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An Investigation on Ventilation Air distribution within Rectangular and Circular Enclosures

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ABSTRACT

One of the important parameters that dictate the effectiveness of ventilation system is air change rate. The air change rate provides overall effectiveness of ventilation in an enclosed space and it needs to be examined with age of air particles to decide about overall effectiveness. Age of air particles is the time air at a location spends inside the enclosure and hence provides useful information about presence of secondary flows. A range of tools have been used to predict local and global effectiveness of a typical ventilation system for a specific work environment. Computational fluid dynamics has proven to be an effective tool in this regard. The present work highlights various issues needed to keep in consideration while designing meeting rooms in busy manufacturing environments. Two meeting room configurations have been proposed and ventilation requirements for these enclosures have been estimated. Furthermore for two different fan configurations, interior enclosure conditions have been predicted.

Keywords: Computational Fluid Dynamics, Drill Bit, Velocity Ratio, Static Pressure

INTRODUCTION

The current trend in busy working environments is to develop work specific enclosures for satisfactory work outputs. The work place environment is changing from generic work environments to specific work environments. Work specific enclosures need to be designed specifically for the requirements of the work. Provisions must be made for comfortable working environments in such locations. One of the most important requirements is provision of ventilation air with little noise. The ventilation effects can be best represented by movement and distribution of air in the space. The air distribution and its travel path are fairly complex and these depend on many geometric, fluid and flow parameters. These parameters include air inlets and outlets, doors and windows, wall cracks etc. Presence of various thermal sources also affects local flow patterns

(Murakami et. al., 1992 and Lim et. al., 2007 and Jiang et. al., 2004)

MODELLING AND SIMULATION

To understand effectiveness of rectangular and circular meeting room configurations in providing better ventilation performance, a detailed fluid dynamic investigation of the proposed system was carried out.

Three dimensional Navier Stokes Equations (eq. (1)), are numerically solved for turbulent flow of air in the enclosures to obtain the flow velocity components in x, y and z directions. The pseudo-time iteration method is used to iterate the solution until steady state solution is achieved. The pressure correction technique is implemented by assuming a pressure distribution to estimate velocity components and then correcting the pressure gradient to satisfy the continuity equation (eq. (2)). This technique is explained in details in several computational fluid dynamics textbooks [1, 2, and 3]. A simple flow chart for the computational technique is shown in fig. 1.

$$\frac{\partial(\rho u)}{\partial t} + div(\rho u \mathbf{u}) = -\frac{\partial p}{\partial x} + div(\mu\, grad\, u) + SM_x$$

$$\frac{\partial(\rho v)}{\partial t} + div(\rho v \mathbf{u}) = -\frac{\partial p}{\partial y} + div(\mu\, grad\, v) + SM_y$$

$$\frac{\partial(\rho w)}{\partial t} + div(\rho w \mathbf{u}) = -\frac{\partial p}{\partial z} + div(\mu\, grad\, w) + SM_z \qquad \ldots\ldots\ldots\ldots\ldots (1)$$

$$\frac{\partial \rho}{\partial t} + div(\rho u) = 0 \qquad \ldots\ldots\ldots\ldots\ldots (2)$$

Numerical flow models were developed for enclosure configurations as shown in figs. 2(a) and 2(b) considering the symmetry about the vertical and axial planes through the axis of the duct-fan supplying the ventilation air. The models used half of the geometry to reduce memory requirements and hence symmetry boundary condition was used. Commercial CFD package was utilized to regenerate the flow geometry. The geometric details of the models developed are shown in fig. 3.

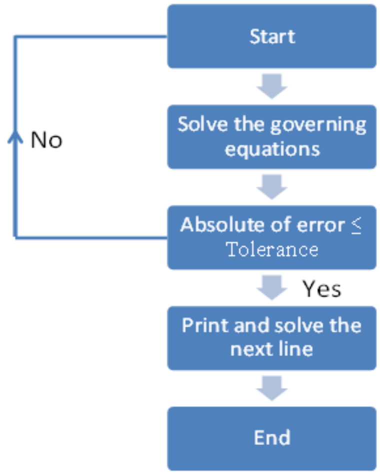


Fig. 1 Computational fluid dynamics flowchart

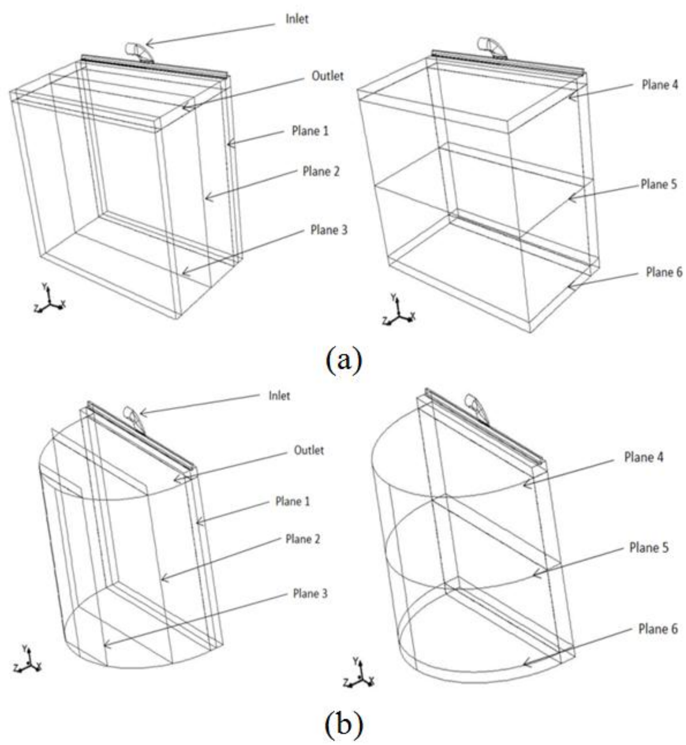


Fig. 2: (a) Rectangular Room (b) Circular Room

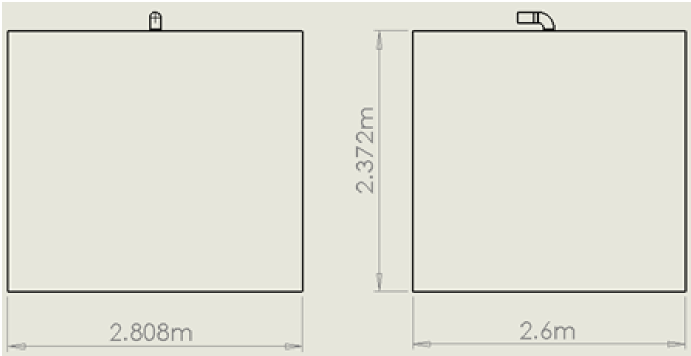


Fig. 3 Geometric details of the rooms

The developed models were then supplied with necessary boundary conditions and solver settings. These details are summarised in table 1.

Table 1 Boundary conditions and solver settings

Parameter	**Description**
Solver	Pressure based steady
Gradient	Green Gauss Node based
Discretisation scheme	2^{nd} order upwind
Turbulence model	k-ε
Inlet	Velocity inlet
Outlet	Pressure outlet at zero gauge
Walls	No slip

The flow of air inside the room has been assumed to be steady to simplify the numerical modelling of the flow phenomena. The air flow velocities within the enclosures have negligible compressibility effects and hence pressure based solver has been chosen. Green gauss node based gradient and 2nd order upwind spatial discretisation scheme have been specified for better accuracy of the results. K-ε turbulence model has been shown to provide fairly accurate solution for such problems. The outlet has been assumed to be at atmospheric pressure and no-slip boundary conditions have been specified to the walls of the enclosures.

RESULTS AND DISCUSSION

The static pressure differential between the inlet and outlet for a given flow rate for both type of enclosures was obtained from the flow models. As expected the pressure drop values increased with increasing flow rates for both the geometries. However, the flow rates against pressure differential characteristics for the rectangular and circular geometries have negligible difference (fig. 4). The rectangular box has a slightly higher pressure drop for a given flow rate. The European Standard specifies the requirement of 144 m3/hr ventilation air for the room occupied by 4 adults. Therefore, the fan power requirement can be estimated for effective ventilation. From the curves it can be seen that any fan providing 144 m3/hr flow rate and 40 Pa pressure will be able to satisfy the requirements. The characteristics of a fan at two different speeds were superimposed on room ventilation requirement curve to obtain operating conditions.

As can be seen from the figures, for two angular speeds the fan characteristics are different. The steady state operating points for the fan were obtained from the co-ordinates of the points of intersection of system and fan curves. This gives:

a) Flow rate of 175 m3/hour for fan speed of 1900 rpm for both rectangular and circular enclosures

b) Flow rate of 203 m3/hour for fan speed of 2250 rpm for both rectangular and circular enclosures

The flow features obtained in the enclosures have been discussed to analyse the global effectiveness of the proposed ventilation system. The air distribution within the enclosures has been depicted through the use of pressure and velocity contours on the various planes shown in figs. 2(a) and 2(b) respectively for both rectangular and circular enclosures. The axes shown are defined as:

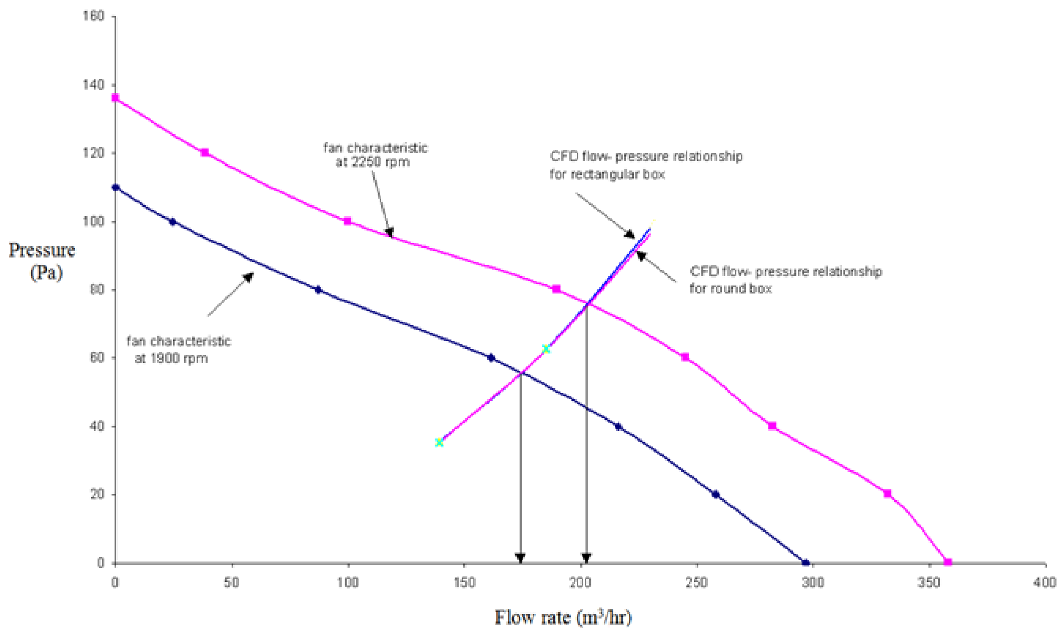


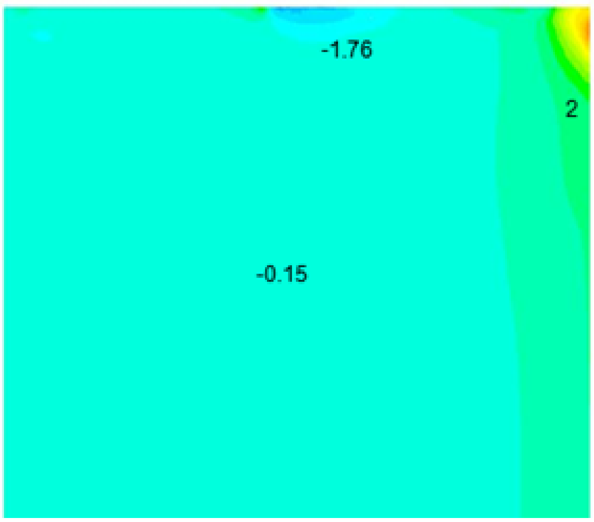
Fig. 4 System and fan characteristics

- x axis : direction of centreline of duct fan. Fan exit to inlet to cowl is +ve direction
- y axis : vertical upward direction
- z axis : horizontal direction from air exit slot towards the box interior.

The flow parameter's distribution within these enclosures at 175 m3/hr and 203 m3/hr i.e. 3 m/sec and 5 m/sec of inlet velocity, is discussed below.

Figures 5(a) and 5(b) show the pressure and velocity distributions on plane 1 in rectangular enclosure at 175 m3/hr. The results depict low pressure of -1.76 Pa in the vicinity of the air inlet to the enclosure and high pressure in excess of 2 Pa at the wall. The pressure distribution at other locations is fairly uniform. Figure 5(b) depicts high velocity of 2.75 m/sec at the top of the enclosure where the air has been introduced. This distribution of pressure and velocity of the flow inside the enclosure is in agreement with the expected distribution.

Figures 6(a) and 6(b) show the pressure and velocity distributions on plane 1 in rectangular enclosure at 203 m3/hr. The results depict the same trend as seen in case of 175 m3/hr i.e. low pressure and high velocity region in the vicinity of the air inlet to the enclosure. The pressures and velocities at the corresponding locations are however higher for 203 m3/hr.



(a)

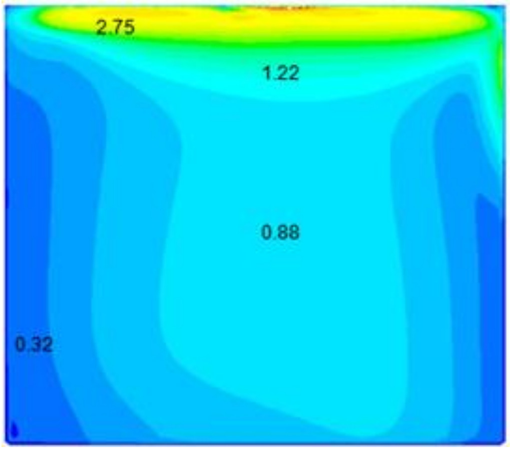
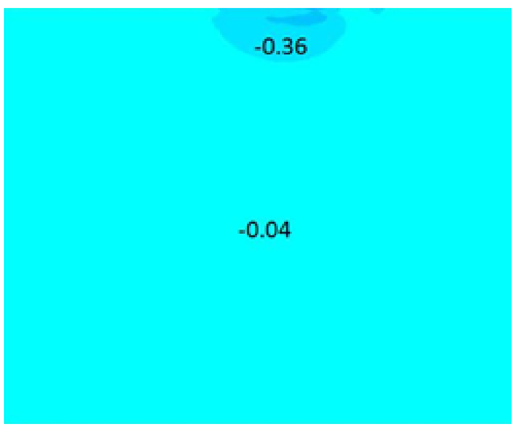
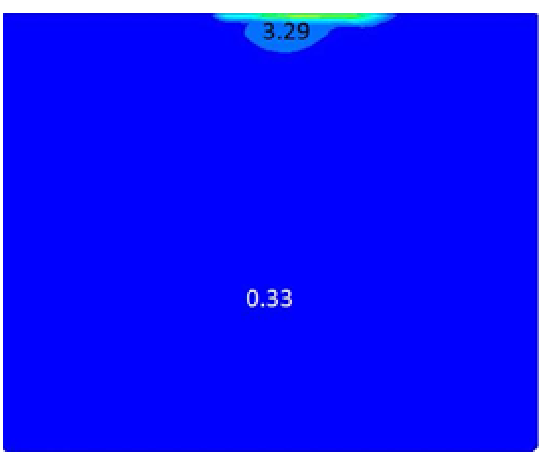


Fig. 5: (a) Pressure field on plane 1 at 175 m3/hr
(b) Velocity field on plane 1 at 175 m3/hr



(a)

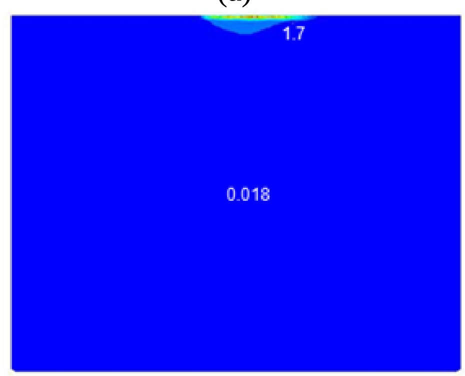


(b)

Fig. 6: (a) Pressure field on plane 1 at 203 m3/hr
(b) Velocity field on plane 1 at 203 m3/hr

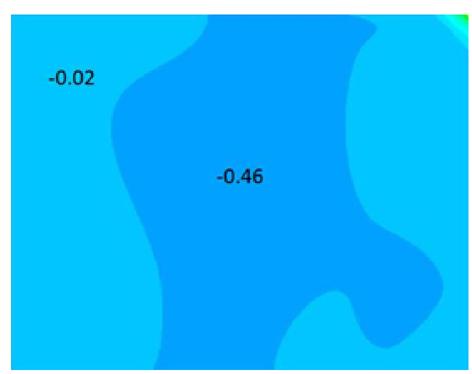
In order to analyse the uniformity of the air flow inside the circular enclosure at various flow rates, pressure and velocity fields have been shown in figs. 7 and 8. It can be seen that the area near the air intake depicts region of lower pressure and higher velocity. Pressure and velocity are higher for higher inflow rate of 203 m3/hr as compared to lower flow rate of 175 m3/hr.

(a)

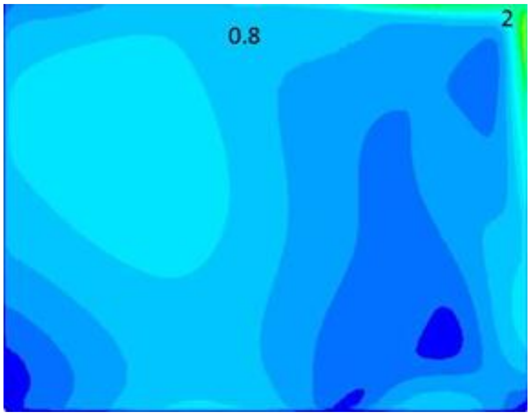


(b)

Fig. 7: (a) Pressure and (b) velocity fields on plane 1 at 175 m3/hr



(a)



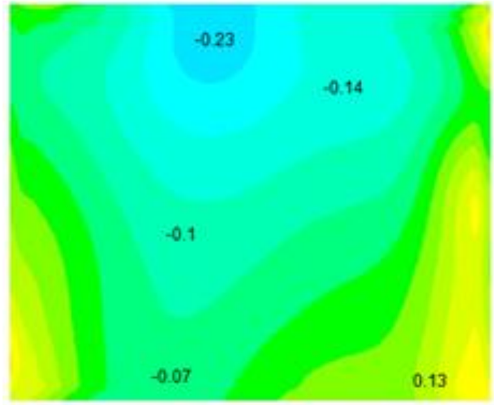
(b)

Fig. 8: (a) Pressure and (b) velocity fields on plane 1 at 203 m3/hr

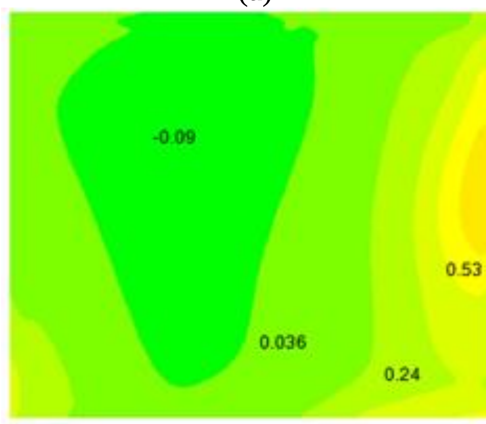
The aforementioned results indicate that the air flow inside the enclosures is fairly uniform at the mentioned flow rates and hence fulfils the requirement of adequate ventilation and air supply. In order to further investigate the uniformity inside the enclosures, pressure fields have been included here at locations away from the inlet of the air flow i.e. planes 2 and 3 in figs. 2(a) and 2(b).

Figures 9(a) and 9(b) depict the pressure distribution at plane 2 and 3 respectively for rectangular enclosure at an inflow rate of 175 m3/hr. The results show that the pressure increases from -1.76 Pa to -0.23 Pa and from -0.23 Pa to -0.09 Pa in the direction away from the inflow to the enclosure. The results further suggest that the flow has been distributed in the enclosure reasonably uniformly.

Figures 10(a) and 10(b) depict the pressure distribution at plane 2 and 3 respectively for circular enclosure at an inflow rate of 175 m3/hr. The results show the same trend as for the rectangular enclosure i.e. the pressure increases, and hence velocity decreases, away from the location of inflow to the enclosure. The pressure increases from -0.17 Pa to 0.004 Pa and from 0.004 Pa to 0.007 Pa from plane 1 to 2 and from plane 2 to 3 respectively.



(a)



(b)

Fig. 9: Pressure distribution at (a) Plane 2 and (b) Plane 3

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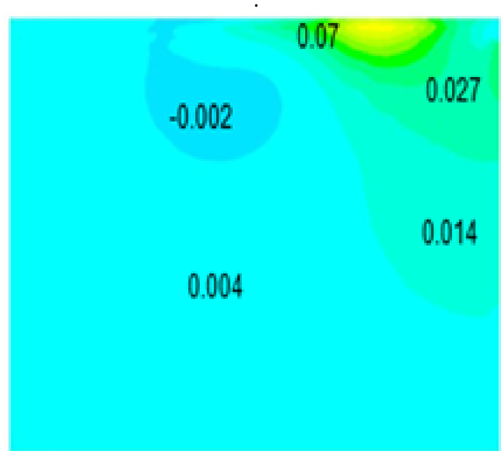
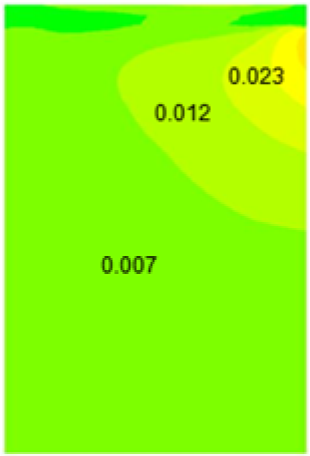


(a)



(b)

Fig. 10: Pressure distribution at (a) Plane 2 and (b) Plane 3

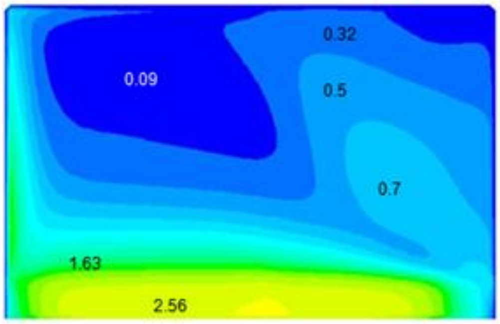
To investigate the distribution of the air flow inside the enclosures virtual planes have been constructed inside the enclosures in the vertical direction for analysis purposes. These planes corresponds to planes 4, 5 and 6 in figs. 2(a) and 2(b) where plane 4 is the one nearest to the inflow and 6 being the farthest.

Figures 11(a), 11(b) and 11(c) depict the velocity distribution at the aforementioned planes for rectangular enclosure at a flow rate of 175 m3/hr. Figures 12(a), 12(b) and 12(c) depicts the velocity fields on the vertical planes for circular enclosure at inflow rate of 175 m3/hr.

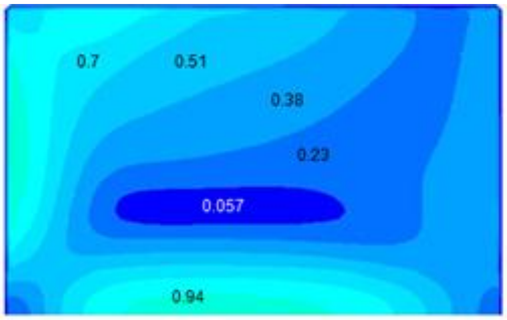
The results show that the flow velocity is highest near the inflow and reduces at the locations away from the inlet. Furthermore, flow velocity reduces as the vertical distance from the inlet increases. For example, in figs. 11(a-c), the velocity decreases from 2.56 m/sec to 0.94 m/sec and then from 0.94 m/sec to 0.64 m/sec as the flow propagates from plane 4 to 5 and 5 to 6 respectively in rectangular enclosure supplied with an inlet velocity of 3 m/sec.

Furthermore, in case of circular enclosure with an inflow rate of 175 m3/hr, the velocity of the flow reduces from 0.2 m/sec to 0.001 m/sec and from 0.001 m/sec to 0.0003 m/sec moving from plane 4 to 5 and plane 5 to 6 respectively.

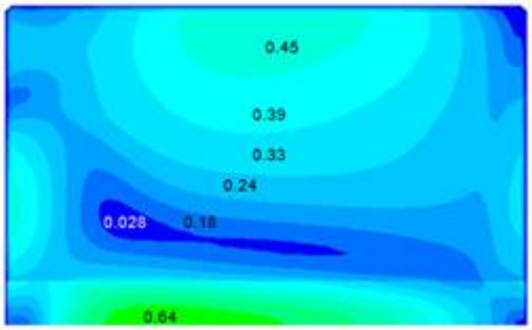
In the simulations the enclosures have been assumed to be empty. Hence, the fan capacity will be adequate to supply the minimum ventilation air, even after including the reduction of air flow which may occur due to occupancy of the room, assuming that the ratio of the volume of the occupying objects to the volume of the empty box is less than 0.1.



(a)

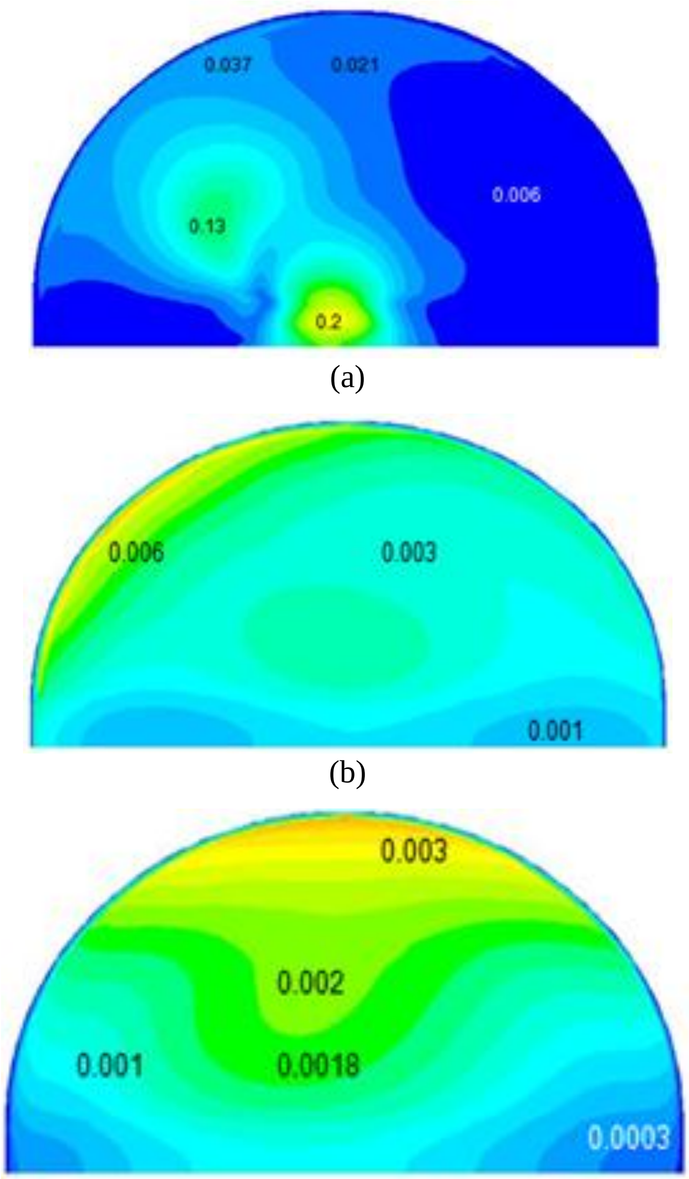


(b)



(c)

Fig. 11: Velocity contours at (a) Plane 4, (b) Plane 5 and (c) Plane 6



(a)

(b)

(c)

Fig. 12: Velocity contours at (a) Plane 4, (b) Plane 5 and (c) Plane 6

CONCLUSION

The ventilation system was modelled reasonably accurately through the commercial CFD software available. The flow models developed indicate that the fan capacity is adequate to provide minimum ventilation air requirement as per European standard. The air is also reasonably well distributed for the purpose. There is a scope for further modifications in the system.

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