Displacement ventilation: a design guide

#3447

Paul Appleby explains how to design a system of air movement which makes full use of convection currents in the room to be ventilated.

It has long been recognised that the effectiveness of any ventilation or air conditioning system depends largely on the method of air supply and removal. To quote CIBSE (1986b): "The usual results of poor air terminal selection and/or positioning are draughts, stagnation, poor air purity, large temperature gradients and unwanted noise". A combination of which can render the system uncontrollable, the internal climate uncomfortable and the occupants inconsolable.

Many of these problems occur because of the compromises imposed on the building services designer by physical and aesthetic constraints. Apart from room thermostats, the air terminal devices are usually the only part of the system which the building occupants see. In every way they are the main interface between the system and the user — this is, of course, why they are so important.

Many modern buildings experience a net heat gain across most of their floor area, all year round. Heat rises, hence the bulk of the air movement in these buildings is in an upwards direction. This upward movement of room air can be very strong, if there are large numbers of computers running, for example, and the traditional method of supplying air via ceiling diffusers and removing it through openings, also in the ceiling, can lead to downdraught when horizontal jets are displaced downwards by strong upward air currents. Furthermore some of the air supplied across a ceiling may very well find its way into the extract openings without fulfilling its potential in heat transfer or contaminant dilution.

Air supplied at low level and extracted at high level travels upwards with the convection currents. Downward airflow into the occupied zone will only occur if there is not enough air supplied into the zone to replace the volume of air rising out of the occupied zone, or air next to a cold surface, such as a window, is cooled and hence falls into the occupied zone (see figure 1).

Terminology

The term displacement ventilation is used in the title of this article because it is a commonly used term. It is used to describe a system by which air is delivered to a room at a low velocity and close to the floor and rises through the room due to a combination of forces, but predominantly due to the convection currents rising from warm surfaces (figure 1).

However the term displacement ventilation can be used to describe a plug of air moving downwards, upwards or across the full cross section of a room, pushing contaminated air ahead of it. Here we will use the more accurate term buoyancy-assisted mechanical ventilation which gives due weight to the main driving force, buoyancy. The phenomenon (rather than the system) can be described by the phrase buoyancy-assisted room air diffusion (rad).

Air terminal devices

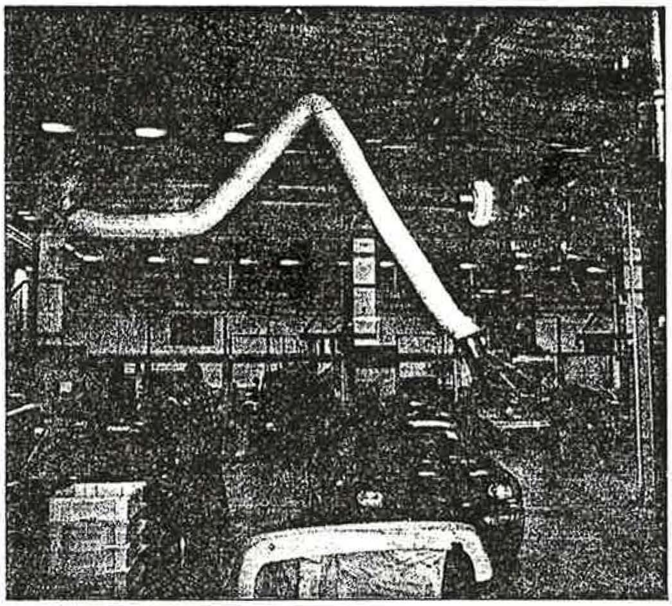
A number of manufacturers produce supply air terminal devices (atds) which can be used with buoyancy-assisted mechanical ventilation or air conditioning. There are essentially two approaches:

□ supply air at or near floor level at a low velocity, via a low velocity supply atd;

□ supply air in the form of rapidly diffusing jets via floor, desk or seat-back atds.

The former approach is one which has been widely adopted in the Scandinavian countries mainly through the efforts of manufacturers such as Stratos (formerly Bahco), Stifab (marketed as Repus by Ventilation Jones in the UK), Farex and Halton. Floor, desk and seat-back outlets were developed primarily in Germany by the manufacturer Krantz, which has recently been joined in this expanding market by others, such as Trox and Waterloo.

There have been three generations of low velocity supply atds, examples of each are still manufactured. The first generation allows air to leave predominantly at right angles to the face, which is either flat, corrugated or curved through 90 deg, 180 deg or 360 deg.



Above: A repus low velocity atd (1st generation) at the Porsche workshop, Reading.

Faces are either perforated plates or filter media held in a mesh. Velocity distribution across the face is equalised by internal reducing filters, plates or deflecting nozzles. Second generation devices are flat faced and designed for recessing into walls, internal deflectors provide a relatively even distribution through 180 deg. Third generation devices incorporate means for inducing some room air to mix with primary air within the device.

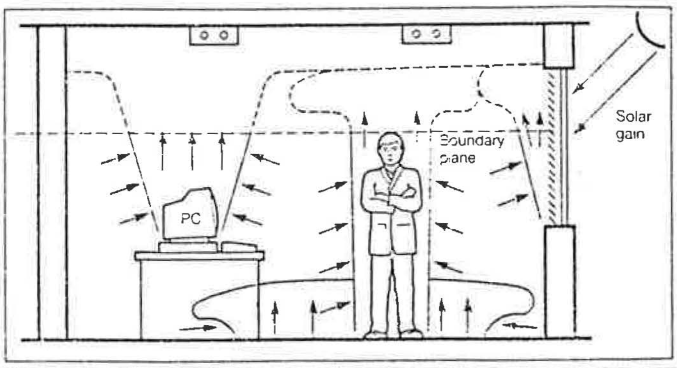
Figure 2 shows the respective air velocity profiles for first and second generation low velocity supply air terminal devices (lv supply atds) measured close to the device.

Figures 3 and 4 show a typical rapid-diffusion floor-mounted atd and its velocity profiles. Rapid diffusion is achieved due to the highly turbulent nature of the multiple jets, which have a swirl imparted to them due to the internal geometry of the device.

Design procedure

It is not intended here to detail all the alternative methods of air supply. However the choice between high or low level supply has a major impact on many aspects of the design process, not just of the ventilation or air conditioning system but also of the building and its internal organisation. The decision

Figure 1: Buoyancy assisted room air diffusion, heat sources and plumes in a typical office.



is complex but will usually be based on a combination of:
☐ economics: ie total cost to user;
☐ aesthetics;
☐ space: use of false ceiling, floor void, wall recesses or floor area;
☐ comfort: potential for draught or excessive temperature gradient;
☐ hygiene: mixing versus displacement - is there a contamination problem?
☐ potential for heat recovery;
☐ how are heat losses (if any) to be catered for?

The approach to design depends on whether the dominant need is for contaminant or temperature control. In applications where both are important it is usually necessary to determine air flow rates for both and use the highest value. Contaminant control usually dominates in an industrial process ventilation problem whilst temperature control is usually the most important requirement in comfort air conditioning.

Room convection fields

The main driving force in the majority of applications will be the convection currents generated by the heat sources in the room. Any surface which is heated to a temperature higher than that of the adjacent air will set up a rising convection plume.

The strength of the plume will depend on the temperature of the source and it will continue to rise and expand, entraining the cooler surrounding room air, until it either reaches the same temperature as its surroundings, meets an equal or greater and opposite force, or an impenetrable surface, such as a ceiling. The plume expands because it is fed with surrounding air continuously as it rises.

A domestic fire must be supplied with a sufficient flow of outside air to prevent it from smoking. Similarly the heat sources in the occupied zone must be fed with enough air from the low level atds or else some of the rising plume will recirculate back into the occupied zone, carrying some contaminants back with it. This may not be too much of a problem if contamination emissions are relatively low. The resultant contamination concentrations could never be greater than with a high level supply system with complete mixing.

Figure 1 shows common room heat sources and their associated convective plumes. The boundary plane is the level at which the net upward airflow is equal to that supplied into the occupied zone.

If recirculation of contaminants which have entered the zone above the occupants' heads is to be avoided then the supply flow rate should be determined by calculating the net flow rate of the convective plumes as they pass through the boundary plane.

This can be determined by subtracting downflow from cold surfaces from the plumes from warm surfaces as they pass through the boundary plane. Although it is hoped that downward air-flows will be minimal since they will carry contaminants from the upper zone with them, as observed by Sandberg (1989).

For this reason buoyancy-assisted systems are only suitable for well insulated or internal spaces. Heat losses must be dealt with by locating heat emitters under windows, well away from the atds. Sandberg also found that downflow into the occupied zone occurs when a door is opened into a small room.

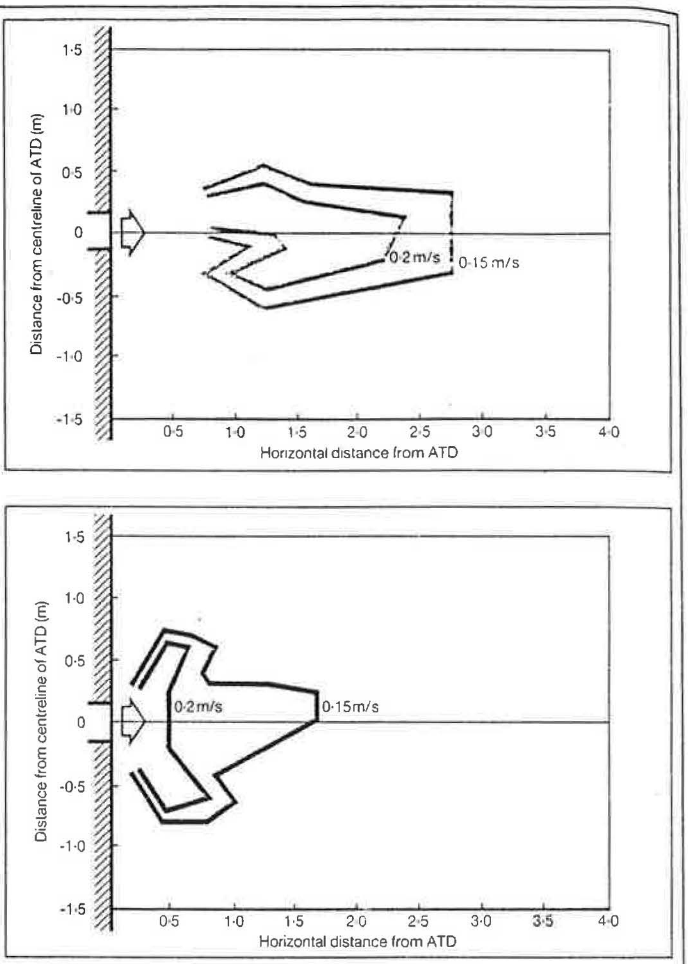
The boundary plane should be as close to the floor as possible: the higher it is taken the greater the plumes will have expanded and hence the greater the air volume to be supplied into the occupied zone.

If most occupants are likely to be seated for most of the day then the plane can be taken at 1·2 to 1·4 m, whereas if standing occupants are important 1·7 to 1·8 m is a reasonable height for it.

To calculate the plume volume flow rate at a height x, above an object, we can use the formula for a point source. From equations given by Kofoed (1988) based on early work by W Schmidt (assuming that the surrounding air is at a mean temperature of 22°C):

$$Q = 0{\cdot}005\,H^{1/3}(x+xp)^{5/3} \qquad (1)$$

where Q = volume flow rate of the plume at a height x (metres) above the source (m³/s) and H = heat output from source (W). xp = the vertical distance from the top of the object to the eqivalent point source, taken as



From top, figures 2a & 2b: Typical velocity contours at 100 mm from the floor for 1st and 2nd generation lv atds (adapted from Skistad, 1988).

equal to the eqivalent diameter of the object considered as a cylinder.

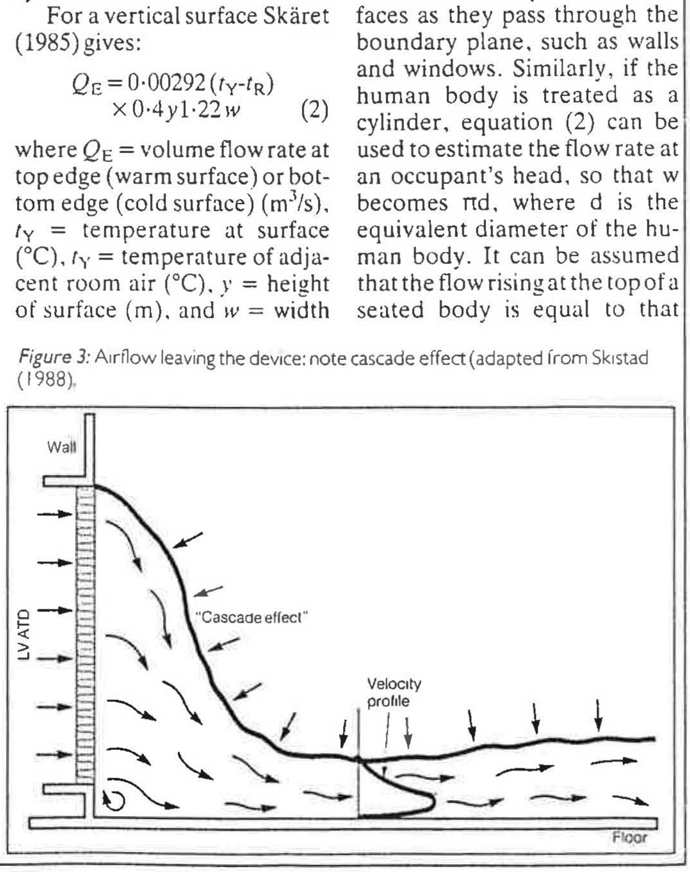
For a vertical surface Skäret (1985) gives:

$$Q_E = 0{\cdot}00292\,(t_Y - t_R) \times 0{\cdot}4y1{\cdot}22w \qquad (2)$$

where Q_E = volume flow rate at top edge (warm surface) or bottom edge (cold surface) (m³/s), t_Y = temperature at surface (°C), t_Y = temperature of adjacent room air (°C), y = height of surface (m), and w = width of surface (m).

This equation can be used to calculate flows up or down surfaces as they pass through the boundary plane, such as walls and windows. Similarly, if the human body is treated as a cylinder, equation (2) can be used to estimate the flow rate at an occupant's head, so that w becomes πd, where d is the equivalent diameter of the human body. It can be assumed that the flow rising at the top of a seated body is equal to that

Figure 3: Airflow leaving the device: note cascade effect (adapted from Skistad (1988).



rising from the same body standing.

For the heat sources shown in figure 1, with a boundary plane 1·7 m from the floor, equation (3) can be used to evaluate the plume flow for a body having a diameter of 0·3m, 1·7m tall and which is 8 K warmer than the mean surrounding air temperature.

This gives a volume flow leaving the top of the head of 0·012 m^3/s. It is interesting to compare this with the recommended UK fresh air requirements (CIBSE 1986a) for odour dilution of 0·008 m^3/s per person.

The pc can be considered as a cube having 0·4 m long sides with a heat emission of 300 W and its upper surface 0·6 m below the boundary plane. Using equation (1) the plume volume flow at 1·7 m from the floor is 0·024 m^3/s.

The window shown in figure 1 is being heated to 12 K above room temperature by the sun. If 0·8 m of the window height is below the boundary plane and it is 1·5 m wide, equation (3) gives a vertical flow rate at the boundary plane of 0·006 m^3/s.

The convective heat gain from heat sources above the boundary plane, such as ceiling mounted luminaires, will not influence the air flow between the zones.

However, radiant heat from the luminaires, the sun and other sources will heat the floor and walls and establish local convective flows. For example part or all of the floor may be heated and a very weak but broad convective flow established.

This plume is likely to reach thermal equilibrium with the surrounding air well below the boundary plane and has the effect of raising the air temperature in the lower part of the occupied zone and hence linearising the overall temperature gradient in the room.

Therefore, in figure 1, the total flow rate required to ensure that stratification occurs above head level is 0·042 m^3/s.

Thermal comfort

High level mixing-type systems are designed to create uniform temperature and contamination conditions. In contrast buoyancy-assisted systems are designed to promote stratification so that the air being breathed is nearly as pure as the air entering through the air terminal devices.

Inevitably, therefore, occupants' feet will be cooler than their heads. This is further aggrevated by the movement of air across the floor at ankle level and the radiant heat gain to the head from the warm ceiling and the luminaires.

Wyon (1989) has found that, under typical operating conditions for summer and winter simulated in a laboratory cooled with air supplied via a 700 mm high semicircular low velocity atd, more than half his subjects reported sensations of discomfort in one or more body sections, mostly sensations of draught across legs and feet.

Discomfort was most prevalent under simulated winter conditions, when air was supplied at 19°C and the air temperature at the centre of the room was 22·5°C, among subjects seated 1·1 m from the atd, and amongst female subjects. Reported sensations were well correlated against predicted discomfort measurements made using a sectioned thermal manikin. Although this work was somewhat limited, in that it was based on only one type of atd, it points towards a real design problem when applying buoyancy-assisted RAD to sedentary occupations and low ceiling heights.

ISO 7730 (1984) recommends that the vertical temperature difference between 1·1 m and 0·1 m above the floor shall be less than 3 K. For comfort air conditioning applications, it is wise to use a smaller gradient, perhaps 2 K.

Laboratory measurements have shown that, in most cases, approximately half the difference in temperature between supply and extract air is lost in the first 50 mm above the floor although limited to a maximum of 4 K since supply temperature should be no lower than 18°C, and from there upwards, as shown in figure 6, the temperature gradient is usually near linear, regardless of air flow rate.

Hence for a ceiling height of 2·5 m and temperature rise of 2 K between feet and head a maximum rise between supply and extract air of 6 K is possible, whereas if a 3 K gradient is allowed then a supply to extract differential of 8 K is possible.

The rise in temperature between supply and extract air is caused by the total gain to the room from all heat sources. In the example given in figure 1 it is necessary to add the heat gains from the sun through a window with slatted blinds closed (370 W) (lights off) to the sources which generate the plumes. Hence with a total heat gain of 760 W the temperature rise is 7·4 K, producing a head to foot differential which is likely to be just under 3 K. If this is not acceptable then it will be necessary to recalculate flow rate using a 6 or 7 K differential between supply and extract air.

For comfort air conditioning it is recommended that the supply air temperature should not fall below 18°C, whereas for non-sedentary industrial applications 16°C may be acceptable.

Considering, for the moment, the low velocity atd: the supply temperature, the design, the height and the face velocity of the atd, as well as the proximity of the device to the nearest occupant are interdependent factors influencing thermal comfort and atd selection.

Face velocities are normally in the range of 0·25 to 0·35 m/s, but height, supply temperature and spread are more critical in the development of the velocity field close to the floor. Supply air leaving the atd falls towards the floor and accelerates at a rate dependent upon its density, and continues accelerating until it is slowed by the mixing process.

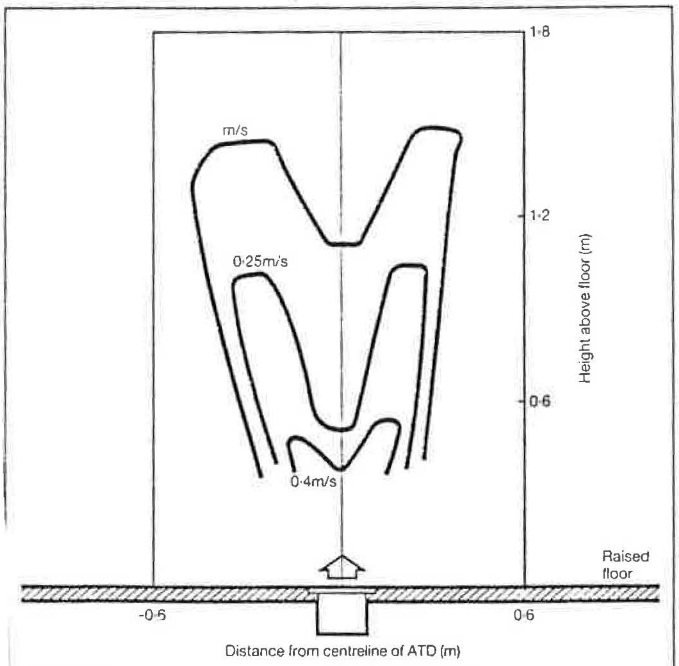
This cascade effect results in velocities at floor level some distance from the atd which may be higher than the face velocity. This means that there is a zone, close to the atd, in which occupants are likely to experience a draught, and particularly across the ankles. This zone is sometimes called the near or proximity zone, and it is defined as the zone within which velocities, usually measured at 50 to 100 mm from the floor, exceed 0·25 m/s.

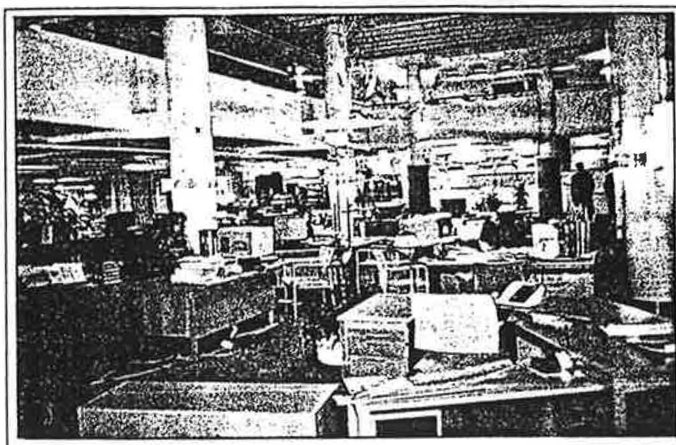
Some manufacturers give a range of near-zone depths for different velocities, eg for non-sedentary applications higher velocities may be acceptable, whilst in others, such as auditoria, lower velocities may be needed. Its size, for a particular flow rate and size of atd, will depend on the amount of spread and mixing induced local to the atd. For example air terminal devices with small perforations will create more local mixing than filter-faced devices, whilst a device which distributes air evenly through 180 deg will have a smaller near-zone than one with a narrower or uneven spread.

Usually the size of the near-zone is fixed by the application: for example under-seat atds in an auditorium may be only 300 mm from the occupants' ankles, while in many commercial applications a 1 m near-zone is quite acceptable.

A combination of the need to limit head to foot temperature difference and realistic selection of air terminal devices to give adequate useable floor area outside the near-zones,

Figure 4: Typical velocity contours for a floor-mounted atd with swirl: a 200 mm dia device discharging 0·035 m^3/s.





Above: An installation of semicircular low velocity atds (Stratos Floormaster) in the Skandia Life building near Stockholm.

means that, for the office air conditioning application, conventional buoy-ancy-assisted RAD using 2nd generation atds is limited to a room gain of about 25 W/m^2 (Sandberg, 1988).

At this stage the designer should look at a number of manufacturer's design procedures: some publish design guides, others computer-assisted selection procedures.

It is claimed that the induction devices, mentioned earlier, reduce the temperature gradient for a given load when compared with the conventional low velocity atd, and hence that a larger heat gain per m^2 can be dealt with, perhaps 50 W/m^2. Furthermore air can be distributed from central plant at a much lower temperature, perhaps 14°C, requiring half the volume, smaller ducts, fans etc and providing lower room humidities.

Similarly, rapid-diffusion floor-mounted atds provide greater mixing local to the device and hence a smaller temperature gradient and greater heat gains can be offset. It is also possible to achieve local mixing: by using fans to draw room air into a floor mixing void, as in the case of floor-mounted air terminal devices; or by induction, as in the case of the seat and desk-mounted devices.

Low velocity atds cannot be used to supply air which is warmer than the room air since its buoyancy will carry it directly to the ceiling. Floor-mounted devices can be used for heating however: for this purpose they are normally located under windows, in which case the rapid-diffusion type would not normally be required.

Contaminant control

In true displacement ventilation contaminants are pushed ahead of a piston of pure air. If this process is to be stable, and not influenced by forces such as buoyancy, gravity or moving bodies, then there must be sufficient momentum in the air to withstand these forces. This normally requires a high and even velocity (perhaps 0·4 m/s) across the whole room, leading to air change rates of up to 500/h.

An good example of this method of ventilation is the laminar downflow clean room. With buoyancy-assisted ventilation, contaminants are usually released at, near, or by the heat sources and carried out of the occupied zone in the plume. However, industrial processes which release toxic substances with low or zero buoyancy will usually require local exhaust ventilation in the normal way.

The volume of air (Q m^3/s) required to produce a reduction in concentration in the occupied zone (C_R ppm) of a known contaminant being released into the occupied zone at a rate of q (m^3/s) is calculated from:

$$Q = q \times 10^6/(C_R \times E)$$

where E is the ventilation index, C_E/C_R (assuming the contaminant is not found in the supply air), and C_E is the concentration in the extract air, ie the concentration that the contaminant would reach through the whole room with complete mixing.

C_R is a desired concentration, which should be some fraction of the occupational exposure limit (oel) - values for individual airborne contaminants are given in HSE (1988).

The ventilation index, E, increases with the thermal energy available in the plumes and the height of the room.

Very little published guidance is available on how to determine E, but although in theory it may be possible to obtain values as high as 10, measurements indicate that a reasonable design maximum is about 2, with 1·4 to 1·7 being more usual.

Recent developments

Most early installations of low velocity atds in Scandinavia were in industrial premises, but there has been particular interest in recent years in their application to offices.

This is a very competitive and cost-conscious sector, hence the evolution of the induction device, already mentioned, which requires smaller central plant and is likely to be able to deal with the sort of heat gains found in computerised offices.

Another way to deal with large cooling loads is by combining low velocity atds with cooled ceilings (see A ceiling for cooling in *Building Services*, July 1988). Although this may be an expensive solution, it enables much greater cooling loads to be offset (100 W/m^2 or more) and the panels provide direct radiant cooling of occupants and the internal surface, thus reducing the perceived temperature at head level and the temperature gradient in the occupied zone.

Danielsson (1988) has recently reported on the use of low velocity atds in a variable volume application, although both Sandberg (1988) and Holmberg (1987) have observed that the boundary plane can fall below the predicted level at low air change rates and low heat gains, resulting in air purity in the breathing zone no better than for a mixing system.

Conclusions

There is little doubt on the suitability of buoyancy-assisted ventilation, with low velocity atds, for tall spaces used for non-sedentary activities with high temperature contaminating processes. It is more difficult to design for the office application, however, but the potential for improved air purity and lack of draught around the neck region may well be worth pursuing, particularly if smoking is to be allowed.

Furthermore the elevated extract air temperatures create good conditions for air to air heat recovery, even in the UK's moderate climate. Both the low velocity induction device and the rapid-diffusion floor-mounted atds offer particular advantages for commercial applications with moderate heat gains, the former being particularly suited to buildings which cannot entertain a raised floor.

Terminal generated noise levels tend to be lower than most of the atds designed for mounting at high level.

Because of their sheer physical size, low velocity atds can be both more costly to buy and more difficult to integrate with the fabric than high level devices. Devices designed for insertion into a floor require a costly raised or channeled floor, although it is sometimes possible to avoid using a false ceiling.

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