length of room at varying the Reynolds number

Effects from heaters in the room together with the fan sets into motion convectional currents of air which is believed to form dual system of the current on the x-z plane of the enclosure. The warmed air from the heaters are forced up making it to rise at a higher velocity along the x-y walls this helps to register high velocity next to the walls on the extreme ends. Velocities at the mid length of the room are lower due to the cooling effects derived from the fan.

When the fan speed is lowered fluid temperature within the room increases and by extension the velocity decreases.

5. Conclusion and Recommendation

From Figure 2. Temperature increases with increase in room depth due to the cooling at top influenced by the fan. In both cases fluid flow started at low temperature when Re number was high than when it was low, this can be attributed to the fact that at high Re number inertial forces were predominant. Also as the fan speed increases temperature increases with increase in room depth at a lower rate but the rate increases with the increase in the room depth. Figure 2 shows that as the fan speed increase the temperature difference with change in Re number near the fan i.e. (lower room depth) lowers. This is show in Figure 2 where the linear graphs almost coincide. Generally, temperature distribution changes with change in fan speed. At the bottom of the room, velocity of air is higher and it decreases with decrease in rooms' depth. This phenomenon is attributed to the heating effect derived from the heat source at the bottom of the room that heightens the kinetic energy of air upon heating it. The fan at the ceiling on the other hand cools down the fluid particles making them denser thus lowering its velocity. When fan speed is lowered, pressure within the room increases. Comparing graphs in Figure 3, its evident that the rate at which velocity decreases with decrease in rooms depth is higher at lower fan speed and lower at higher fan speed. Initial velocities at higher Re number are lower at a lower room depth than at a lower Re number. Effects from heaters in the room together with the fan sets into motion convectional currents of air which is believed to form dual system of the current on the x-z plane of the enclosure. The warmed air from the heaters are forced up making it to rise at a higher velocity along the x-y walls this helps to register high velocity next to the walls on the extreme ends. Velocities at the mid length of the room are lower due to the cooling effects derived from the fan. When the fan speed is lowered fluid temperature within the room increases and by extension the velocity decreases. In general velocity profile and temperature distribution changes with change in fan speed.

6. Recommendations

i)Investigate forced convection in non-rectangular enclosures.
ii)Investigate any environmental impact on forced convection.
iii)Investigate forced convection if the fan is placed at vertical walls of an enclosure

7. Abbreviations

- Partial Differential Equation (PDE),
- Reynolds (Re)
- Prandtl (Pr).

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