Effects of Forced Convection on Temperature Distribution and Velocity Profiles in a Room

Joseph N. Momanyi*, Johana K. Sigey**, Jeconiah A. Okelo*** & James M. Okwoyo***

*Department of Pure and Applied Mathematics, Jomo Kenyatta University of Agriculture and Technology, Nairobi, KENYA. E-Mail: josemomanyi2015{at}gmail{dot}com

**Department of Pure and Applied Mathematics, Jomo Kenyatta University of Agriculture and Technology, Nairobi, KENYA. E-Mail: jksigey{at}jkuat{dot}ac{dot}ke

***Department of Pure and Applied Mathematics, Jomo Kenyatta University of Agriculture and Technology, Nairobi, KENYA. E-Mail: jokelo{at}jkuat{dot}ac{dot}ke

****School of Mathematics, University of Nairobi, Nairobi, KENYA. E-Mail: jmkwoyo{at}uonbi{dot}ac{dot}ke

Abstract—The flow of heat is one form of Newtonian motion. In this project forced convection was investigated in a three dimension rectangular enclosure with heaters placed on opposite walls, two windows on the adjacent opposite walls, and one fan centrally fixed at the top (ceiling).The fan was set to rotate at constant speed. To analyze the flow and heat transfer rates, a complete set of non -dimensional zed equations governing Newtonian fluid and boundary conditions were presented in vector form to eliminate the need for solving the continuity equations. A boussineque fluid motion in a three dimensional cavity is considered. The governing equations with boundary conditions were described using three point central difference approximations for a non-uniform mesh. The resulting finite difference equations are then solved using MATLAB simulation software. The results were presented on graphs to show velocity profiles and temperature distribution in a room.

Keywords—Forced Convection; Heat Transfer; Turbulent Flow in a Room.

Abbreviations—Generalized Gradient Diffusion Hypothesis (GGDH); Higher Order Terms (HOT); Simple Eddy Diffusivity (SED); Wealth Equation Equals Earnings x Time (WET).

I. INTRODUCTION

THE discipline of heat transfer is concerned with only two things; temperature and the flow of heat. Temperature represents the amount of thermal energy available whereas heat flow represents the movement of thermal energy from place to place. Convection is one of the mechanisms by which heat (energy) is transferred. Convection is concerted, collective movement of groups or aggregates of molecules within fluids. Fluids are a subset of the phases of matter and include liquids gases and plasmas. Convection can as well be defined as heat transfer in a gas or liquid by circulation of currents from one region to another. The transfer of heat occurs between the surface and the moving fluid when at different temperatures. It is sustained by both molecular motion and bulk motion of fluid within boundary layer. Boundary layer is a thin layer of a flowing gas or a liquid in contact with a surface. Convection of heat depends on viscosity, thermal conductivity, specific heat and density of the fluid. Majorly viscosity influences the velocity profile of the fluid flow. Fluids that flow readily, such as water or gasoline, have smaller viscosities than 'thick' liquids such as honey or motor oil. Viscosities of all fluids are strongly temperature dependant, increasing for gases and decreasing for liquids as the temperature increases. Convection can either be free convection or forced convection. Free convection (natural convection) is fluid flow due to density variations. Actually it is the fluid flow originated by gravity forces acting on a non-uniform –density fluids the density charges may be due to thermal gradients. Many different natural convection configurations are of interest from simplest hot/cold vertical plate in a fluid medium to external convection around hot/cold bodies, or internal convection within hot/cold enclosures (non-isothermal).

In natural convection, any fluid motion is caused by natural means such as the buoyancy effect but in forced convection, the fluid is forced to flow over a surface or in a tube by external means such as a pump or fan. The heat transfer is complicated since it involves motion as well as heat conduction. The fluid motion enhances heat transfer (the higher the velocity the higher the heat transfer rate).

The convective heat transfer coefficient strongly depends on the fluid properties and roughness of the solid surface and the type of fluid flow (laminar or turbulent). It is assumed that the velocity of the fluid is zero at the wall; this assumption is called non-slip condition. As a result, the heat transfer from the solid surface to fluid layer adjacent to the surface is by pure conduction, since the fluid is motionless.

The convection heat transfer coefficient in general varies along the flow direction. The mean average convection heat transfer coefficient for a surface is determined by (properly) averaging the local heat transfer coefficient over entire surface. Fluid flow from laminar to turbulent occurs over some region which is called transition region. The profile in the laminar region is approximately parabolic and becomes flatter in turbulent flows. Turbulent region can be considered of three regions; Laminar sub layer (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 convection heat transfer coefficient.

The objective of this numerical study is to investigate;

- The temperature distribution in a room caused by forced convection.
- Velocity profiles in a room due to forced convection.
- The impact of varying Renoldys and Pranditl numbers on temperature distribution and velocity profiles in a room.

II. Literature Review

The problem of convective heat transfer in an enclosure has been studied extensively because of a wide application of such process. Eckert & Carson [1] studied natural convection in an air layer enclosed between two vertical plates with different temperatures, the result showed there exists an optimum plate spacing with highest average Nusselt number occurring when natural convective heat transfer along plates reaches its maximum.

Ali & Hussein [2] investigated the effect of corrugation frequencies on natural convective heat transfer and flow characteristics in a square enclosure of veer-corrugated vertical walls. This investigation showed that the overall heat transfers through the enclosure increased with increase of corrugation for low Gashof number; but the effect was reversed for high. Rokin & Sunden [4] researched on turbulent forced convection in a duct with a trapezoidal cross section and the result showed that generalized gradient diffusion hypothesis (GGDH) and wealth equation equals earnings x time (WET) predicts higher Nusselt number than simple eddy diffusivity (SED) at higher Reynolds number but predict lower Nusselt number than SED at low Reynolds numbers. Hyung et al., [3] studied forced convection from isolated heat source in a channel with a porous medium and results showed that in view of pressure drop the employment of a thicker and denser porous substrate in electronic cooling is less desirable. Aydin & Young [5] 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 a diabetic.

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 [6] 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 for H/D=1.0 and Reynodys 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. Also Sigey [7] investigated in detail turbulent flow in a three dimensional enclosure in form of a room with convectional heater build into one of the walls and having a window in the same wall. Kipn'geno [8] also studied turbulent natural convection with localized heating at the bottom wall (the floor) and two windows each on the vertical opposite walls of a rectangular enclosure. The results showed that the room is stratified into regions with those near the floor being warmer and heat distribution reducing differently upwards. Ozturk & Tan [10] did CDF modeling of forced cooling of computer chassis the result showed agreement with experimental data.

Sathiyamoorthy et al., [9] studied natural convection flow in a closed square cavity when the bottom wall was uniformly heated and Vertical walls were linearly heated while the top wall was well insulated. Non-linear coupled governing equations were solved by using penalty finite element method with bi-quadratic rectangular elements. Numerical results were obtained for various values of Raleigh number and Prandtl number. Result were presented in the form of streamlines isotherm contours, local Nasselt number and average Nusselt number as a function of Rayleigh number.

Studies were also undertaken by Sigey et al., [13], 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. Hamid & Mohammed [12] 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 [11] 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 pranditl number lead to a decrease in temperature profiles.

Ghadhimi et al., [14] 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 Azizah et al., [15], on forced convection boundary layer flow along a horizontal cylinder in a porous medium filled by a nano fluid results showed that the presence of nano particles in the base fluid enhances heat transfer rate and temperature profiles. Also heat transfer rate increases with increasing nano particle volume fraction parameter and curvature parameters. Gareh [16] recently 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. Most researchers who have done similar work have used heat source only (natural convection) but for my case have used both heat source and the fan (forced convection). Hence, in these study effects of forced convection on temperature distribution and velocity profiles in a room which has been given minimal attention was investigated.

III. Mathematical Formulation

Figure 1 is a model of a room with heaters placed on the opposite walls (on x-z planes) windows on the two walls (x-y planes) and the fan at the top ceiling (on x-y planes). Turbulent forced convection in a room as result of heating and cooling is experienced in number of practical occurrences such as use of convectional heaters and cooling fans in rooms. The temperature and velocity fields in a room depend on temperature of any heat source windows as well as any other cooling agent such as a cooling fan. This is a numerical study of turbulent flow in a room. We considered forced convection in a three dimensional room with heaters placed on two vertical walls, windows on the other two vertical walls and a fan at the top (ceiling) of the room centrally fixed.

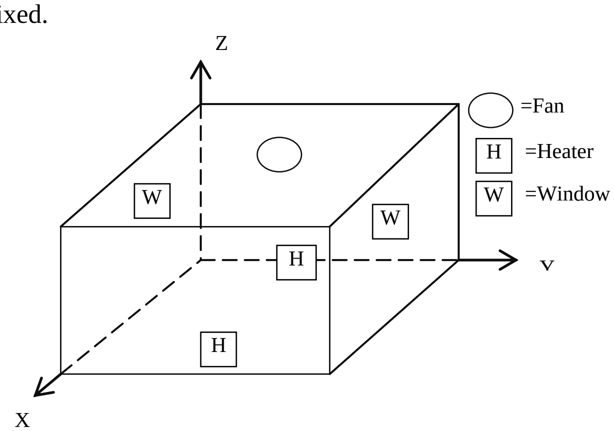

Figure 1: Model showing the Position of Heaters Windows and the Fan in the Enclosure

IV. Governing Equations

We considered the equations governing behavior of Newtonian fluids experiencing heat and mass transfer. These fundamental equations of fluid dynamics were based on the following universal laws conservation; conservation of mass (continuity), momentum and energy. These equations presented in tensor form as well as in Cartesian form useful for computer programming. Consider a fluid in which the density ρ is a function of position X_j (j=1,2,3) let U_j(j=1,2,3) denote the components of the velocity. Hence in writing the various equations, use of the notation of Cartesian tensors with the usual summation convection is applied.

4.1. Conservation Equations

The classical thermodynamics postulates that the thermodynamics state of a fluid is determined by only two independent thermodynamic properties. A third thermodynamic property is related to the two independent properties by the equations of state of the fluid ρ = ρ (T), where ρ is density, T is the thermodynamic temperature. A fluid with velocity component U_i in time t and space with Cartesian co-ordinate X_j is considered in the following equations.

4.2. 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;

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j}\left(\rho u_j\right) = 0 \tag{1}$$

For stead state equation above can be written as

$$\frac{\partial}{\partial x_j}\left(\rho u_j\right) = 0 \tag{2}$$

4.3. 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;

$$\frac{\partial\left(\rho U_i U_j\right)}{\partial X_j} = \frac{\partial P}{\partial X_j} + \frac{\partial}{\partial X_j} \quad (\mu + \mu_t)\frac{\partial U_j}{\partial X_j} \tag{3}$$

Where ρ is density, u is velocity vector, p is static pressure, μ is laminar viscosity and μ_t is turbulent eddy viscosity.

4.4. 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

$$\rho \frac{\partial h}{\partial t} + \rho u_i \frac{\partial h}{\partial x_j} = \frac{\partial \rho}{\partial t} + u_i \frac{\partial p}{\partial x_j} - \frac{\partial q_j}{\partial x_j} + \Phi \tag{4}$$

Where Φ is the viscous dissipation function given by;

$$\Phi = \mathrm{d}_{ij} \frac{\partial u_i}{\partial x_j}$$

And h is the specific enthalpy and q_j is the local rate of transfer per unit area.

Equations 1 to 4 are general continuity, momentum and energy equations.

V. METHOD OF SOLUTION

Given that the density varies linearly with temperature only, it follows that

$$\rho = \rho_R + (T - T_R) \frac{\partial \rho}{\partial T}_{T_R} \quad (5)$$

In which the higher order terms (HOT) of this series have been neglected

The thermal expansion of coefficient at constant pressure is defined as

$$\beta_R = \frac{1}{\rho_R} \frac{\partial \rho}{\partial T}_{T_R} \quad (6)$$

Substituting equation (6) into (5) yields

$$\rho = \rho_R \left((1 - \beta_R)(T - T_R)\right) \quad (7)$$

Applying the boussineq approximation

$$\frac{\partial U_i}{\partial x_j} = 0 \quad (8)$$

$$\frac{\partial U_j}{\partial t} = \frac{\partial U_i U_j}{\partial x_j} - \frac{\partial \rho}{\partial x_j} + \frac{\partial}{\partial x_j} \frac{1}{Re} \frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} - \overline{\mu_i \mu_j}\Bigg] \quad (9)$$

$$\frac{\partial \Theta}{\partial t} + \frac{\partial}{\partial x_j} \overline{u_j \theta} = \frac{\partial}{\partial x_j} - \frac{1}{Pr} \frac{\partial \Theta}{\partial x_j} - \overline{u_j \theta} \quad (10)$$

The turbulent stresses $\overline{u_i u_j}$ and the heat stress $\overline{u_j \theta}$ are given by

$$\overline{u_1 u_j} = \frac{2}{3} k \delta_{ij} - \left[v_t \frac{\partial u_i \partial u_j}{\partial x_j \partial u_j} \right. \quad (11)$$

$$\overline{u_1 \theta} = {}^{-v_1}\!/_{\delta_1} \frac{\partial \Theta}{\partial x_j} \quad (12)$$

Where v_t is the turbulent viscosity obtained from,

$$v_t = c_\mu \frac{k^2}{\varepsilon} \quad (13)$$

Substituting equation (13) into momentum equation (9) and simplifying gives;

$$\frac{\partial U_j}{\partial t} + u_i \frac{\partial U_j}{\partial x_j} + u_j \frac{\partial U_i}{\partial x_j} = -\frac{\partial \rho}{\partial x_j} + \frac{1}{Re} \frac{\partial^2 U_i}{\partial x_j^2} + \frac{\partial^2 U_i}{\partial x_i^2} - \frac{\partial}{\partial x_j} \left[{}^2\!/_3 k \delta_{ij} - v_t \frac{\partial U_i}{\partial x_i} \frac{\partial U_j}{\partial x_i} \right. \quad (14)$$

Also using equation 8 and 14

$$\frac{\partial U_j}{\partial t} + U_i \frac{\partial U_i}{\partial x_j} = -\frac{\partial \rho}{\partial x_j} + \frac{1}{Re} \frac{\partial^2 U_i}{\partial x_j^2} \quad (15)$$

Substituting (12) into energy equation (10) and using equation (8) we get;

$$\frac{\partial \Theta}{\partial t} + 2 U_i \frac{\partial \Theta}{\partial x_j} = \frac{1}{Pr} \frac{\partial^2 \Theta}{\partial x_j^2} \quad (16)$$

An equation 5 to 16 step by step deriving specific equations for the problem and bars represents turbulence.

Descritization

A hybrid finite difference scheme combining both forward and central difference methods

Equation (15) for momentum with respective substitution can be written in Cartesian coordinates in two dimensional flows as;

$$\frac{\partial u}{\partial t} = -\rho \frac{\partial P}{\partial x} + \frac{1}{R_e} \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} - v \left(\frac{\partial u}{\partial x} + \frac{\partial u}{\partial y} - \frac{\partial \rho}{\partial y} \right. \quad (17)$$

The momentum equation was crucial for the analysis of velocity profiles as well as the impact of pressure force by the fan in the room.

The equation was descritized as

$$\frac{u_{ij}^{n+1} - u_{ij}^n}{k} = -\rho \Bigg[\frac{P_{i+1,j} - P_{i-1,j}}{2h} + \frac{1}{R_e h^2} u_{i+1,j}^{\,n} - 2u_{i,j}^n + u_{i-1,j}^{\,n} + \big(u_{i,j+1}^{\,n} - 2u_{i,j}^n + u_{i,j-1}^{\,n} - \frac{v}{2h} (u_{i+1,j}^{\,n} - u_{i-1,j}^n) + u_{i,j+1}^{\,n} - u_{i,j-1}^{\,n} \quad (18)$$

Equation (16) for energy with respective substitutions

$$\frac{\partial \Theta}{\partial t} = \frac{1}{P_r R_e} \frac{\partial^2 \Theta}{\partial x^2} + \frac{\partial^2 \Theta}{\partial y^2} - 2v \left(\frac{\partial \Theta}{\partial x} + \frac{\partial \Theta}{\partial y} \right. \quad (19)$$

The energy equation was also important for the analysis of temperature distribution within the room.

It was descritized as

$$\frac{\Theta_{i,j}^{n+1} - \Theta_{i,j}^n}{k} = \frac{1}{PrRh^2} \Theta_{i+1,j}^{\,n} - 2\Theta_{i,j}^n + \Theta_{i-1,j}^{\,n} + \big(\Theta_{i,j+1}^{\,n} - 2\Theta_{i,j}^n + \Theta_{i,j-1}^{\,n} - \frac{v}{h} (\Theta_{i+1,j}^{\,n} - \Theta_{i-1,j}^n) + (\Theta_{i,j+1}^{\,n} - \Theta_{i,j-1}^{\,n}) \quad (20)$$

Equations 18 and 20 were descritized so that it can be appropriately fed into the software to generate the results.

VI. RESULTS AND DISCUSSION

Equations 17 and 19 were descritized as indicated in equations 18 and 20 respectively. Linear algebraic equations were formed from the descritized equations. Using the algebraic equations tri-diagonal matrix was obtained. This matrix was then solved using the software which generated the results. As sample calculation using equation 20, setting Re =5500, Pr=0.7, $\Delta x = \Delta y = 0.5$, i=1, j= 1, 2, 3…. together with boundary conditions a set of algebraic equations were obtained as follows,

$$38480\theta_{12} + 38580\theta_{11} - 38520\theta_{10} = 2\theta_{01} + 2\theta_{10}$$
$$38480\theta_{13} + 38580\theta_{12} - 38520\theta_{11} = 2\theta_{02} + 2\theta_{11}$$
$$38480\theta_{14} + 38580\theta_{13} - 38520\theta_{12} = 2\theta_{03} + 2\theta_{12}$$

$$3848\theta_{15} + 3858\theta_{14} - 3852\theta_{13} = 2\theta_{04} + 2\theta_{13}$$
$$3848\theta_{16} + 3858\theta_{15} - 3852\theta_{14} = 2\theta_{05} + 2\theta_{14}$$
$$3848\theta_{17} + 3858\theta_{16} - 3852\theta_{15} = 2\theta_{06} + 2\theta_{15}$$

The set of algebraic equations are written in matrix form as follows;

$$\begin{bmatrix} -3852 & 3858 & 3848 & 0 & 0 & 0 \\ 0 & -3852 & 3858 & 3848 & 0 & 0 \\ 0 & 0 & -3852 & 3858 & 3848 & 0 \\ 0 & 0 & 0 & -3852 & 3858 & 3848 \\ 0 & 0 & 0 & 0 & -3852 & 3858 \\ 0 & 0 & 0 & 0 & 0 & -3852 \end{bmatrix} \begin{bmatrix} U_{10} \\ U_{11} \\ U_{12} \\ U_{13} \\ U_{13} \\ U_{14} \\ U_{15} \end{bmatrix} = \begin{bmatrix} -5.682598458*10^5 \\ -5.682598458*10^5 \\ -5.682598458*10^5 \\ -5.682598458*10^5 \\ -5.682598458*10^5 \\ -5.682598458*10^5 \end{bmatrix}$$

Solving the matrix using MATLAB the solutions are as follows;

$$U_{10} = 2.95644*10^3$$
$$U_{11} = 1.7319*10^3$$
$$U_{12} = 1.0340*10^3$$
$$U_{13} = 5.9063*10^2$$
$$U_{14} = 2.95276*10^2$$
$$U_{15} = 1.47523*10^2$$

Refer results in table 1 in the row when Re =5500.The rest of the results were obtained using same procedure as above.

6.1. Temperature and Velocity Results and Discussion

Table 1: Temperature against Room Height Varying Reynolds Number

Room height / Re number	1	2	3	4	5	6
5500	$2.95644*10^3$	$1.7319*10^3$	$1.0340*10^3$	$5.9063*10^2$	$2.95276*10^2$	$1.47523*10^2$
5000	$3.21357*10^3$	$1.92666*10^3$	$1.1231*10^3$	$6.41286*10^2$	$3.2051*10^2$	$1.60073*10^2$
3500	$4.6197*10^3$	$2.7687*10^3$	$1.6135*10^3$	$9.0965*10^2$	$4.0199*10^2$	$2.2972*10^2$

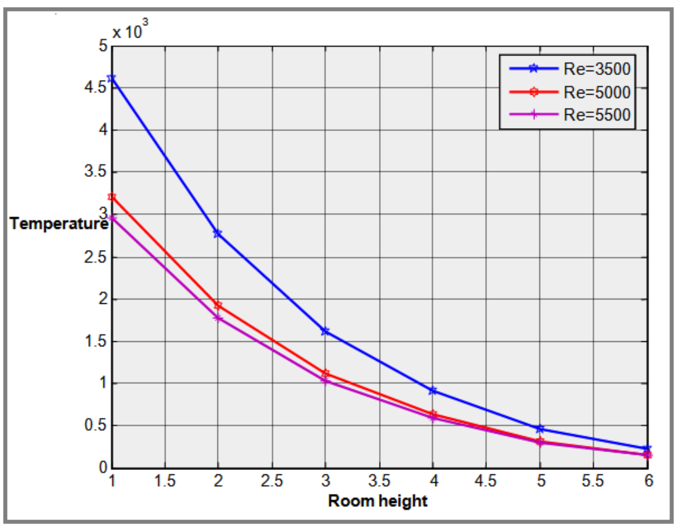

Figure 2: Graph of Temperature against Room Height

The heaters, windows and the fan were switched on simultaneously. Temperature decreased with increase in room height due to the cooling at top influenced by the fan. Also fluid flow started at low temperature when Re number was high than when it was low. This was because at high Re number inertial forces were predominant.

Table 2: Temperature against Room Height Varying Pranditl Number

Room height / Pr number	1	2	3	4	5	6
0.05	$2.95645*10^2$	$1.7732*10^2$	$1.0340*10^2$	$5.9063*10^1$	$2.95276*10^1$	$1.47523*10^1$
0.1	$2.08824*10^3$	$1.2496*10^3$	$7.2737*10^2$	$4.1441*10^2$	$2.07010*10^2$	$1.02946*10^1$
0.7	$4.22039*10^3$	$2.5189*10^3$	$1.463*10^3$	$8.3106*10^2$	$4.1476*10^2$	$2.05148*10^2$

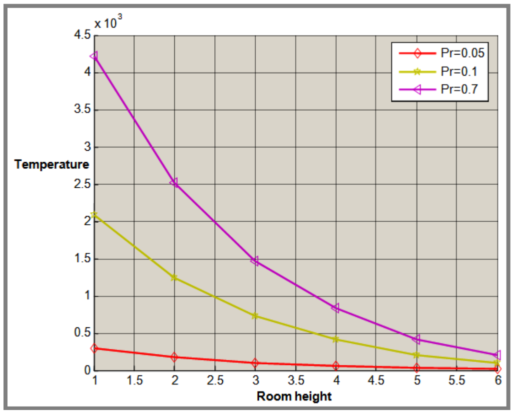

Figure 3: Temperature against Room Height with Varying Prandtl Numbers

The heaters, windows and the fan were switched on simultaneously. From the graph, temperature decreased with increase in room height due to the cooling impact at top of the room by the fan. Also fluid flow started at low temperature when Prandtl number was low than when it was high. At low Pranditl number thermal diffusivity was dominant while at high Pranditl number momentum diffusivity was dominant.

Table 3: Velocity against Room Height Varying Reynolds Number

Room height / Re number	1	2	3	4	5	6
5500	8.655*10^3	5.2249*10^3	2.299*10^3	5.8315*10^2	4.0788*10^1	8.1576*10^1
5000	7.898*10^3	4.9136*10^3	2.136*10^3	5.6832*10^2	4.0480*10^1	7.4161
4500	7.1083*10^3	4.5022*10^3	2.073*10^3	4.8349*10^2	3.8372*10^1	6.6745

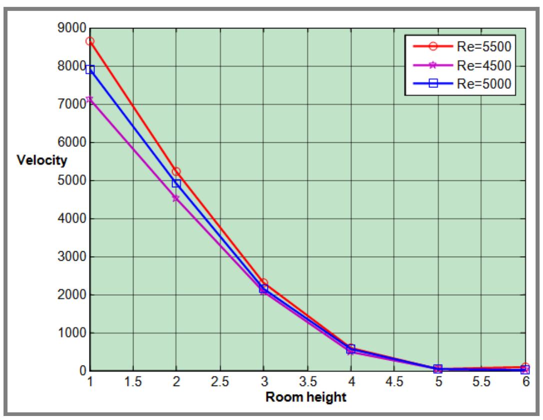

Figure 4: Velocity against Room Height Varying Reynolds Number

Velocity decreased as the fluid particles flow up the room because at bottom of the room we had the heat source and at top we had the cooling fan. The heat source acted as the propeller of the fluid particles and the fan cooled down the fluid particles making them denser hence decreasing velocity. Initial velocity at high Re number was higher than at low Re number. At high Re number inertial forces were dominant while at low Re number viscous forces were dominant.

Table 4: Velocity against Room Height Varying Pressure

Pressure	Room Height					
	1	2	3	4	5	6
123kpa	$-3.84994*10^3$	$-2.243*10^3$	$-6.866*10^2$	$-2.03376*10^2$	$-5.352*10^1$	$-1.338*10^1$
223kpa	$-5.66154*10^3$	$-3.467*10^3$	$-1.51506*10^3$	$-4.04016*10^2$	$-1.0632*10^2$	$-2.658*10^1$
443kpa	$-8.4731*10^3$	$-5.278*10^3$	$-2.26746*10^3$	$-6.04656*10^2$	$-1.5912*10^2$	$-3.978*10^1$
663kpa	$-1.128474*10^4$	$-7.124*10^3$	$-3.01986*10^3$	$-8.05296*10^2$	$-2.1192*10^2$	$-5.298*10^1$

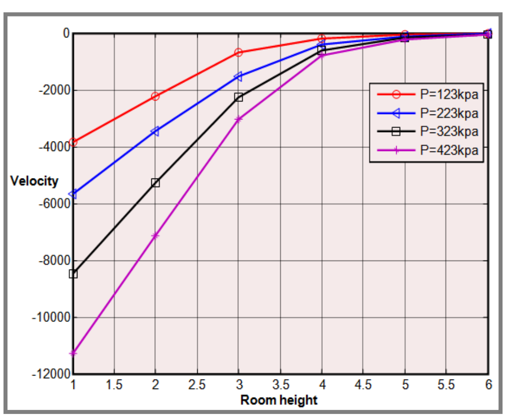

Figure 5: Velocity against Room Height Varying Pressure

Analysis of pressure variation down the room from fan area i.e. down flow, velocity was highest at lowest pressure and lowest at highest pressure. The fan caused low pressure (according to Bernoulli's principle) on assumption that influence of gravitational force was negligible. Generally at different levels of pressure variation velocity increased down the room.

Table 5: Temperature against Room Height Varying Pressure

Pressure	Room height					
	1	2	3	4	5	6
123kpa	$2.9564*10^2$	$1.773*10^2$	$1.034*10^2$	$5.906*10^1$	$2.9527*10^1$	$1.4752*10^1$
223kpa	$7.657*10^2$	$4.593*10^2$	$2.678*10^2$	$1.529*10^2$	$7.6476*10^1$	$3.8208*10^1$
333kpa	$1.1434*10^3$	$6.8579*10^2$	$3.999*10^2$	$2.284*10^2$	$1.1420*10^2$	$5.7056*10^1$

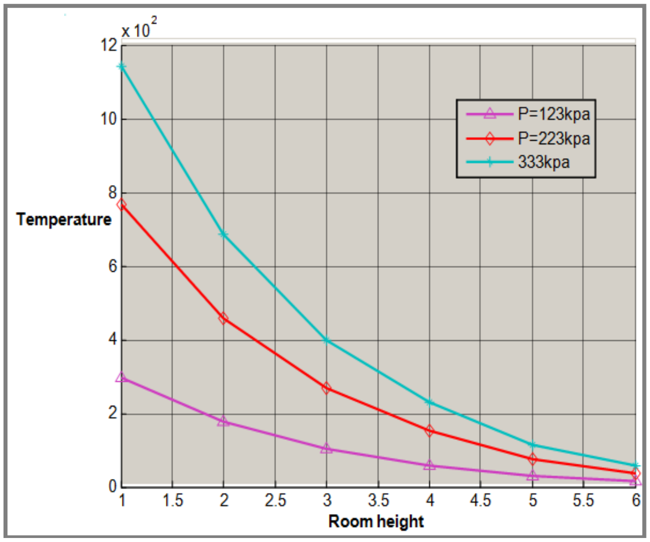

Figure 6: Temperature against Room Height with Varying Pressure

Temperature decreased with increase in room height. Also temperature was directly proportional to pressure. Decrease in pressure lead to decrease in temperature this was due to the influence of the fan.

VII. Conclusion

In figure 4 the heaters caused the fluid to gain energy i.e. warm up became less dense and gained kinetic energy and lead to an upward motion while the fan sped up cooling as well as fluid flow rate. According to Bernoulli's principle rotation of the fan induced low pressure around it i.e. according to figure 5. This brought about variation of pressure decreasing from the fan area downward. Hence at ceiling the fluid lost energy became denser resulting in downward motion. The windows too increased the rate of cooling but not as faster as the fan. In figure2 amount of heat transfer was high at high Re number than low Re number. At high Re number inertial forces were predominant. Similarly this was also reflected in figure 4 where at high Re number initial velocity was high than when Re number was low since at high Re number inertial forces were predominant. In figure 3, the fluid started diffusing at low temperature than at high temperature since at low Pr number thermal diffusivity was dominant. In Figure 6, temperature decreased up the room and reduction of pressure by the fan lead to low temperatures. Therefore, velocity of the fluid was highest near the heaters followed by immediately leaving the fan area and lowest as it approaches the fan area. The temperature was highest near the heater region followed by the window regions and then the fan area had the lowest temperature. The results show that forced convection affects velocity profiles and temperature distribution in a room.

VIII. Recommendations

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

References

[1] E.R. Eckert & W.D. Carson (1961), "Natural Convection in an air Layer Enclosed between Two Vertical Plates with Different Temperature", *International Journal of Heat and Mass Transfer*, Vol. 2, Pp. 106–120.

[2] M. Ali & S.R Hussein (1993), "Effect of Corrugation Frequencies on Natural Convective Heat Transfer and Flow Characteristic in an Enclosure of Veer – Corrugated Vertical Walls", *International Journal of Energy Research*, Vol. 17, Pp. 679–708.

[3] J.S. Hyung, Y.K. Seo & M.H. Jae (1995), "Forced Convection from Isolated Heat Source in a Channel with Porous Medium", *International Journal of Heat and Fluid Flow*, Vol. 16, Pp. 527–535.

[4] M. Rokin & B. Sunden (1995), "Numerical Investigation of Turbulent Forced Convection in Adduct with a Trapezoidal Cross Section", *Advances in Engineering Heat Transfer*, Vol. 1, Pp. 321–332.

[5] O. Aydin & J. Young (2000), "Natural Convection in Enclosures with Localized Heating in below and Symmetrically Cooling from Sides", *International Journal of Numerical Methods for Heat & Fluid Flow*, Vol. 1, Pp. 518–529.

[6] O. Manca (2003), "Effects of Heated Wall Position on Mixed Convection in a Channel with an Open Cavity", *Numerical Heat Transfer*, Vol. 43, Pp. 259–282.

[7] J.K. Sigey (2004), 'Three-Dimensional Buoyancy Driven Natural Convection in an Enclosure", *PhD Thesis, JKUAT*, Kenya.

[8] J. Kipn'geno (2006), "Natural Convection with Localized Heating and Cooling on Opposite Vertical Walls of an Enclosure", *Msc Thesis*, Kenyatta University, Kenya.

[9] T. Sathiyamoorthy, J.R. Basak & N.C. Mahanti (2007), "Effect of the Temperature Difference Aspect Ratio on Natural Convection in a Square Cavity for Non-Uniform Thermal Boundary Conditions", *Journal of Heat Transfer*, Vol. 129, Pp. 1723–1728.

[10] E. Ozturk & I. Tan (2007), "CDF Modeling of Forced Cooling of Computer Chasis", *Engineering Applications of Fluid Mechanics*, Vol. 4, Pp. 304–313.

[11] M.Z. Salleh (2011), "Numerical Solutions of Forced Convection Boundary Layer Flow on a Horizontal Circular Cylinder with Newtonian Heating", *Malaysian Journal of Mathematical Sciences*, Vol. 5, Pp. 161–184.

[12] R.G. Hamid & R.S. Mohammed (2011), "Investigation of Turbulence Mixed Convection in Air Filled Enclosures", *Journal of Chemical Engineering and Material Science*, Vol. 2, Pp. 87–95.

[13] J.K. Sigey, F. Gatheri & M. Kinyanjui (2011), "Buoyancy Driven Free Convection Turbulent Heat Transfer in an Enclosure", *Journal of Agriculture, Science and Technology*, Vol. 12, No. 1.

[14] M. Ghadhimi, H. Ghadamian, A.A. Hamidi, F. Fazelpour & M.A. Behghadan (2012), "Analysis of Free and Forced Convection in Air Flow Windows using Numerical Simulation of Heat Transfer", *International Journals of Energy and Environmental Engineering*, Vol. 3, Pp. 1–10.

[15] M.R. Azizah, A. Syakila & P. Ioan (2013), "Forced Convection Boundary Layer Flow along a Horizontal Cylinder in Porous Medium Filled by a Nano Fluid", *International Journal of Humanities and Management Sciences*, Vol. 1, Pp. 23–28.

[16] S. Gareh (2014), "Numerical Study of Forced Convection in a Rectangular Channel", *Journal of Chemistry and Material Research*, Vol. 1, Pp. 7–11.

Joseph Momanyi Nyabuto, holds a Bachelor of Education degree in Mathematics & Physics from Kenyatta University, main Campus, Kenya and is currently pursuing a Master of Science degree in Applied Mathematics from Jomo Kenyatta University of Agriculture and Technology, Kenya.

Affiliation: Jomo Kenyatta University of Agriculture and Technology, (JKUAT), Kenya.

Teaching Experience: He is currently senior teacher at Borangi S.D.A secondary school near Kisii, Kenya. He has been full time teacher since 2001 up date. He has interest in the study of fluid mechanics and especially heat flow in various enclosures both free and forced convection and its application in engineering.

Prof. Johana Kibet Sigey, holds a Bachelor of Science degree in mathematics and computer science first class honors from Jomo Kenyatta University of Agriculture and Technology, Kenya, Master of Science degree in Applied Mathematics from Kenyatta University and a PhD in Applied Mathematics from Jomo Kenyatta University of Agriculture and Technology, Kenya.

Affiliation: Jomo Kenyatta University of Agriculture and Technology, (JKUAT), Kenya.

Teaching Experience: He is currently the director, JKUAT, Kisii CBD Campus. He has been the substantive chairman - Department of Pure and Applied Mathematics – JKUAT (January 2007 to July-2012). He holds the rank of Associate Professor, in applied mathematics in Pure and Applied Mathematics Department – JKUAT since November 2009 to date. He has published 15 papers on heat transfer in respected journals.

Dr. JeconiaOkelo Abonyo, holds a PhD in Applied Mathematics from Jomo Kenyatta University of Agriculture and Technology as well as a Master of science degree in Mathematics and first class honors in Bachelor of Education, Science; specialized in Mathematics with option in Physics, both from Kenyatta University. He has dependable background in Applied Mathematics in particular fluid dynamics, analyzing the interaction between velocity field, electric field and magnetic field. Has a hand on experience in implementation of curriculum at secondary and university level. He has demonstrated sound leadership skills and ability to work on new initiatives as well as facilitating teams to achieve set objectives. Has a good analytical, design and problem solving skills.

Affiliation: Jomo Kenyatta University of Agriculture and Technology, (JKUAT), Kenya. 2011-To date Deputy Director, School of Open learning and Distance e-Learning SODeL Examination, Admission & Records (JKUAT), Senior lecturer Department of Pure and Applied Mathematics and Assistant Supervisor at Jomo Kenyatta University of Agriculture and Technology. Work involves teaching research methods and assisting in supervision of undergraduate and postgraduate students in the area of applied mathematics. He has published 10 papers on heat transfer in respected journals.

Dr. James Mariita Okwoyo, holds a Bachelor of Education degree in Mathematics and Physics from Moi University, Kenya, Master Science degree in Applied Mathematics from the University of Nairobi and PhD in applied mathematics from Jomo Kenyatta University of Agriculture and Technology, Kenya.

Affiliation: University of Nairobi, Chiromo Campus School of Mathematics P.O. 30197-00100 Nairobi, Kenya. He is currently a lecturer at the University of Nairobi (November 2011 – Present) responsible for carrying out teaching and research duties. He plays a key role in the implementation of University research projects and involved in its publication. He was an assistant lecturer at the University of Nairobi (January 2009 – November 2011). He has published 7 papers on heat transfer in respected journals.