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Numerical Simulation of Ventilation Air Movement in Partitioned Offices

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Abstract

Good air quality can only be assured throughout an office complex if each workspace receives an adequate supply of ventilation air. The likelihood of achieving this situation would be increased if the building engineer had a means of easily predicting the air movement in each office configuration. A simple computer-based solution to this need is proposed. To this end, the development and validation testing of a numerical solution technique to simulate the ventilation air movement in a room or office is described. The predictions of the two-dimensional, isothermal, inviscid formulation are seen to be in good agreement with experimentally measured airflows in configurations of interest. The computer code is then used to illustrate the airflow in offices served by a single row of supply air diffusers, when partitions are used to divide the space into smaller workspaces. It is observed that the partitions distort the airflow patterns to the extent that it would be difficult to provide desirable ventilation airflows to all the workspaces formed by the partitions.

KEY WORDS:
Ventilation simulation, Office ventilation, Numerical simulation, Isothermal airflow, Inviscid airflow

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Introduction

Ventilation airflow distribution is the key to good air quality in indoor environments. Unless airflow distribution is understood, efforts to provide desirable indoor air quality may be wasted, or at best, very ineffective. This realization has prompted considerable effort toward computer simulations of ventilation air movement in buildings over the past two decades.

Nielsen (1975) developed a procedure for predicting two-dimensional steady-flow patterns in ventilated rooms and incorporated the k-ε turbulence model proposed by Launder and Spalding (1974) into his system of equations. His numerical procedure, based on the work of Gosman et al. (1969), solved for the vorticity, stream function, temperature and the two turbulent terms k and ε. With the temperature difference between inlet and outlet specified, the predicted velocity and temperature distributions are in reasonably good agreement with experimentally obtained data.

Timmons et al. (1980) also used the vorticity-stream function approach for their 2-D inviscid model. Using established jet theory, as well as his own experimental findings, Timmons (1984) was able to formulate a solution method for the velocity with a Poisson form of equation. An iterative scheme was used in the solution. For the cases considered, this approach produced results which correlate quite well with experimental data. However, as Timmons noted, this approach cannot accommodate obstructions in a room, and therefore has rather limited usefulness.

Hjertager and Magnussen (1977) investigated the effect of buoyancy forces, in the form of the Boussinesq approximation in the vertical momentum equation, in 3-D numerical solutions of room airflows. They also incorporated the k-ε turbulence model, and used a numerical approach introduced as the SIMPLE algorithm by Patankar and Spalding (1972). Comparisons with experiments show good agreement in the isothermal flows, whereas flows in which buoyancy effects were significant showed less consistent agreement. Haghighat et al. (1990) showed that the flow is significantly affected by buoyancy for corrected Archimedes numbers (Ar_c) greater than 75. They introduced two additional terms in the k-ε formulation to account for the buoyancy effects in addition to the usual Boussinesq terms, and achieved good agreement between computed and experimental Nusselt numbers. In connection with their study of natural convection flows in a cavity, Chen et al. (1990) showed that the buoyancy production terms in the k-ε formulation do not affect the velocity and temperature profiles directly, but significantly alter turbulence energy profiles.

Nielsen et al. (1979) also investigated the effect of buoyancy in ventilated rooms. They used a primitive variable approach in this 2-D code, incorporating the two-equation turbulence model proposed by Pope and Whitelaw (1976). Initial wall jet development assumptions, based on experiments, were used for boundary conditions, and turbulent wall functions were used to describe the near-wall regions in the vicinity of solid surfaces. They demonstrated the need to include buoyancy terms in the turbulence equations for Archimedes numbers greater than 10^{-3}. Their agreement is generally within 5% of the maximum velocity in the flow. The method was extended to three-dimensional flows by Gosman et al. (1980). Reinhartz and Renz (1984) showed that the flow in a vertical plane of the diffuser, in a rectangular room ventilated by the diffuser, is well predicted by the 2-D approach of Nielsen.

The general flow pattern and contaminant concentration were satisfactorily predicted for six different air supply configurations in downward flow clean rooms, by the 3-D code developed by Murakami et al. (1987). This code also used the k-ε model for turbulence transport. At low Reynolds number flows, agreement was not as good as at high Reynolds number flows.

The PHOENICS general purpose, commercial code developed by Rosten and Spalding (1981) has been used by Holmes (1982), Markatos (1983), Howarth (1985), Jones and O'Sullivan (1985), Whittle (1986) and Awbi and Setrak (1986 and 1987) to obtain velocity and temperature distributions in rooms. Qingyan and van der Kooi (1988) developed the ACCURACY code to determine appropriate boundary conditions, such as wall and inlet air temperatures, for the PHOENICS code. This combination was then used (Qingyan et al., 1988) to study the influence of the indoor airflow and temperature distribution on room energy consumption, and to evaluate ventilation efficiencies.

Several shortcomings are apparent in the work reported to date. The usual k-ε model with standard coefficients, obtained from high-speed jet experiments, is not well suited to low Reynolds number flows, such as characterize airflow in buildings. Furthermore, the codes are becoming more and more sophisticated; this may serve the purpose of researchers, but it means that it is becoming ever less likely that the practitioner will make use of them to provide better indoor air quality. Consequently, while some research is in order to adapt a suitable turbulence model to such computations with all the complexity tolerable in computer codes used on the largest machines available, there is also need for a more simplified approach to obtain useful information for the improvement of indoor air quality with a less complicated code which may be directly used by building engineers. This latter approach is the aim of the work reported in this paper.

Present Approach

The aim in the present work was to develop a solution technique to predict the ventilation airflow in a room or office. It was important, within this objective, to maintain simplicity as much as possible in order that the resulting technique may be more widely used than is likely to happen with a very complicated, sophisticated approach. The work reported here is from an isothermal, inviscid solution of the two-dimensional mass and momentum conservation equations, cast in cartesian form. This ensures that inertial effects of the flow are simulated, but the viscous/turbulence and buoyancy effects are not. The equations solved are the following:

$$\delta(\rho u)/\delta x + \delta(\rho v)/\delta y = 0 \quad (1)$$

$$\delta(\rho u^2)/\delta x + \delta(\rho uv)/\delta y = -\delta p/\rho x \quad (2)$$

$$\delta(\rho uv)/\delta x + \delta(\rho v^2)/\delta y = -\delta p/\delta y \quad (3)$$

An upwind solution scheme was used to solve the system of equations. Under-relaxed point and line Gauss-Seidel iterative methods were used, with the approach of the SIMPLE algorithm, as described by Patankar and Spalding (1972), used in the overall solution. More complete details of the approach used herein are given by Soultogiannis (1990).

The computational grid was arranged in such a way that the physical boundary of the ventilated space corresponded with the outer boundary of the cells nearest the walls. This permitted the inflow and outflow velocities to be specified along the approriate grid boundary. The solution points nearest the walls were located a half grid dimension away from the nearest wall. Except for the inflow and outflow locations, flow across the solid boundaries of the enclosure was not permitted. Convergence of the solution was accepted when the sum of the absolute mass residuals of all control volumes (grid cells) was less than 1% of the total inlet mass flow.

In the results shown herein, comparisons are first made with the results computed by the Exact3 code for a configuration studied experimentally by Timmons (1984). For this comparison, the Exact3 code (Kurabuchi et al., 1990), which is capable of 3-D simulations, is set up to simulate the two-dimensional case depicted by the present approach, corresponding to the experiments of Timmons. In the solution obtained with the Exact3 code, the k-ε turbulence model representation is used. This three-way comparison allows the present non-viscous approach to be compared with an approach using the k-ε model and with experimentally obtained data. Following this brief verification of the inviscid simulation approach, results obtained by it are presented for ventilation airflow in several office configurations containing partitions between workstations.

Results and Discussion

Code Verification

The code was tested by comparing results of its predictions with measured values from experiments by others, as well as with predictions made with the Exact3 code configured as a two-dimensional code, but using the k-ε turbulence model in its simulations. Figure 1 shows the velocity vectors for a solution of the flow through a room, as represented in the figure, which corresponds to the configuration used by Timmons et al. (1980) and Timmons (1984) in their experiments. Figure 1a shows the velocity vectors as obtained with the current non-viscous formulation, whereas Figure 1b shows the corresponding plot as obtained from the Exact3 (2-D) solution. The basic flow pattern predicted by the two methods is the same, with the counter-clockwise circulation in the room. The solution obtained with the Exact3 code used more cell divisions (each arrow represents the flow in one cell), and was able to predict more detail, showing small recirculation re-

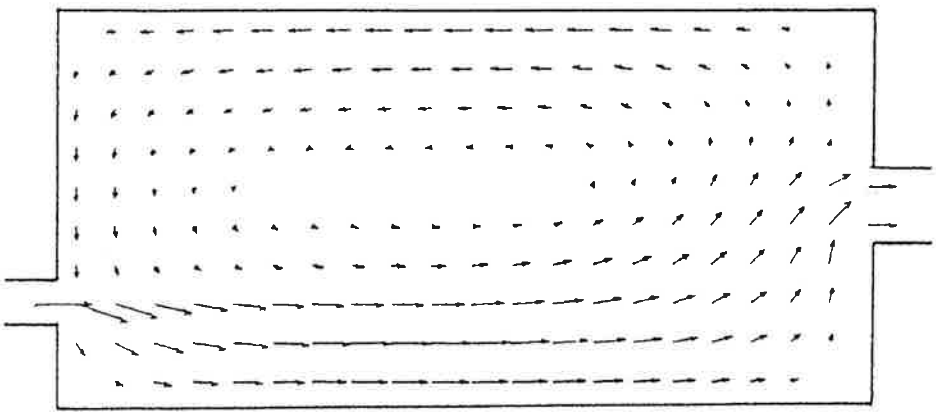

Fig. 1a Velocity vectors obtained for the configuration shown, at a Reynolds number of 18,000, as obtained with the proposed inviscid formulation

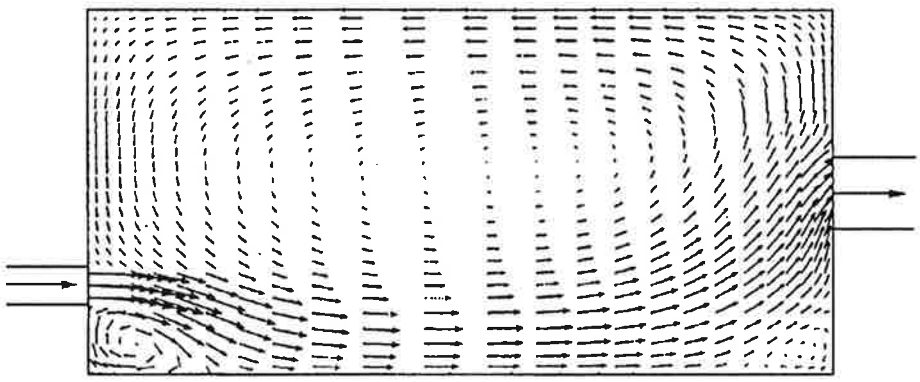

Fig. 1b Velocity vectors obtained for the configuration shown, at a Reynolds number of 18,000, as obtained with the Exact3 code using the k-ε turbulence model

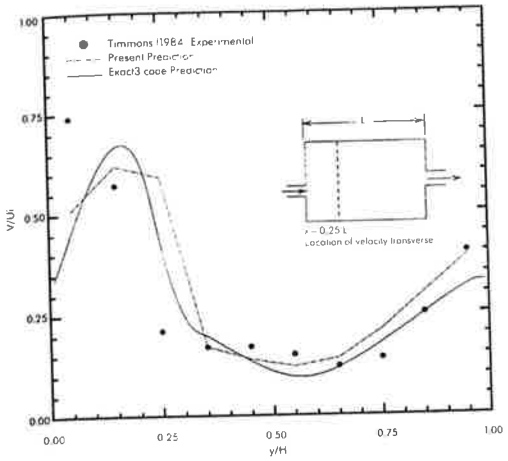

Fig. 2 Comparison between predicted and measured absolute velocity ratios, at the plane shown in the insert sketch, for a Reynolds number of 18,000, showing both the inviscid and the k-ε turbulence simulated results

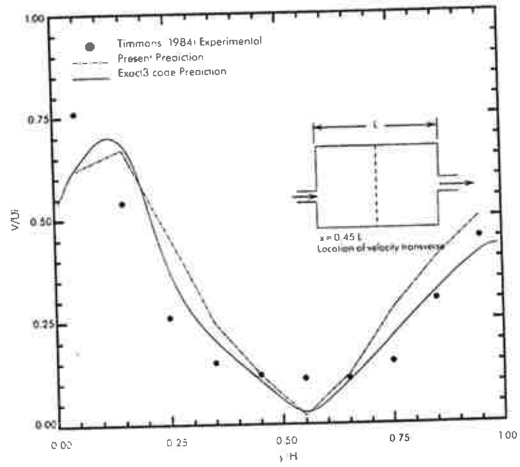

Fig. 3 Comparison between predicted and measured absolute velocity ratios, at the plane shown in the insert sketch, for a Reynolds number of 18,000, showing both the inviscid and the k-ε turbulence simulated results

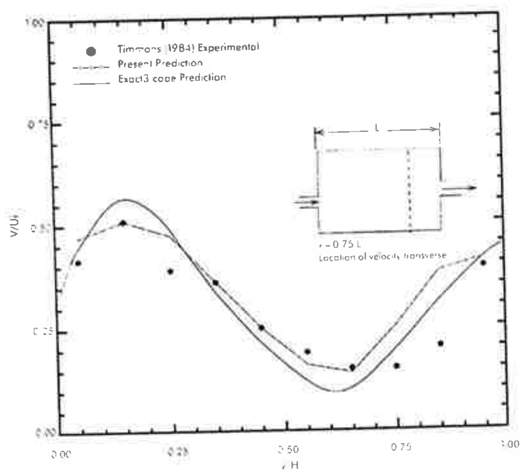

Fig. 4 Comparison between predicted and measured absolute velocity ratios, at the plane shown in the insert sketch, for a Reynolds number of 18,000, showing both the inviscid and the k-ε turbulence simulated results

gions in each bottom corner. More detail could similarly be expected from a finer grid using the inviscid formulation.

Figures 2, 3 and 4 show predicted velocities, at the selected locations shown for this configuration, compared with reported measured values by Timmons (1984). All of these results are obtained with Reynolds numbers (based on the inlet width) of 18,400. It may be noted that there is little to choose between the results of the two numerical simulations. They agree quite well with one another, and in each case show reasonably good comparisons with the experimentally obtained data.

The velocity vector diagram (Figure 1) shows that the computed flow is in the form of a recirculation in the vertical plane of the room; this is in qualitative agreement with flow patterns shown by Timmons et al. (1980) for a similar configuration and Reynolds number range. The cross-plots shown in Figures 2, 3 and 4 show only absolute velocities, and not the recirculation. Since none of the measured values fall below a normalized velocity of 0.1, and the flow must be in the pattern shown in Figure 1 for the absolute velocities to agree as well as they do in the cross-plots, it is quite reasonable to conclude that the measured values of normalized velocity of less than 0.2 have large error/uncertainty bars, relative to the magnitude of the velocities shown. If reasonable uncertainty bars are added to the measured values, it might be expected that the values of absolute velocity predicted by either of the numerical solution techniques would fall within the uncertainty range of the experimental measurements.

Figure 5 shows another cross-plot of measured and predicted velocities, in this case for the configuration and measured values as reported by Nielsen et al. (1979). It may be observed that the predictions, by both numerical simulation techniques, follow the trend as observed in measured velocities, but the Exact3 (2-D) predictions are noticably closer to the measured values of velocity, especially near the wall. For the inviscid simulation, the wall jet remains stronger than measured values show, and the gradient near the wall is much sharper than measured or simulated with the k-ε turbulence model included, as in the Exact3 code.

The major weakness of the inviscid approach is possibly the missing prediction of the decay of the wall jet. This effect arises due to the lack of viscous/turbulent dissipation in the computations, which is somewhat evident in the comparisons shown. In a modest sized room, in the Reynolds number range

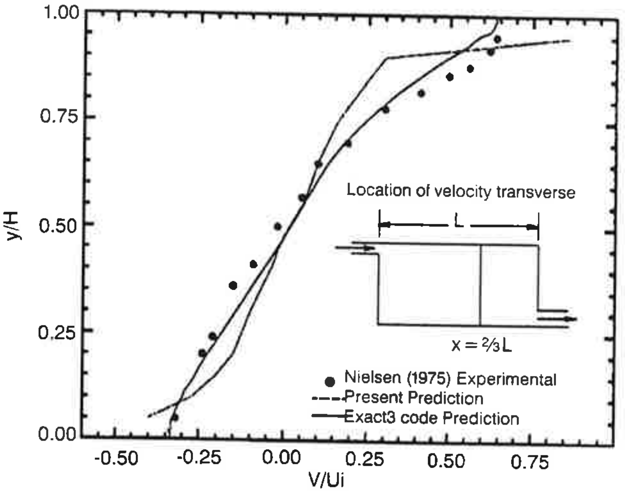

Fig. 5 Comparison between predicted and measured velocity ratios, at the plane shown in the insert sketch, for conditions as used by Nielsen (1975), showing both the inviscid and the k-ε turbulence simulated results

used for the results shown, this effect is not a major one. However, in larger rooms, experiments of Timmons (1984) show that the flow can break into two or more recirculation regions whereas the present inviscid code predicts a single recirculation region for even the longer rooms. It may be of further interest to point out that the Exact3 code, using the k-ε turbulence model, also failed to predict the break-up of the large recirculation eddy in the long room. At low Reynolds numbers (below 10^4), deviations from experimental results may also be expected since the viscous effects become more dominant in this range. However, codes containing sophisticated turbulence models (such as the k-ε two-equation model) using data obtained at high speeds, also have difficulty predicting the flow at low Reynolds numbers in cases such as these.

Application to Partitioned Rooms

As indicated earlier, the aim was to simulate the ventilation airflow in a room or office. A major point of interest in this application is the effect of office partitions on the flow of ventilation air in cases where an open office concept is being used, with a single row of ventilation air supply slots serving several workstations.

Figure 6 shows the velocity vectors for the base case, in which a room has ventilation air entering along one side of the ceiling and exhausting at the other side of the room, also through the ceiling. The main component of the flow is seen to follow the wall down to the floor, across the floor and up the other wall to the return, with a minor component of circulation through the space in general. For this case, the inlet velocity is 3 m/s, resulting in a Reynolds number of 7.7×10^4.

Figure 7 shows the velocity vectors for the same basic geometry as for Figure 6, except that there is a thin (1 cm thick) partition dividing the room into two workspaces. This overall room is 2.4 m high by 3.2 m long, with the partition extending 1.4 m upwards. With the partition set on the floor as illustrated here, a recirculation is set up on the side of the partition fed by the ventilation supply. Much of the airflow then sweeps over the top of the workspace in the second workstation, with only a minor circula-

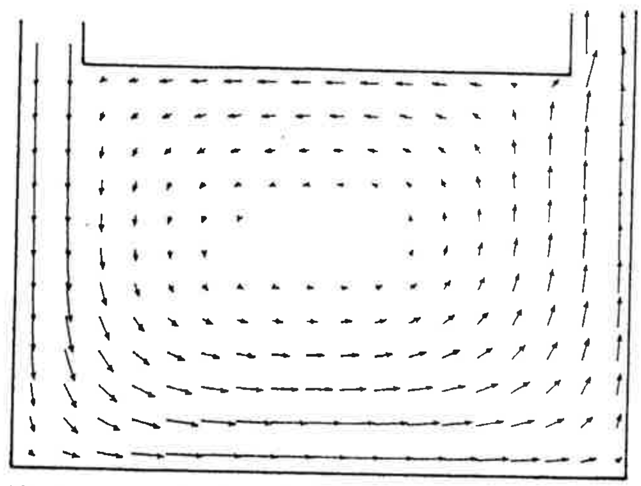

Fig. 6 Velocity vectors for flow entering from above and leaving via the ceiling, with no office partition, at a flow rate corresponding to a Reynolds number of 77,500, as simulated by the inviscid code

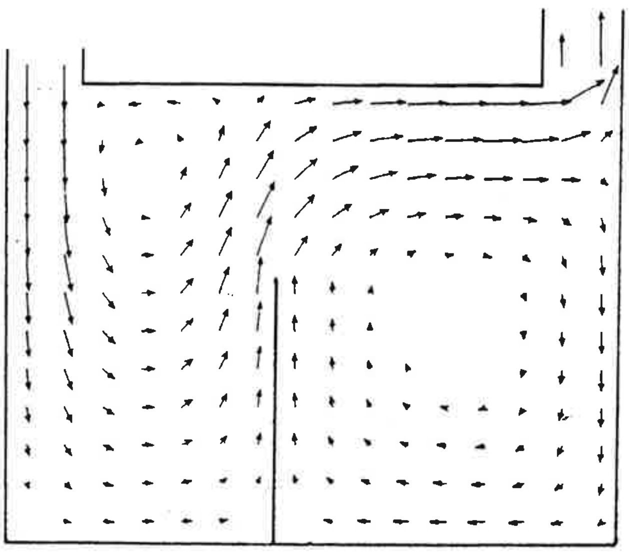

Fig. 7 Velocity vectors for flow entering from above and leaving via the ceiling, with a thin, 1.4 m high office partition set on the floor, at a flow rate corresponding to a Reynolds number of 130,000, as simulated by the inviscid code

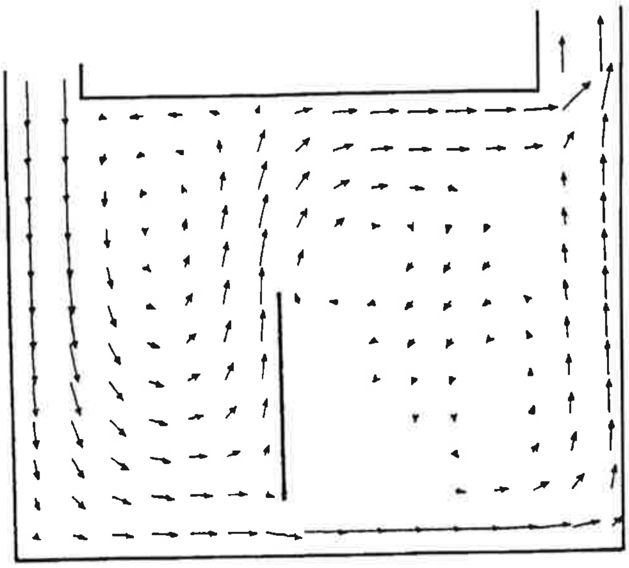

Fig. 8 Velocity vectors for flow entering from above and leaving via the ceiling, with a thin office partition extending from 0.3 m above the floor to 1.4 m above the floor, for a flow corresponding to a Reynolds number of 130,000, as simulated by the inviscid code

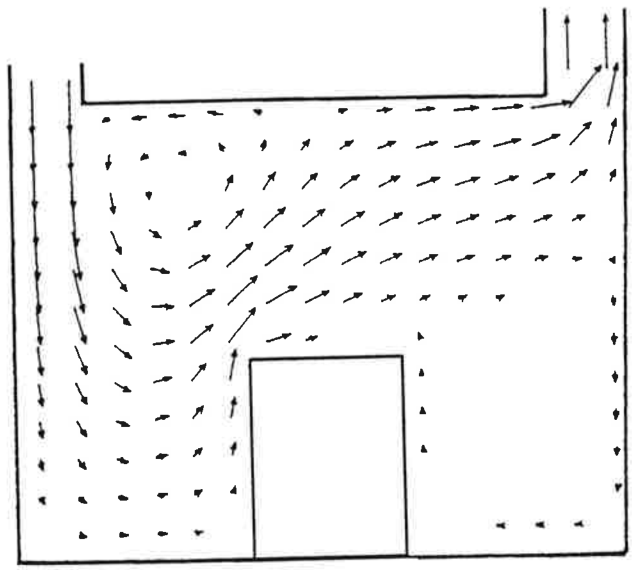

Fig. 9 Velocity vectors for flow entering from above and leaving via the ceiling, with a 0.8 m thick by 1 m high office partition set on the floor, for a flow corresponding to a Reynolds number of 130,000, as simulated by the inviscid code

tion pattern in the second workstation. If the top of the partition is maintained at 1.4 m above floor level, but the bottom 30 cm is eliminated, the flow vector diagram would be as shown in Figure 8. If the top of the partition is considered the upper boundary of the workstation, then the workstation on the supply side of the partition has 14% more airflow through it when the bottom of the partition is removed, as in Figure 8, and the second workstation experiences a 40% increase in air circulation as a result of eliminating the bottom of the partition, as compared with the flows for the configuration shown in Figure 7. When the partition extends to the floor, increasing the height or the thickness of the partition will reduce the air circulation to the second (downstream) workstation. For example, increasing the height of a 20 cm wide divider from 80 to 140 cm results in about 65% reduction in air circulation in the second workstation, with the supply and return airflows held fixed. The workstation on the supply side of the partition experiences only about 20% reduction in circulation by this change. A thick divider, as shown in Figure 9 (80 cm wide by 100 cm high), causes a very severe decrease in the air circulation through the second workspace.

Figures 10, 11 and 12 show the velocity vector plots for the case with two office partitions separating the supply and return slots. The case with both partitions down to the floor (Figure 10) shows strong circulation in the first workspace, but most of the airflow passes over the next two workspaces, with only minor circulation in these. When the first partition has some of the base eliminated (Figure 11) the flow through the second workspace is much improved, although the strong flow under the partition may be perceived as a source of draft by an occupant of the center workspace. The third workspace has slightly improved circulation for the case shown in Figure 11 as compared with the case of Figure 10. When the second divider is raised (Figure 12) the flow moves strongly under both partitions, causing a major improvement in the circulation in the third workspace, but reducing the air circulation in the second, except for the layer near the floor, through which the flow moves at a considerable rate.

On the basis of these observations, it would appear that using a single row of supply diffusers to serve several workspaces separated by a partition, whether the partition is set down to the floor, or raised above the floor, does not provide ideal air circulation. When partitions are placed between the supply and the return, a much higher flow rate of air is required to achieve circulation of the same magnitude as for the simple cases without partitions, and this leads to the likelihood of complaints about draft. It would be much more desirable to provide a supply and a return for each workstation, whenever possible, in order to be able to better control the airflow through each, independent of neighbouring workstations.

Conclusions

The results obtained with the isothermal, inviscid solution were in reasonably good agreement with measured values for simple configurations having no office partitions. On this basis, the results obtained with partitions may be considered to be a good indication of the true flow pattern, provided buoyancy effects are not significant. Experimental data with configurations of this nature are needed.

Based on simulated flow patterns in offices with several workstation partitions located between the ventilation supply and return, it is apparent that it would be difficult to ensure good air quality for each of the workstations created in this way. It would be preferable to have a separate supply and return for each workstation, thereby permitting some adjustment of the airflow to each workspace, without directly affecting the flow in neighbouring workspaces.

Acknowledgement

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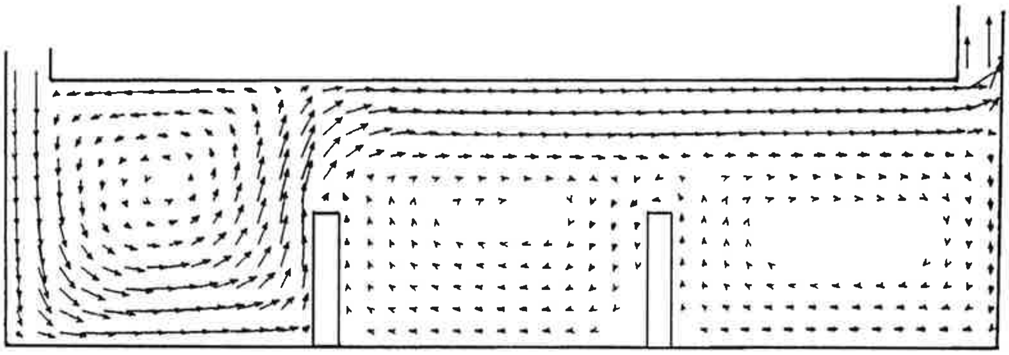

Fig. 10 Velocity vectors for flow entering from above and leaving via the ceiling, with two 1.4 m high by 20 cm thick office partitions set on the floor to divide the space equally, for flow corresponding to a Reynolds number of 260,000, as simulated by the inviscid code

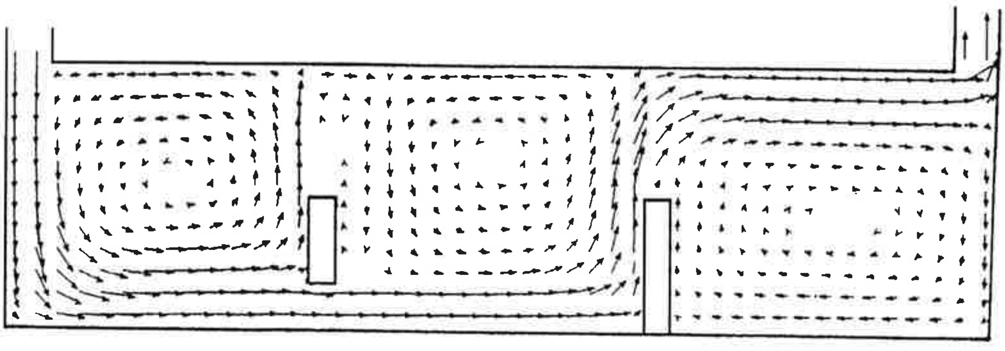

Fig. 11 Velocity vectors for conditions as for Figure 10, except that the first partition has 0.4 m removed from the bottom, with the top as before, as simulated by the inviscid code

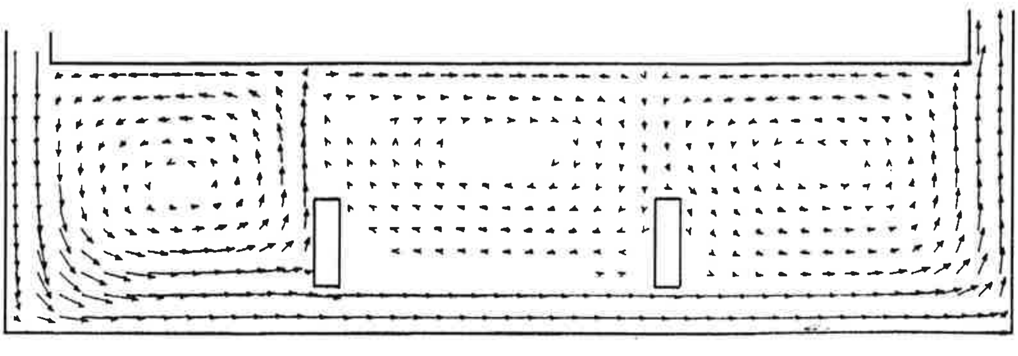

Fig. 12 Velocity vectors for conditions as for Figure 10, except that both partitions have 0.4 m removed from the bottom, with the top as before, as simulated by the inviscid code

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