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MODELLING SUPPLY DEVICES IN ORDER TO PREDICT IMPROVEMENTS IN INTERNAL AIR QUALITY

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1 Synopsis

The air distribution effects of floor mounted swirl diffusers are investigated and described in this paper. Results are based on a case study of an office typical of those in urban commercial environments. The effects of the swirl applied to the supply air as well as temperature differentials between supply and room air are explored. The investigation is restricted to situations where cooling is required.

The results of the work, which is undertaken by way of CFD analysis, are presented in terms of appropriate ventilation effectiveness parameters. It is observed that variations to the form of swirl devices, changes to temperature differentials and the presence of internal heat sources combine to affect the internal air distribution to such an extent that its nature may vary from displacement to characteristics approaching those of mixing systems. Air quality at head level can change from very good to poor in reponse to relatively small changes in some of the parameters.

2 List of Symbols

Symbol		Units
ε_a	air change efficiency	
ε_p	local air change index at point *p*	
τ_n	nominal time constant	s
$\bar{\tau}_p$	local mean age of air at point *p*	s
$<\bar{\tau}>$	room mean age	s

3 Introduction

It is often assumed that displacement systems are superior to mixing systems in providing good air quality in the occupied zone of a ventilated space. This is especially so in large spaces when there is a cooling requirement. In such cases, fresh air is normally supplied at floor level so that it displaces heat and pollutants upwards into the unoccupied space above. Such an arrangement helps to minimise the air flow rate, thereby saving energy. However, the distribution of fresh air in such a system, and the air quality in the occupied zone, are affected by a number of factors, including:-

(i) the discharge characteristics of the floor mounted diffusers;

(ii) the presence of any momentum generating sources within the space, especially heat sources;

(iii) obstacles at or near ground level.

There are two types of floor mounted diffuser in common use in displacement systems. The tower type consists of a perforated cylinder, typically one metre high, supplied by ductwork, and located around the perimeter of the space and at other convenient points. Exit air velocities are of the order of 0.25 ms^{-1}. The flush type consists of circular grilles arranged on a grid over the whole floor area, and most often supplied by an under-floor plenum. Exit air velocities are of the order of 0.9 ms^{-1}, and these diffusers are often provided with swirl vanes in order to enhance their ability to create a plane of upward moving air. Indeed, swirl supply terminals are frequently specified for the provision of mechanical ventilation and, depending on the terminal exit angle and temperature difference, the characteristics of the resulting air distribution may approximate to either displacement or mixing ventilation. Neither type of diffuser can produce a perfect displacement air flow; this would require air to be supplied evenly over the whole floor plane, as is the case in laminar flow clean rooms.

Because the velocity and momentum of the air at entry are low, the second and third factors are relatively more important in displacement systems than in mixing systems. This became apparent in a recent study of the application of ventilation effectiveness parameters to some design studies [1]. The present study, therefore, examines the effect on air quality of the flow characteristics of the flush mounted floor diffusers with varying degrees of swirl, and the presence of heat sources. This is done by studying a simplified but typical design problem, and using the appropriate ventilation effectiveness parameter to evaluate air quality.

4 The Design Case and the Method of Assessment

The internal space on which the study is based is an office located in a typical urban environment. The office, which is shown in Figure 1, is 4.8m by 3m in plan and 3.2m high. One of the 3m x 3.2m walls is an external wall with a single glazed window, which extends the full 3m width of the room and vertically 2.2m from a 1.0m high cill. All other walls, the floor and ceiling are assumed to be backed by similar spaces at the same temperature. The office has two occupants and two PC's as heat sources. Ventilation is provided by six 150mm diameter floor mounted swirl diffusers and these are balanced by two ceiling mounted extract units. The dimensions and performance data for the diffusers were chosen to be typical of commercially available units. The ability of the system to produce good air quality is assessed by determining velocity vectors, temperatures and ventilation effectiveness throughout the space. The most appropriate ventilation effectiveness parameter in this situation was considered to be the Local Air Change Index (LACI, see appendix A for definitions), as contours of this parameter give a more realistic picture of the distribution of fresh air than either of the other parameters.

The discharge characteristics of the diffusers were investigated by testing the effect of different degrees of swirl. Rather than associate a swirl number with the diffuser, it was found to be more convenient for modelling purposes to specify a discharge angle. An angle of zero is taken to be the case where the air leaves the diffuser vertically, perpendicularly to the floor surface, in which case it may be expected to behave as a simple circular jet with zero swirl.

The effect of varying the temperature differential of the air at the inlet had, of necessity to be linked to adjustments in the external temperature, as these two factors interact. Each combination of factors was arranged to give approximately the same average space temperature.

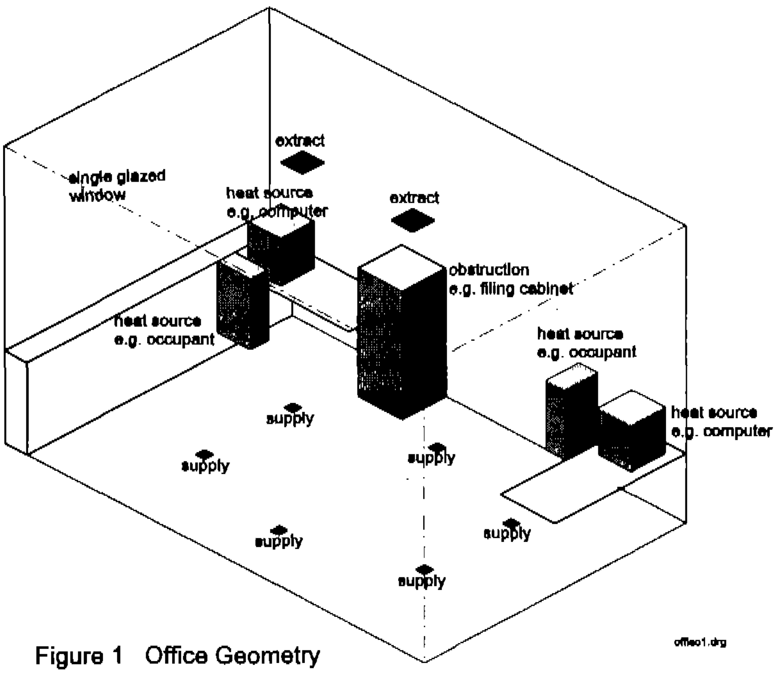

Figure 1 Office Geometry

5 Modelling Procedure

The problem was treated by means of Computational Fluid Dynamics, using the 'Flovent' CFD model by Flomerics. This provided the velocity vectors and temperature contours. The LACI contours were obtained by post-processing the Flovent solution, as has been previously reported [2], except that the post-processing software has now been rewritten in order to remove restrictions on the size of the model, and to compute contaminant removal effectiveness as well as air change efficiency parameters. The computed parameters are returned to Flovent for subsequent analysis and display.

The method of modelling of the diffusers required careful consideration. 'Flovent' is restricted to a rectangular Cartesian grid, whereas the diffusers are circular, and the addition of swirl imparts a circular rotational pattern to the jet. This problem was overcome by dividing the equivalent square representing the diffuser into four quarter segments through which air was supplied vertically upwards. A two dimensional horizontal thrust was then applied to the air as it left the diffuser in order that the resultant was directed at 45° to the co-ordinate axes in plan and at the required angle to the vertical, and with the required discharge velocity of $0.9ms^{-1}$. This created the desired type of jet pattern, as is shown in Figure 2 by the velocity vectors on a horizontal plane 0.125m above floor level for a typical configuration.

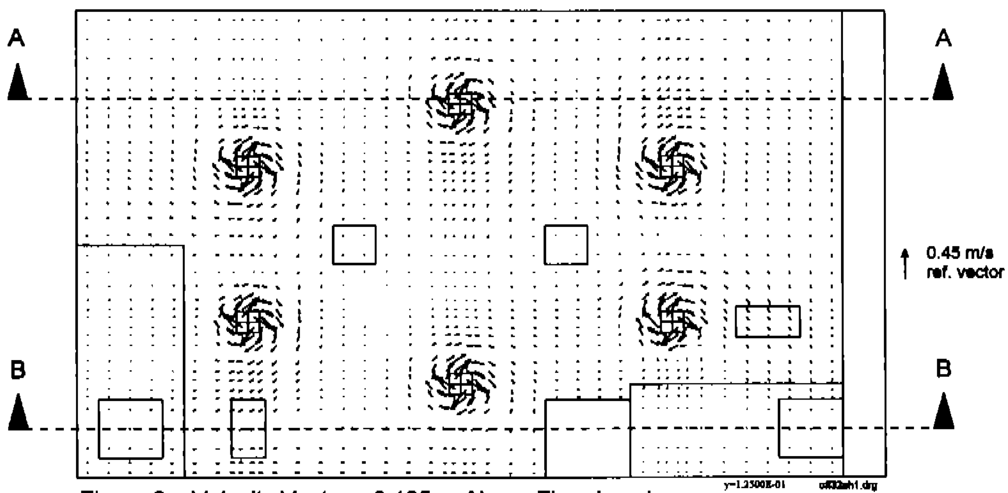

Figure 2 Velocity Vectors 0.125m. Above Floor Level
(Case 1: Supply at 60 Deg. from Vertical,)

For the purposes of this study, analysis has been restricted to a flow rate of $0.00952kgs^{-1}$ through each of the 6 floor mounted supply terminals, corresponding to a nominal time constant of 16 minutes, or 3.8 air changes per hour. The consequences of a variety of swirl angles, external ambient temperatures and supply air temperatures have been explored. Table 1 summarises the combinations which have been tested and are reported here. All are cooling cases with 180W of internal gains.

Table 1 Summary of Test Cases

Case no.	Diffuser discharge angle	Condition	Internal heat gains	External ambient temperature	Supply air temperature	Results plotted in figure nos.	Air Change Efficiency %
1	60°	Cooling	180W	25°C	19°C	3,4,5	77.8
2	40°	Cooling	180W	25°C	19°C	6	58.7
3	20°	Cooling	180W	25°C	19°C	7,8	51.7
4	60°	Cooling	180W	23°C	20°C	9	51.9
5	60°	Cooling	180W	21°C	21°C	10	53.9

6 Consideration of Results

The air distribution pattern from the diffusers is determined by three competing effects:

(i) the inertial force due to the jet exit velocity, tending to produce a vertical plume,
(ii) the buoyancy force due to the temperature difference, causing the air (in all the cooling cases) to form a layer at floor level, and
(iii) the coanda effect, also pulling the air towards the floor surface.

The addition of swirl is expected to reduce the effect of the first of these, and to help to create a displacement flow. As the swirl angle decreases, there may be a point at which the flow switches from a stratified displacement pattern to a predominantly vertical jet. However, entrainment, recirculation, internal heat sources, and internal room geometry all complicate the situation. The results for each of the cases given in Table 1 have been plotted to show contours of LACI, as this gives the distribution of fresh air. Referring to figure 2, the vertical sections are plotted either on plane A-A where there is one diffuser and no disturbing features, or on plane B-B through one of the heat sources. Table 1 also gives the air change efficiency for each case. It should be noted that an air change efficiency of 100% correponds to perfect displacement ventilation, whereas 50% corresponds to perfect mixing ventilation.

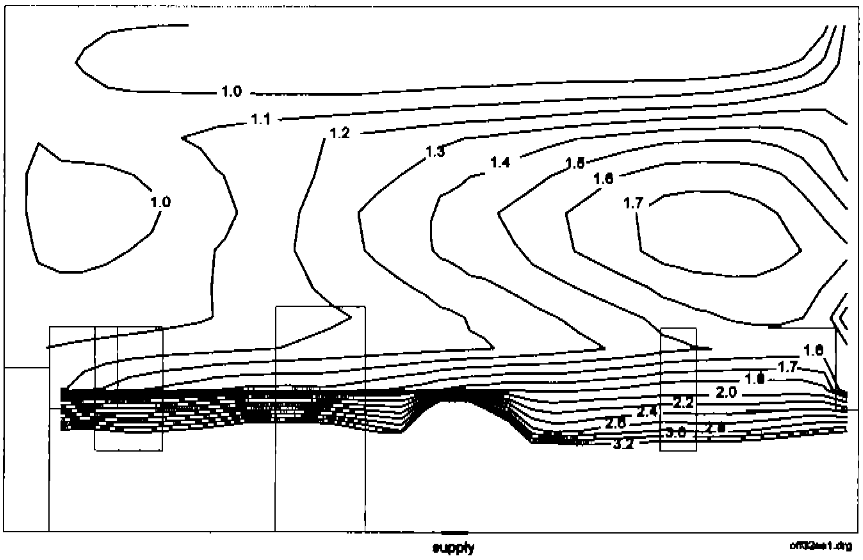

Figure 3 Case 1, L.A.C.I. Contours on Plane A-A.
(Supply at 60 Deg. from Vertical. Ambient 25 Deg.C, Suppl19 Deg.C.)

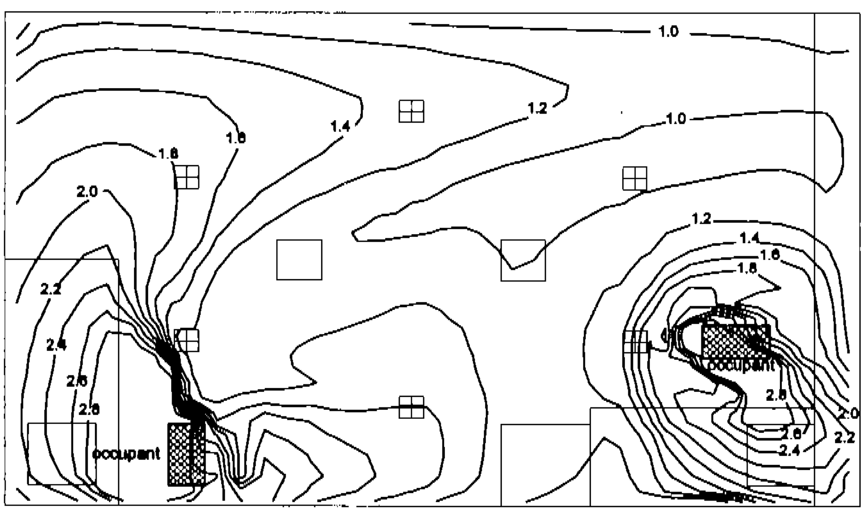

Figure 4 Case 1, L.A.C.I. Contours1.5m Above Floor Level.
(Supply at 60 Deg. from Vertical. Ambient 25 Deg.C, Supply 19 Deg.C.)

Case 1 - swirl angle 60° - external temperature 25 °C, supply temperature 19 °C.
Figure 3, a section at A-A, shows a clear displacement pattern. Incoming air is trapped at floor level, with substantially horizontal LACI contours decreasing in value with height. Figure 4 shows the LACI at 1.5m above floor level, equivalent to head height for a seated person, with

values greater than 1 over most of the plane. It also shows islands of much higher values around the two heat sources (occupants), showing that the plumes induced by these sources are sucking fresh upwards from below. The vertical section at B-B in Figure 5 shows one of the plumes more clearly, but also shows that it induces a slight recirculation which brings some old air back down to lower levels. Despite this it will be seen from Figures 3, 4 and 5 that high values of LACI extend over much of the volume. The air change efficiency is very high at 77.8% indicating a successful displacement ventilation system.

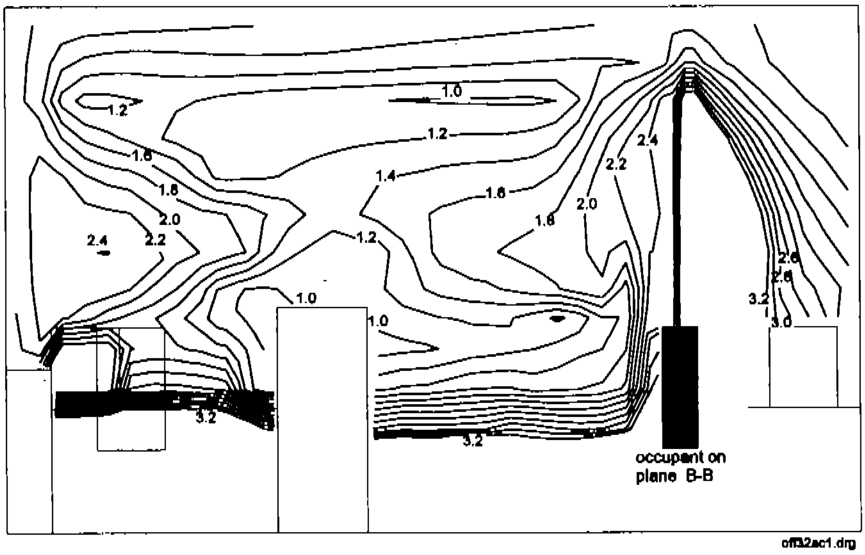

Figure 5 Case 1, L.A.C.I Contours on Plane B-B
(Supply at 60 Deg. from Vertical. Ambient 25 Deg.C, Supply 19 Deg.C.)

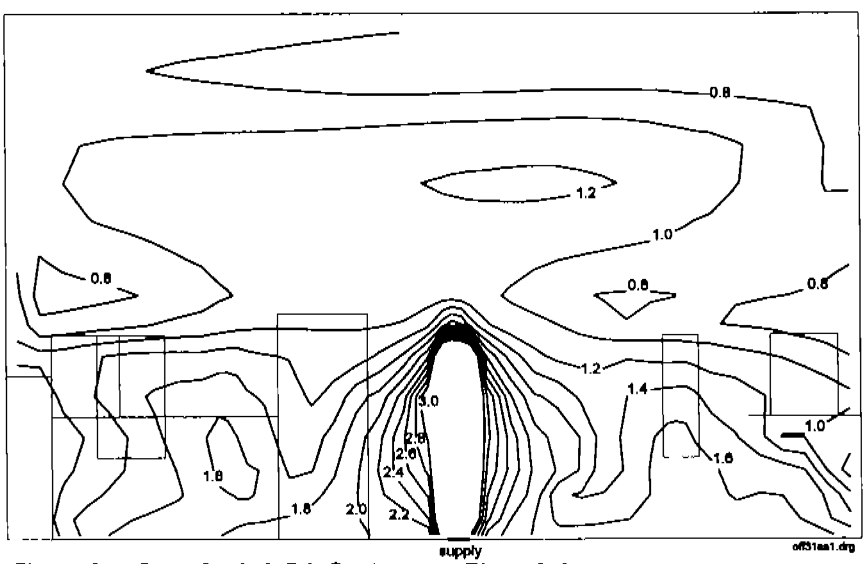

Figure 6 Case 2, L.A.C.I. Contours on Plane A-A.
(Supply at 40 Deg. from Vertical. Ambient 25 Deg.C, Supply 19 Deg.C)

Cases 2 and 3 - swirl angles 40° and 20° - external temperature 25 °C, supply temperature 19 °C.

Figure 6 shows that at a swirl angle of 40° the air enters as a jet before stratifying at about head height. The LACI contours now have a vertical bias, with high values clustered around the jet and much lower values at head level. Figure 7 for a swirl angle of 20° is very similar, suggesting that the expected transition has taken place between 60° and 40°. Figure 8, which is a horizontal section at head height for the 20° case, is significantly different to Figure 4 for 60°, and shows islands of high LACI above the diffusers.

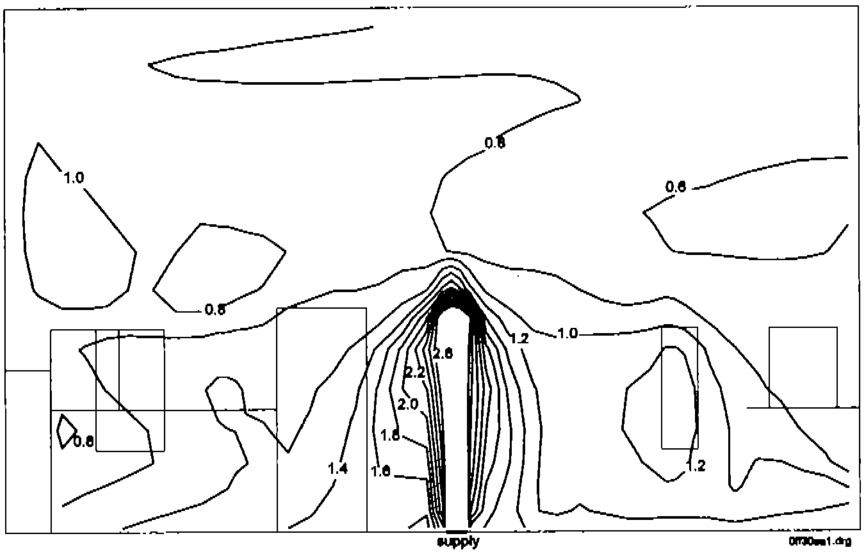

Figure 7 Case 3, L.A.C.I. Contours on Plane A-A.
(Supply at 20 deg. from Vertical. Ambient 25 Deg.C, Supply 19 Deg.C.

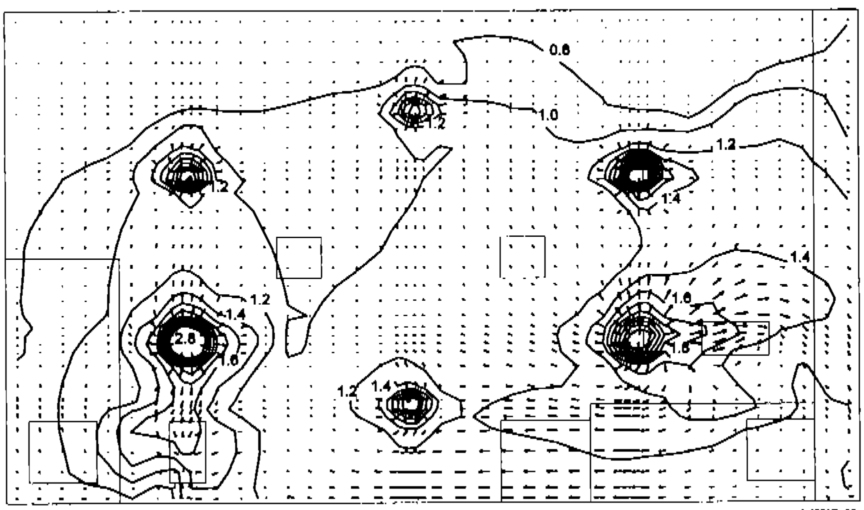

Figure 8 Case 3, L.A.C.I Contours and Velocity Vectors 1.5m Above Floor Level
(Supply at 20 Deg. from Vertical. Ambient 25 Deg.C, Supply 19 Deg. C.)

The air change efficiency values of 58.7% and 51.7% for Cases 2 and 3 respectively are considerably less than for Case 1, and when considered in conjunction with the LACI distribution are indicative of a mixing system rather than a displacement sytem.

Figure 9 shows a section at B-B for Case 3 and, like Figure 5, it shows a vertical plume of air convected upwards. However, in this case, the LACI at the breathing zone is only approximately 1.4 as opposed to a value in excess of 3.

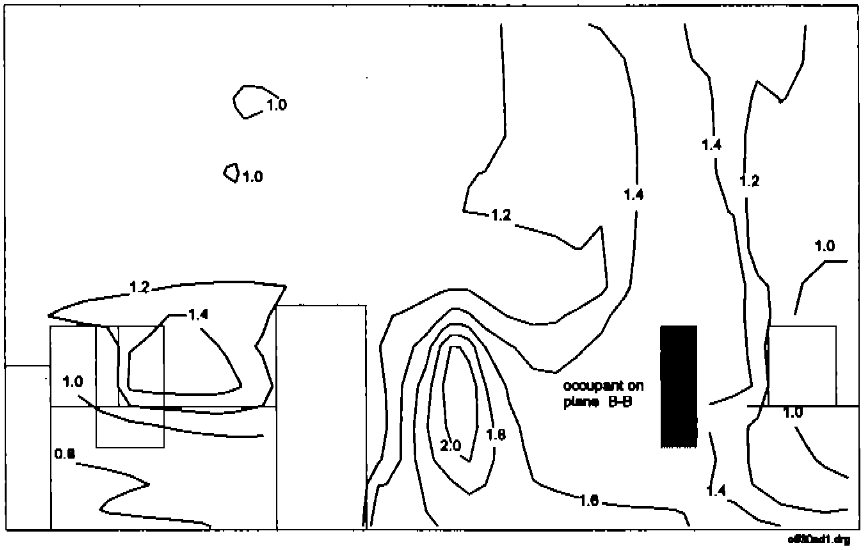

Figure 9 Case 3, L.A.C.I. Contours on Plane B-B
(Supply at 20 Deg. from Vertical. Ambient 25 Deg.C, Supply 19 Deg. C.)

Case 4 - swirl angle 60° - external temperature 23 °C, supply temperature 20 °C.
The distinctive displacement pattern of Figure 3 is partly due to the negative buoyancy of the 6°C external to internal temperature difference. Reducing this to 3°C, as is shown in Figure 10, compresses the contours downwards with values of considerably less than 1 at head level, which indicates poor air quality and is the opposite of what might be expected. Stratification is much more apparent than in Case 1 (Figure 3) and a consequence of the large volume of air with an LACI of less than 1 is that the air change efficiency of this case at 51.9% is much less than that of Case 1. It appears that the plumes of warm air over the heat sources are pulling a large quantity of fresh air to high level, effectively short-circuiting the rest of the space.

Case 5 - swirl angle 60° - external temperature 21 °C, supply temperature 21 °C.
With a temperature differential of zero, the only buoyancy forces are those due to the internal heat sources. It will be seen from Figure 11 that the result is similar to Case 4 although the contour lines of low LACI representing old air are less compressed. Again, it is probable that the heat sources are providing a convective route for fresh air to high level resulting in large volumes of old air near the centre of the room and so leading to a relatively low air change efficiency of 53.9%.

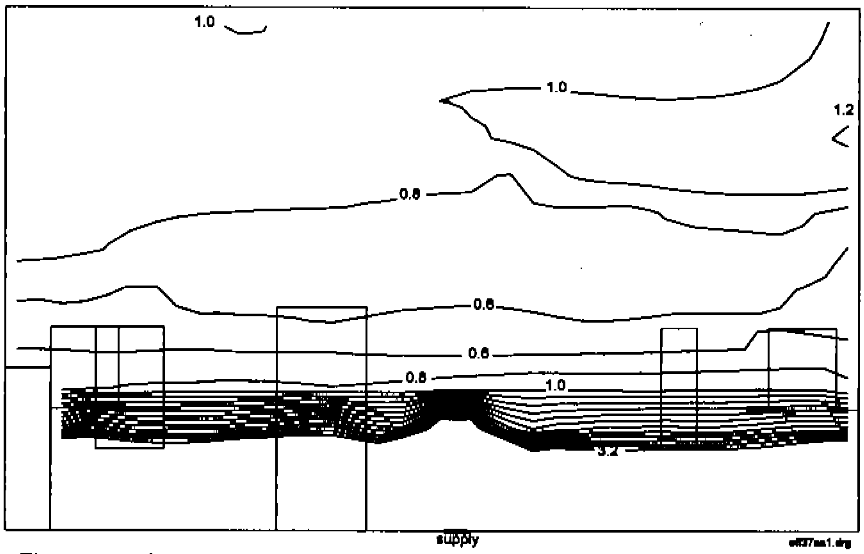

Figure 10 Case 4, L.A.C.I. Contours on Plane A-A
(Supply at 60 Deg. from Vertical. Ambient 23 Deg.C, Supply 20 Deg.C.)

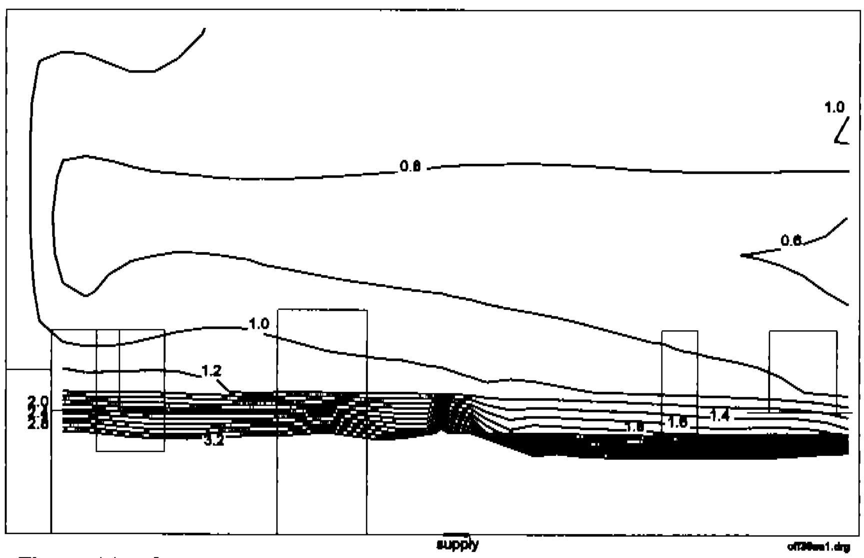

Figure 11 Case 5, L. A. C. I. Contours on Plane A-A.
(Supply at 60 Deg. from Vertical. Ambient 21 Deg.C, Supply 21 Deg.C.)

7 Conclusions

The first conclusion is that the addition of swirl to floor mounted diffusers enables them to create a true displacement ventilation pattern, provided that the amount of swirl is sufficient. Comparison of cases 1,2 and 3 show that in this particular example, a swirl angle of at least 60° to the vertical is necessary to overcome the inertial force of the jet.

The second conclusion is that even relatively low power heat sources can have a significant effect on the movement pattern of air from low to high level. This is because the pool of fresh air at low level has very little momentum. The plume which is set up over the heat source contains most of the fresh (i.e. 'young') air. This is most noticeable in Case 4, where the LACI has high values within the plume, but low values elsewhere. If the plume is due to an occupant, then it can be argued that the occupant is effectively bathed in a plume of high quality air. This may be a particularly fortunate characteristic of displacement ventilation. However, if the heat is due to some other type of source, the net effect could be for the occupant to find himself breathing the poor quality air at head level in other parts of the space.

The overall conclusion of this particular case study is that a system which is designed to provide a displacement pattern may sometimes behave quite differently, with a dramatic change in the fresh air quality at head level.

8 Appendix - Definition of Ventilation Effectiveness Parameters

i Local Mean Age of Air, $\bar{\tau}_p = \int_0^\infty t.A_p(t).dt$ where $A_p(t)$ is the age distribution curve for air arriving at point p.

ii Room Mean Age, $<\bar{\tau}>$ The average value of the local mean ages of air for all points in a room.

iii Local Air Change Index (LACI), $\varepsilon_p = \frac{\tau_n}{\bar{\tau}_p}$ where τ_n is the nominal time constant of the room. The nominal time constant is the reciprocal of the fresh air change rate. A value of LACI greater than 1 indicates that a point is receiving air more efficiently than it would with a perfect mixing system.

iv Air Change Efficency, $\varepsilon_a = \left(\frac{\tau_n}{2<\bar{\tau}>}\right)100\%$

9 References

1 Simons M W and Waters J R The Effects on Ventilation Parameters of Various Ventilation Strategies AIVC 17th Conference Proceedings pp 457-466 (1996)

2 Simons M W and Waters J R The role of Ventilation Effectiveness Parameters in the Design of Air Distribution Systems Building Services Engineering Research & Technology 19(3) PAGES???? (1998)