# Giving designs a grilling

by Mike Holmes

**Many factors can determine the performance of room air terminal devices. Mike Holmes looks at test results in this area, and evaluates simple equations that may be adopted to predict diffuser behaviour.**

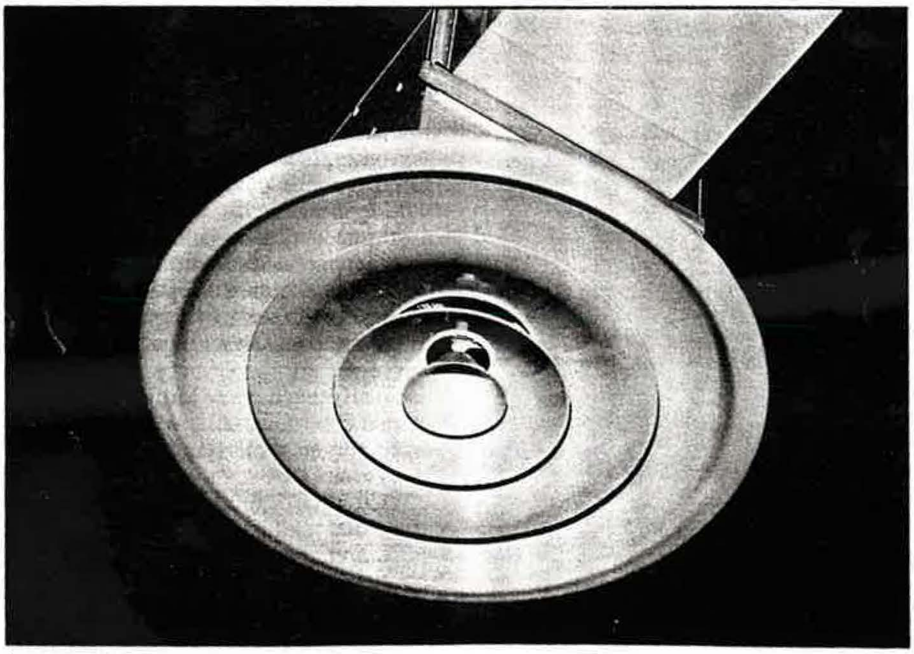
The room air terminal device is the link between the air conditioning or ventilation system and the occupant. Until recently most devices could be categorised as grilles, slots, nozzles and ceiling diffusers, the name 'diffuser' being the generic term for any device that is used to introduce air into a space.

The word diffuser implies that the device gently spreads conditioned air into the space so that it slowly mixes with the air already there and thus maintains a constant temperature. While this is the objective, the process is usually far from gentle, involving rapid mixing between room air and treated air in the immediate vicinity of the devices.

There are exceptions, however; the terminals used in displacement ventilation systems. These can be said to be true diffusers, as cool air is slowly eased into the space at floor level, with the resulting airflow pattern depending far more on the heat gains within the space than the aerodynamic properties of the diffuser.

The influence of the heat gain can also be a significant factor in determining the performance of a more conventional air terminal device[1]. However, I will limit what is said here to situations where the pattern of room air distribution is controlled by the supply air terminal.

Induction Air Systems has launched a new diffuser with automatic regulation of airflow distribution. A sensing actuator on the diffuser operates on supply air temperature and changes the outlet diffuser cones to maintain correct air velocities.



## Jet behaviour

The conventional air terminals all introduce air in the form of a jet and, in order to understand the basic principles of room air distribution, it is important to have some knowledge of the theoretical behaviour of such jets. The flow from these devices can be placed in one of the following categories; free axisymetric jet, free two dimensional jet, radial bounded jet and two-dimensional jet.

The bounded or wall jet is usually generated by a device mounted in the ceiling; typical devices would be circular diffusers (radial) and slots (wall jet), while free jets can be generated by nozzles and side wall grilles. In both cases the airflow pattern depends upon one simple jet characteristic ie the momentum flow within the jet remains constant and it is this momentum flow which controls the room air distribution pattern.

The reason for constant moment is simple; the pressure within the jet is a constant. In practice, small losses in momentum occur when a jet bends (to attach to a surface) or runs over a surface (inducing friction). However these are usually very small and can be neglected for present purposes.

All jets develop in a similar way. There is an initial region where the airflow pattern is dependent upon conditions at the terminal device. The length of this region depends upon the physical characteristics of the terminal. Some way away from the diffuser the flow pattern becomes independent of the device and is solely related to the physical characteristics of the mixing of two air streams. The important thing here is that all cross-sectional velocity profiles look the same. It is also this region of jet flow that most theoretical equations and manufacturer's data relate to.

## Manufacturer's data

In an ideal world all data presented by manufacturers would be on an identical basis, for example conforming to *BS 4773 Part 1, Methods for testing and rating air terminal devices for air distribution systems*. Unfortunately this is not the case. While the test procedures described in this standard are being reviewed it provides a sound way to obtain the basic characteristics of the devices being tested (figure 1). These are throw and spread to a terminal velocity of 0·5 m/s.

The velocity used is considered by some

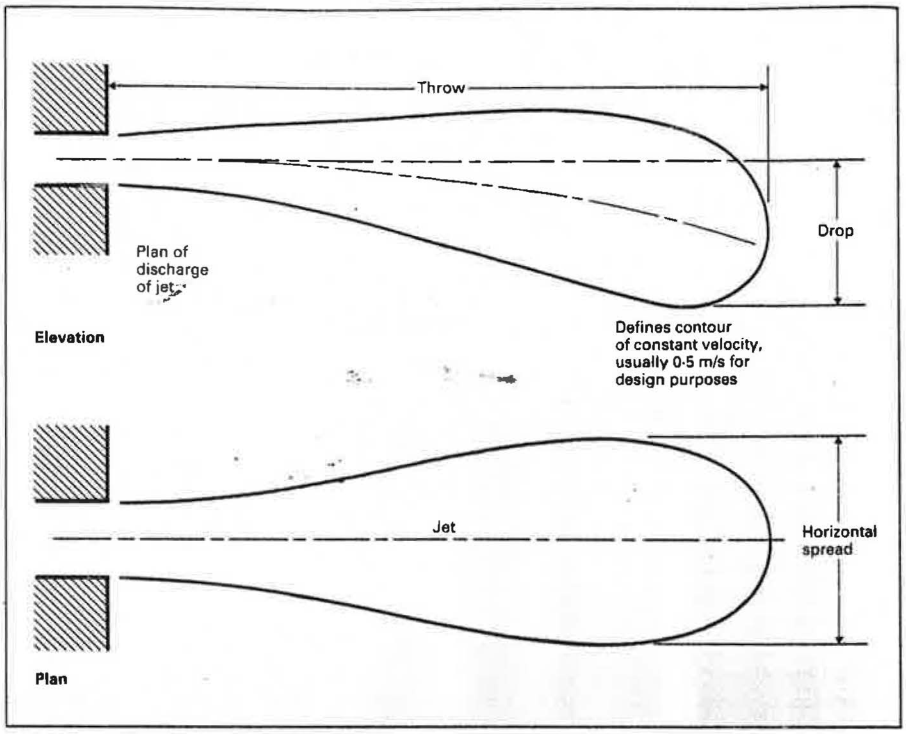



**Figure 1:** The throw, drop and spread of a jet to a terminal velocity of 0·5 m/s.

to be high because 0·5 m/s is well in excess of typical room velocities of 0·1-0·2 m/s. It is, however, relatively difficult for manufacturers to obtain reliable measurements down to such low speeds. In particular, temperature variations across the space can affect the measurements. The test facility and instrumentation required would therefore be more expensive than that needed to measure to velocities of only 0·5 m/s.

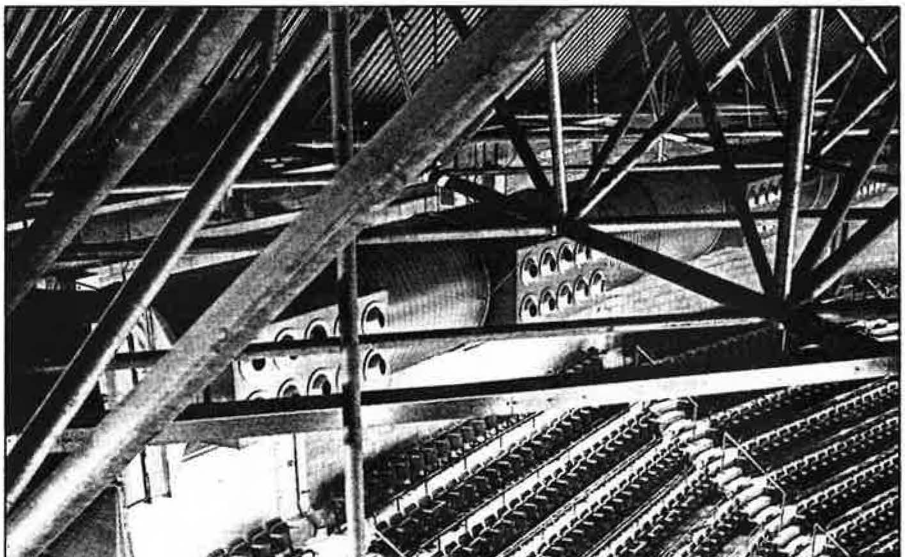
Furthermore, it has been established[2,3,4] that a terminal velocity of 0·5 m/s provides a sound basis for good room air distribution design (for spaces of normal height, a throw to 0·5 m/s equal to three-quarters of the zone length will usually give acceptable conditions).

So what is the value of the test results, and what other methods might be used by manufacturers? With regard to the latter, there would appear to be two main contenders, both based on relating the performance of the device to what happens in the occupied space. On the face of it the best test method would be to report average air velocities in the conditioned space, but such data are only valid for the size of space and conditions during the test. It is difficult to extrapolate the data and, while I sympathise with the rationale behind such a procedure, as a designer I find the data of limited value.

The second method is to measure the air speed at head height (1·8 m above the floor), about 150 mm from a wall towards which the device is throwing. This measurement should give the maximum level of discomfort in the space.

Specification of the maximum allowable velocity at this location has been used as the basis of a design method for vav systems[5]. The test is carried out using a velocity sensor mounted on a wall that is slowly moved away from the terminal device. The velocity is recorded as the terminal velocity to a throw equal to the distance from the wall to the diffuser. Two factors must be taken into consideration when applying results from this method; first, the terminal velocity and throw depend upon the height of the space and, second, a jet flowing along the ceiling will often detach from the ceiling when the room length exceeds three room heights.

Therefore the test results can be seen once again to be dependent upon the test configuration although this is largely avoided if *BS 4773 Part 1* is used. Further measurements obtained by application of methods presented in this standard can be used directly in the design of room air distribution systems.



Air is blown at high level via nozzle diffusers in this sports arena.

## Special devices

There are cases where it is necessary to make use of special terminal devices. One good example is the balcony slot used for distributing air in the concert hall at the Birmingham Convention Centre (see *Building Services*, January 1990). In this case the acoustical consultant had specified a very low noise level but the air distribution design required supply velocities in the region of 3 m/s – no conventional device would meet both requirements. Jets themselves are, however, silent; it is the vanes and other devices within the terminal that make the noise. All that was necessary was to use an understanding of basic jet theory to size the balcony slot diffuser.

What, then, is this theory? It was stated earlier that a jet is a constant momentum device and that, at some short distance from the jet, all velocity profiles are similar. This distance is about 5-10 effective slot widths or grille diameters (the effective slot width of a conventional slot diffuser is about 7-10 mm), but for most practical purposes this distance can be taken as zero. These two assumptions lead to the following jet equations:

□ for two-dimensional jets (slots, linear grilles)

$$\frac{U_x}{U_o} = K_1\frac{\sqrt{b}}{\sqrt{x}}$$

□ for three-dimensional jets (grilles, circular ceiling diffusers)

$$\frac{U_x}{U_o} = K_2\frac{\sqrt{A}}{x}$$

where:
$U_x$ is the maximum velocity at distance x from the device;
$U_o$ is the mean velocity at the device;
b is the effective slot width;
A is the effective area of the device;
$K_1$ and $K_2$ are constants that depend upon the turbulence level generated by the device.

The values of $K_1$ and $K_2$ are around 2·5 and 6 respectively. Furthermore, these equations only apply to isothermal jets – the characteristics of non-isothermal jets are beyond the scope of this article. The important thing to realise is that simple equations can be used to predict the performance of air terminal devices and, furthermore, have been used effectively in practice. Other examples are the nozzles in the Durngate Theatre, and the basic design for the National Gallery extension (a proprietary device was selected using the simple calculations as a basis for selection).

Finally, the equations can be used to extrapolate and confirm manufacturer's test data. If it is a linear device then it requires the throw to be increased by a factor of four to reduce the terminal velocity by 50%, while a factor of only two is necessary if a three-dimensional or radial device is used. If the catalogue data does not approximate to these rules look to find the basis of the information presented.

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References
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[2] Jackman P J, "Air movement in rooms with sidewall-mounted grilles – design procedure", *HVRA Laboratory Report No 65*, 1970.
[3] Jackman P J, "Air movement in rooms with sill-mounted grilles – design procedure", *HVRA Laboratory Report No 71*, 1971.
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[5] Holmes M J, "Designing variable volume systems for room air movement", *HVRA Application Guide 1/74*, 1974.

# From the ceiling downwards

by Dr Hans-Werner Roth

**Displacement ventilation systems usually use floor mounted diffuser systems, but Hans Werner Roth describes a new approach – a ceiling-mounted displacement system.**

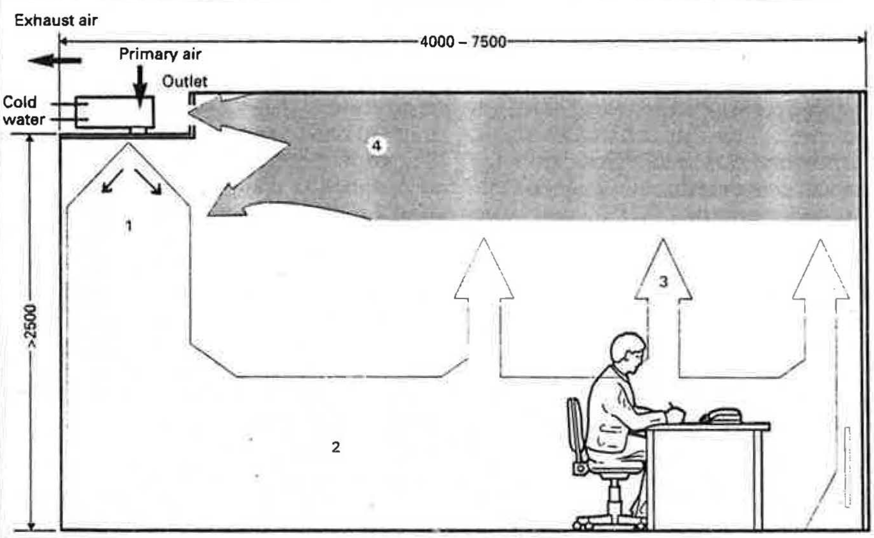
In recent years, building design teams have been stimulated by the re discovery of laminar displacement flow systems, passive wall cooling and passive cooling ceilings. Proven systems, such as variable flow air conditioning, turbulent mixing flow systems and air-water systems with induction units have been subjected to increasing criticism.

After an initial period of euphoria with regard to laminar displacement flow, experience with commissioned systems showed that the specific cooling capacity had to be restricted to around 30-40 W/m² of office space. This limit prevents the temperature difference between feet and head level from exceeding 1·5-2 K (at a room temperature of 22°C).

The passive-cooled ceiling offers a suitable alternative by separating the cooling from the ventilation requirements. Ventilation and pollutant control is handled by a laminar displacement flow system, while sensible cooling is achieved by a passive cooled ceiling.

One of the main arguments in favour of this system is the high degree of thermal comfort, mainly determined by low room air velocities and levels of turbulence.[1]

**Figure 1:** Qualitative flow diagram, based on the LTG Indivent system.



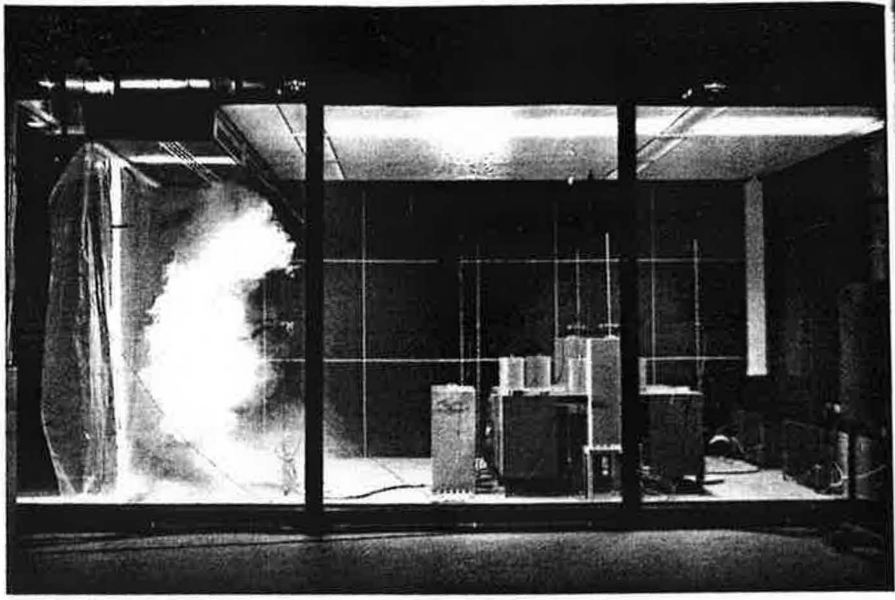

1: Turbulent mixing flow = > temperature drop due to mixing room air

2: Displacement flow = > supply air mixed with room air from corridor to facade

3: Thermal currents and room air circulation transports pollutants and heat loads upwards

4: Backflow to exhaust and return air outlet and to mixing with supply air

An air-water system with a comparable cooling capacity, and using an induction or fan coil system, is often felt to offer a poorer standard of comfort. The reasons for this include, for example, incorrect selection (frequently an excessively high cooling capacity), installation and poor commissioning and maintenance.



**Figures 2a & 2b:** Illustration of the air flow regimes by means of smoke trace (overall cooling load of 50 W/m$^2$, and primary air at 50 m$^3$/h).

## The role of the diffuser

The reasons why cill-mounted systems cannot tolerate errors during cooling relates to the rules of air diffusion. If the jet impulse is too weak, the cold jet of air prematurely enters the occupied zone. If the impulse is too strong, the room air velocities are excessively high. This physical interrelation limits the performance range of cill-mounted induction units to typical loadings of 50-60 W/m$^2$.

The air flow can be made more stable if it is turned 'upside down'. With this arrangement the cooler air flows along the floor, a phenomenon comparable to local displacement flow. In the mixing zone between the ceiling and the floor, the 'under-temperature' of the current of supply air must be reduced to eliminate the risk of 'cold feet', an undesirable comfort parameter.

Figure 1 shows a schematic illustration of this air diffusion arrangement. A diffuser with an integrated cooling unit has been installed in a ceiling on the corridor side of the room. It is comparable to a ceiling mounted induction unit since it uses cold primary air from a duct system operating at a pressure of between 150 and 300 Pa, in order to draw in recirculated air from the room and convey it through a cooling element.

This mixture of outside air and recirculated air is then diffused into the room via micro-jet outlets. The temperature difference between the room air and the supply air is quickly reduced in the mixing zone (1).

At the same time, the air velocities reduce to between 1·6 and 1·8 m/s or less, depending on the cooling load. One cooling jet (2) created in this way is turned around at floor level and slowly flows towards the facade. The fact that the supply air always has a lower temperature than the mean room temperature ensures a sufficient penetration depth up to the facade.

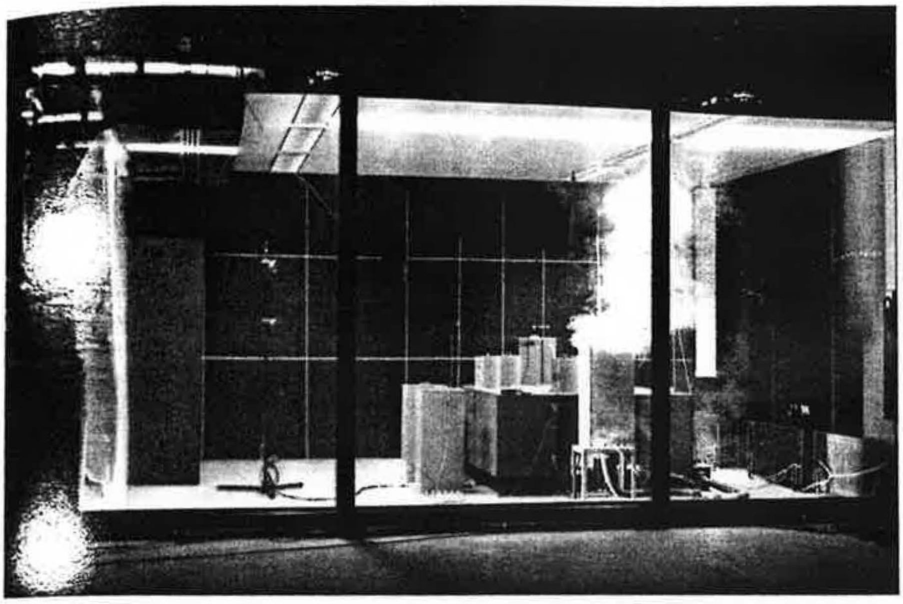
The upward flow (3) is created by heat sources such as people, equipment and surfaces heated by solar radiation and supports the room air circulation wave, but does not necessarily develop it.

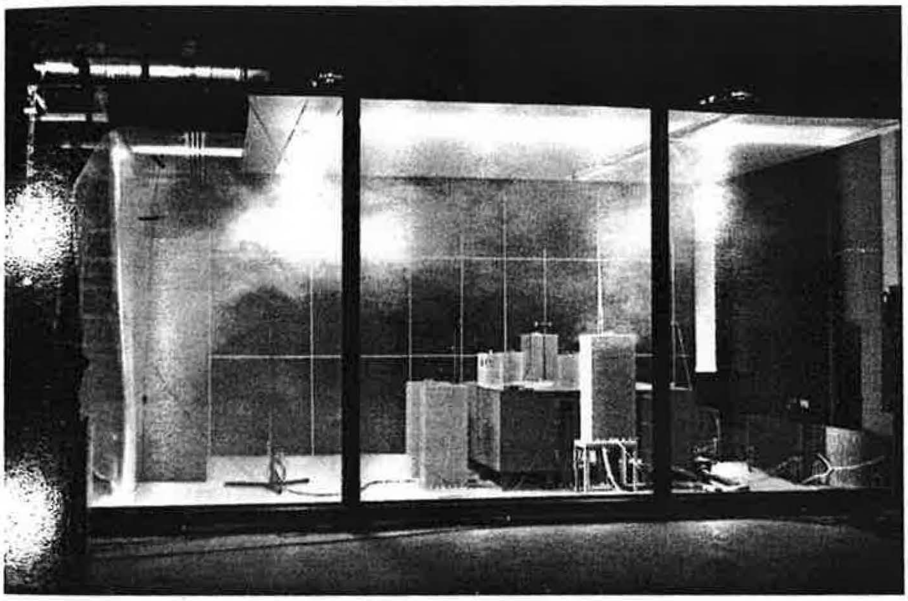
The flow pattern itself does not change in the heating cycle, when separate perimeter heaters are used. Primary air with a temperature of 18°C, for example, is supplied to the diffuser in its bypass position. The convector radiator below the window is designed to compensate the perimeter heat loss and reheats the primary air locally to the room temperature. Thus free cooling, which is a proven feature of variable flow systems, can be exploited to its full potential.

Figures 2a and 2b illustrate the flow smoke patterns in a full scale mock-up of an office with a room load of 50 W/m$^2$.

The heat loads on and at the desk are generated by electrically-heated metal containers. The jet of supply air ventilates the occupied zone to a height of 1-1·5 m, and can thus effectively extract air pollutants which are not coupled to heat sources.

A pocket of warmer room air with an increased pollutant concentration (odour substances, cigarette smoke etc) forms below the ceiling.





**Figures 3a & 3b:** Illustration of the natural thermal current in the vicinity of a simulated person with cooling loads the same as in figures 2a and 2b.

## Practical considerations

Although the system allows a room to be put to a wide variety of uses, ie work places can be set up in the immediate vicinity of windows, the minimum floor-to-ceiling height should be around 2·3 m, with a diffuser-to-wall clearance of 100 mm.

The ventilation system requires a twin line high pressure plant, with heating always provided at the facade by means of static heating panels. Depending on the type of unit used, regulation can be carried out both on the water side, by means of valves, and on the air side by means of an actuator-controlled bypass damper.

If a cooling capacity of more than 60-70 W/m$^2$ (at a room depth of 6 m and a room temperature of 22°C) is required in the case of constant volume flow units, then the occupied zone directly beneath the diffuser is not suitable as a workplace since the air velocities may be excessive.

In open plan offices, the ceiling diffusers may only be used if they are installed along the central aisle in two rows at an interval of around 2 m from one another,

## Test results

In figures 3a and 3b, a metal body used to simulate a person is injected with smoke. The smoke rises to the ceiling, spreads out and is extracted in an increased concentration. As a result of the induction below the ceiling diffuser, a sub-current from the ceiling zone is mixed into the supply air.

The amount of smoke absorbed indicates that, as in the case of laminar displacement flow systems, the natural thermal current at the person's body is not influenced by the ventilation system. As a result, the occupant does not perceive the air conditioning system.

This method of ventilation combines the benefits of turbulent mixing flow – small temperature gradients, stable room flow characteristics – with the benefits of laminar displacement flow – low air velocities and improved air quality.

Installing the units on the corridor side of a room offers greater freedom for the design of the ceiling than is usually the case with passive ceilings. The diffuser can be installed in a ceiling step, as shown in figure 1, or alternatively in a smooth, suspended ceiling construction.

Ceiling diffuser with integrated cooling unit, type LHG 800

Primary air volume/ axis $V_P$ = 50m$^3$/h  Axis width b = 1·5 m

Total cooling load Q/A 50 W/m$^2$  Q/b = 275 W/m$_{\text{room width}}$

$t_{ZU} - t_{AB} = -6K$
$t_{ZU} - t_R = -4K$

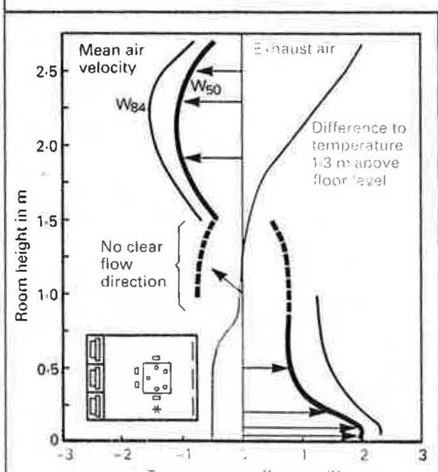



**Figure 4:** Graph of velocity and temperature at 50 W/m$^2$.

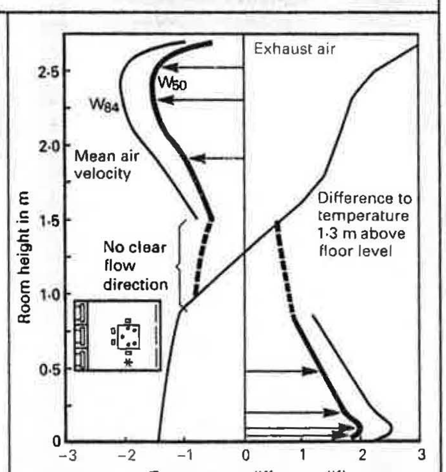



**Figure 5:** Graph of velocity and temperature at 125 W/m$^2$.

and if the supply of air to the workplace is not impeded by partition walls.

Investigations are currently being carried out regarding the extent to which current penetration depths of more than 7 to 8 m can be achieved.

Figure 4 shows a typical office with resultant air temperature and velocity curves. The ceiling diffusers are located on the left in the ceiling bulkhead, and the facade has one heating panel on the right.

Temperature and velocity curves have been drawn over the room height for the measuring point indicated directly behind the simulated chair (with a specific cooling capacity of 50 W/m$^2$).

The difference in temperature between seated head level and feet (0-1·3 m) is 0·6 K. The temperature only increases by around 2 K above a height of 1·5 m. The temperature curve is very similar to that obtained in the case of room ventilation using inductive floor diffusers.

As with laminar displacement flow systems the velocity is greatest at floor level, but, due to the low turbulence level, is not perceived as a draught, being typically less than 2 m/s.

Of greater significance is the temperature itself, which may not exceed 20 21°C. Over a height of around 0·5 m, the room air velocity is around 1 m/s. A pronounced backflow can be measured below the ceiling. If in extreme cases the room temperature is allowed to rise to 27°C and above, cooling loads of up to 125 W/m$^2$ can be handled (see figure 5). In this case the difference in temperature up to a height of 1·3 m increases to 1·5 k.

This test shows that, in the case of 2·5 air changes/hour, it is possible to extract high cooling loads without causing draughts. If one considers that in the majority of offices cooling loads range from between 30m and 60 W/m$^2$, then there is no reason to reject water-air systems.

These ventilation systems can be combined with window ventilation without difficulty, and the room temperature can be controlled at the reference value within a matter of minutes. Passive-cooled ceilings react slowly, and experience will have to show whether, in conjunction with mechanical ventilation, it is permissable to open windows.

Dr Hans-Werner Roth is director of research, development and testing at LTG Lufttechnische GmbH, Stuttgart, Germany.

**Reference**

[1]Bunn R and Wyatt T, "The future for cooling ceilings?", *Building Services*, November 1991, p33.

# Breaking the sound barrier

by Neil Jarman

**Designing for grille and diffuser noise is of paramount importance. Neil Jarman looks at the data currently provided by manufacturers and provides some design tips.**

Noise levels generated by grilles and diffusers have a significant effect upon total noise levels within a room and therefore designing for grille and diffuser noise is important. If a system is commissioned to the correct operating conditions and the grilles are too noisy, then changes have to be made to the grilles themselves.

Unlike most other components in the system, you cannot fit acoustic lagging or a silencer to a grille. There is, however, no reason why the installed diffusers and grilles should be too noisy, if noise generation is given proper consideration in the design programme.

## Empirical vs manufacturers' data

There are several empirical formulae in existence for the prediction of diffuser noise, but inevitably there is some marginal error in these. Manufacturers' regenerated noise data for grilles and diffusers should be available from potential suppliers and should therefore be used in preference. If one manufacturer's data appears doubtful then comparison with empirical predictions (using formulae in the *CIBSE Guide*) or other manufacturer's data can be beneficial. Also, read the small print – again!

Manufacturers' noise data often includes predicted NC/NR values in the room. Reference to the small print will usually indicate that the prediction is based upon a room absorption of 8 dB, ie that the room noise level is taken as 8 dB less than the sound power level radiated to the room from a single grille. For most practical situations these predicted NC/NR values are optimistic. This is illustrated in figures 1 and 2 which show a large cellular office 6 m by 6 m, carpeted and fitted with an acoustic suspended ceiling.

In these examples, if the diffuser type selected was claimed to be NR35, assuming an 8 dB room effect, then, in the first example, resultant noise levels would be NR35, but in the second example NR41.

For most engineers, the second example is more realistic of current design practice. The modern, heavily serviced office has more than a single diffuser. It is also reasonable to assume that office workers will be as close as 1·5 m from the diffusers.

This is not a criticism of any individual manufacturer; while all diffuser and grille

## Diffuser noise and vav systems

Making appropriate diffuser selections for vav systems can be a difficult exercise. A diffuser which gives NR37 at 100% volume while not 'dumping' at low volumes can be difficult to find.

Using special diffusers can help, but one should ask whether it is necessary to achieve the noise criterion at 100% volume. Typically, 100% volume may only be supplied for a few hours of a few days each year. Designing to achieve the noise criteria on 80% volume can result in better supply air conditions when operating on low volumes, while occasionally compromising the noise criteria by only 3-4 dB.