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Technical Note 3/90

Design recommendations for room air distribution systems

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Addendum to Technical Note 3/90
Design Recommendations for Room Air Distribution

This sheet is provided to enable the reader to fully utilize the equations relating to non-isothermal jets which appear on page 45 under Table C1 of the Technical Note.

The equations below are the ranges which apply to the three regions for both types of jets shown in Table C1.

	Axisymmetric jet	Plane jet
Non-buoyant region	$F^{-1/2} \left(\frac{\rho_o}{\rho_a}\right)^{-1/4} \frac{X}{D} < 0.5$	$F^{-2/3} \left(\frac{\rho_o}{\rho_a}\right)^{-1/3} \frac{X}{D} < 0.5$
Intermediate region	$0.5 \leqslant F^{-1/2} \left(\frac{\rho_o}{\rho_a}\right)^{-1/4} \frac{X}{D} \leqslant 5$	$0.5 \leqslant F^{-2/3} \left(\frac{\rho_o}{\rho_a}\right)^{-1/3} \frac{X}{D} \leqslant 5$
Buoyant plume region	$5 < F^{-1/2} \left(\frac{\rho_o}{\rho_a}\right)^{-1/4} \frac{X}{D}$	$5 < F^{-2/3} \left(\frac{\rho_o}{\rho_a}\right)^{-1/3} \frac{X}{D}$

Ranges relating to non-isothermal jets

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1. INTRODUCTION

The technology of the design and application of room air distribution systems has developed over many years. Contributions to this development have been made by a number of research, academic and professional institutions in several countries, and in this BSRIA has played a significant part.

This work has contributed to the establishment of standard design procedures that may successfully be followed for many applications. However, where novel features of the room or the thermal conditions are encountered a particularly onerous demand is placed on the designer. An understanding is required of the many complex factors that determine the behaviour of air streams within rooms and that influence the resulting thermal environment.

This report discusses these factors and provides guidance on good design practice in the application of air distribution systems. It is based on a project report produced from a group-sponsored study completed in 1988. A copy of the full report ''Design recommendations for room air distribution systems'', Contract 7554, Report No 3[1], is held in BSRIA's Library.

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Department of Environment

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2. DESIGN OBJECTIVES

The main objective of building design is to provide an internal environment suitable for the occupants and any processes to be undertaken inside. With respect to air distribution design, the objectives may be defined as thermal comfort and ventilation effectiveness to be achieved without intrusive noise generation. These criteria are described below.

2.1 THERMAL COMFORT

Thermal comfort can be defined in terms of the perception of satisfaction that a subject experiences with his or her thermal environment. The significant environmental parameters are:-

air temperature

radiant temperature

air velocity

and humidity

Of these, humidity is not significantly dependent on the air distribution system; neither is the mean air temperature because that can usually be controlled in accordance with requirements, which in combination with the mean radiant temperature, form such comfort-related indices as dry resultant temperature (see CIBSE Guide[2]).

The comfort-related parameters that are mainly influenced by air distribution are:-

spatial air temperature difference

air velocity

and possibly, asymmetric thermal radiation.

2.1.1 Air temperature difference

Research on human comfort has resulted in the general recommendation that the temperature difference between head and feet level should be less than 3 K. Thus, the limiting temperature gradient depends on whether the occupants are considered to be standing or seated. In the relevant international Standard[3] the seating position is assumed because there it recommends that the 3 K differential should be between 1.1 m and 0.1 m above the floor. This gives a gradient of 3 K/m.

However, if standing is assumed the limiting gradient becomes 1.8 K/m. For office accommodation, a limit of 2 K/m or 1.5 K/m is recommended.

It is also recommended that the limits of variation of temperature across the occupied zone of the room should be within 3 K, that is +/- 1.5 K about the mean room air temperature[2].

2.1.2 Air velocity

In the ISO Standard[3] it is recommended that for the comfort of sedentary occupants the mean air speed to which they are exposed should be less than 0.15 m/s in winter and 0.25 m/s in summer.

However, more recent studies by Christensen[4] suggest more stringent limits. These studies were conducted with room air turbulence levels normally associated with mechanical ventilation. The derived limiting velocities at various temperatures are indicated on (Figure 1) where the relationships between temperature and velocity are shown for 10% and 20% levels of dissatisfaction. Relating this Danish data to the recommended design temperatures for sedentary occupants in the UK[2] gives the following air speed limits, see table opposite.

Design Temperature °C		Mean air speed (upper limit) m/s	
		10% dissatisfaction	20% dissatisfaction
	19	0.10	0.13
winter			
	20	0.11	0.15
summer			
	22	0.13	0.19

For industrial and other applications for more active occupation, higher velocities may be allowed or even required, while in special facilities, such as sports halls for badminton and table-tennis, there could be more stringent limits on air velocities.

2.1.3 Radiation asymmetry

Most occurrences of radiation asymmetry are due to hot or cold surfaces that are generated independently of the air distribution system. A cold window is a common example. However, it is possible for high internal air temperature gradients to give rise to significant differences in the boundary surface temperatures. So, for example, in applications where air temperatures at high level may be allowed to rise substantially checks need to be made to ensure that the radiant effects of high temperatures at the ceiling or roof do not cause discomfort to the occupants.

For comfort, the upper limit of the difference between the thermal radiation from the upper and lower parts of a room corresponds to a difference of 5°C in the radiant temperature on the opposite sides of a small horizontal plane. This 5°C limit is given in ISO Standard 7730[3] in relation to sedentary occupation in winter.

2.2 VENTILATION EFFECTIVENESS

In addition to the provision of an acceptable thermal environment, the purpose of an air distribution system in occupied accommodation is to provide 'fresh' air for respiration and to remove contaminated air. Ventilation requirements are given in the CIBSE Guide[5] (Section B2) and the recommended level for sedentary occupancy of 8 litres/per person is based on the need to dilute and remove body odours.

However, simply supplying and exhausting the correct quantity of air will not be adequate unless the supply air is distributed properly and the exhaust air carries away the contamination.

To achieve effective ventilation an understanding of room air movement behaviour as well as a knowledge of the location and strength of the pollution sources is required. There are two basic types of flow pattern that may be applied for ventilation purposes, "piston" flow and mixed flow.

Piston flow, otherwise known as "plug flow", consists of air movement in one direction such that clean air enters at one side (or top or bottom) of a room and sweeps across it and leaves at the opposite side having picked up pollutants released in the room (Figure 2). This type of flow is most suitable for cases where the pollutant source is in a fixed location or plane and where those requiring protection are upstream of that location. Another type of flow that is largely uni-directional is buoyancy driven where slightly cool air is introduced at low velocity at low level and convective flows cause upward motion with the heated, often contaminated air exhausted at high level. This is known as displacement ventilation and is most effective in tall spaces with high-temperature heat loads. This type of ventilation has been the subject of a separate study by BSRIA and is reported elsewhere[6].

Mixed flow is where the air movement is such that the incoming fresh air thoroughly mixes with the room air (Figure 3). This has the effect of diluting the airborne pollutants before they are removed in the exhaust air. This type of flow is applied in cases where the pollutant source is diffuse or mobile and where sufficient dilution can be obtained to provide a safe and healthy atmosphere. In relation to thermal comfort, mixed flow produces a more uniform air temperature distribution than piston flow or displacement ventilation flow. Although perfect mixing is often assumed to occur, in reality variations in the degree of mixing will occur in different parts of a room. Poor ventilation results in the extreme case known as "short circuiting" when the clean air supply enters the exhaust system without substantial mixing with the polluted air (Figure 4).

There are differing definitions of ventilation effectiveness depending largely on the objective in view and the type of air flow system used, displacement or mixed. For further information on this subject reference should be made to the Air Infiltration and Ventilation Centre's review of the subject in AIVC Technical Note 21[7].

2.3 ACOUSTIC ENVIRONMENT

An important aspect of room air distribution design is the achievement of acceptable levels of noise. Air flow generated noise through air terminal devices must be of a quality and at a sufficiently low level to be unobtrusive in the environment in which they are applied.

The CIBSE Guide[2] contains recommendations on the noise levels appropriate for various applications. These recommendations are based on NR curves which attempt to express equal human tolerance in each frequency band. NR values range from 20 for such applications as recording studios to 50 and above as may occur in a space accommodating some manufacturing processes.

A standard method of acoustic testing air terminal devices is given in BS 4773 Part 2[8], and resulting data presented by manufacturers in their catalogues may be used in noise level analysis. In an open space, the noise generated by airflow through an air terminal device radiates from it such that as the distance from the source increases the sound pressure level decreases. When bounded by adjacent surfaces, the sound pressure field may become non-uniform and this effect, known as directivity, results in some positions equi-distant from the source having higher noise levels than others. Within an enclosure, the sound level is a combination of the effect of the direct sound radiation from the source and the sound reflected from the room surfaces. The magnitude of the reverberant component depends on the sound absorbing or reflecting characteristics of the surfaces. For the same source, a hard surfaced room will have a higher sound pressure level than a room with carpets and furnishings.

Methods for determining both the noise generation characteristics of air terminal devices and the resulting noise levels in rooms are presented in the HEVAC Air Diffusion Guide[9].

Additional information produced by BSRIA on the noise generated by air flow through grilles and dampers is included in its Laboratory Report No 75[10]

3. AIR DISTRIBUTION PRINCIPLES

3.1 SUPPLY AIRSTREAM CHARACTERISTICS

3.1.1 Velocity decays

A basic understanding of the performance of airstreams discharging into rooms is fundamental to the design or analysis of room air distribution systems. The relationships that govern airstream (or jet) behaviour, which are generally empirically based, are presented and reviewed below. This information relates to turbulent jets as normally associated with mixed-flow systems. The performance of low velocity air terminal devices associated with displacement flow systems is not considered here but is included in another BSRIA Technical Note[6].

Immediately air leaves a supply air terminal device the boundary of the jet begins to spread due to the process of turbulent diffusion or mixing with the surroundings. The influence of the diffusion process is to entrain room air and reduce the speed of air movement within the airstream as it crosses the space. At a certain distance downstream of the air terminal the diffusion process reaches the centre-line or axis of the jet and its centre-line velocity then starts to decay. The diffusion continues until the jet can no longer be distinguished from the general room air movement in the surroundings. The process described above is one in which a relatively small volume flow rate at a high air speed (the supply air velocity) is converted to a high volume flow but low air speed in the main body of the space.

A typical velocity decay in a jet is shown in Figure 5. The decay occurs in four stages (Nevins[11]). In the first stage, which extends over a distance of approximately four terminal diameters or widths, the velocity at the centre-line remains constant. The second stage is a transitional zone which extends to eight or ten diameters. The main zone which dominates the characteristic and is the area of main engineering interest extends for 10 to 100 diameters depending on the initial velocity and the supply terminal shape. In this zone the relationship between the velocity on the centre-line of the jet and the distance from the source depends on the type of supply air terminal device. Some examples are shown on Figure 6. The final zone of the jet is where the velocity quickly decays to a low value and the jet becomes indistinguishable from its surroundings.

The basic aerodynamic performance of an incoming jet can be characterised in terms of the throw, the spread and the drop of the jet. These are parameters which, associated with a specified terminal velocity, define in practical terms the envelope or contour of constant velocity of the jet. The throw is the longitudinal distance from the supply outlet of the jet to the extremity of the envelope. The horizontal spread is the maximum dimension of the jet envelope, measured in the horizontal plane, and the drop is the vertical distance from the centre-line of the jet outlet to the underside of the envelope. Figure 7 defines the throw, the horizontal spread and the drop of a horizontally projected jet.

The throw of a vertically projected jet is measured in the vertical direction, and the spread in the horizontal direction (see Figure 8).

The velocity chosen to define the jet envelope is called the terminal velocity. The preferred value is 0.5 m/s and BS 4773 Part 1[12] defines methods of test to determine the throw, spread and drop to this velocity.

3.1.2 Temperature difference decay

In heated or cooled (non-isothermal) supply airstreams the mixing process reduces the difference between the temperature of the air in the jet and that in the surrounding air as the movement progresses across the space. Experimental evidence has shown that the reduction of the temperature difference occurs at a slightly faster rate than the velocity decay. However, in general air distribution design, the decay of temperature difference may be assumed to occur at the same rate as the velocity decay (CIBSE Guide (1970)[5]).

3.1.3 Buoyancy effects

If a horizontally projected jet is either cooler or warmer than the surrounding air, it will be deflected from its horizontal course because of the differences in air density. A cool jet will deflect downwards so that its drop will increase (Figure 9), while a warm jet will deflect upwards and the distance of the upper surface from the horizontal centreline is known as the rise (Figure 10).

The trajectory of a heated or cooled jet has been found to depend on its Archimedes number in relation to its surroundings. Archimedes number is the ratio of buoyancy to inertia forces and is a function of temperature difference, initial jet velocity and the size of the supply terminal. Full details of the relationship is included in Appendix A.

The higher the temperature difference and the lower the velocity the greater will be the deviation from a horizontal trajectory. Tables 1 to 5 show predicted deviations for a number of typical conditions. These were derived from the relationship given in Appendix A.

Non-isothermal jets discharged vertically will be influenced by buoyancy forces such that the throw will either be increased or decreased depending on the direction of discharge of the jet and the temperature of the jet relative to its surroundings.

A downward discharged warm air jet will behave in the same way as an upward discharged cold air jet. In these cases the buoyancy forces will act to reduce the effect of the inertia forces and therefore the throw of the jet will be reduced. This is important to the design of a heating system which makes use of nozzles at ceiling height. The successful operation of the system would hinge on the ability of the nozzles to project the warm air to the occupied zone, whilst ensuring that if operated under near isothermal conditions (or indeed cooling conditions) excessive air movement at ground level would not be induced.

Empirical expressions have been derived to predict the throw of heated jets projected downwards. These are outlined in Appendix B. From them Tables 6 and 7 have been produced to enable predictions for particular applications to be made for both circular and linear (plane) jets.

The downward discharge of cold air or upward discharge of warm air is a configuration where the buoyancy forces act in the same direction as the inertia forces. Experimental data for buoyant jets discharged into calm or stably stratified environments have been reviewed by Chen and Rodi[13]. It is shown in the above case that a buoyant jet issuing into a calm environment initially decays in velocity until it eventually behaves as a pure plume. As a pure plume, it is driven by buoyancy forces only and moves at a constant finite velocity. Although, in practice, air movement currents in a room can destroy the formation of a pure plume, care must still be exercised to ensure that the performance of such jets is appropriate for the application.

Appendix C shows the equations derived for the centre line velocity and temperature in the non-buoyant, intermediate and buoyant plume region for both circular (axisymmetric) and plane (linear) jets. Based on these equations Tables 8 and 9 show data for both circular and linear chilled jets projected vertically downwards for terminal velocities of 0.3 m/s and 0.5 m/s. At the specified terminal velocities, the jet air temperature does not differ by more than 1°C from the room temperature.

For information on the trajectory of jets projected at an angle, reference should be made to the original project report[1].

3.1.4 Surface effects

When an airstream flows along a surface such as a wall or ceiling, the interaction of the jet and the surface generates a force such that the jet will be attracted to and continue to flow along the surface. This phenomenon, which is very beneficial for air diffusion design, is known as the surface or Coanda effect. With correct design it allows cool air to be discharged beneath a ceiling with limited deflection down into the occupied zone. The attachment to the ceiling counteracts any negative buoyancy forces.

Provided the initial velocity is greater than 1.5 m/s, cool airstreams discharged horizontally from ceiling diffusers normally continue to flow just below the ceiling surface because the Coanda effect is so strong.

Airstreams discharged from side-wall grilles may or may not continue horizontally depending on the ratio of buoyancy, inertia and Coanda effects. A BSRIA report[14] provides a means of calculating the required aspect ratio of a side-wall grille to avoid excessive drop of the jet. For the coanda effect to be significant, the grille should be mounted such that the upper edge is within a distance of half the width of the grille from the ceiling.

3.2 VELOCITY DISTRIBUTION AT EXHAUST OPENINGS

Whereas the velocity decay of a supply air jet (down to say 10% of the value at the source) occurs over considerable distances, the velocity distribution at exhaust openings is such that an equivalent 10 fold increase in velocity occurs over very much shorter distances.

A typical plot of velocity contours (see Figure 11) shows that with an exhaust terminal mounted in a surface, the velocity at a distance of one diameter from the opening is nowhere greater than about 10% of the duct velocity.

Thus exhaust terminals have much less influence on the movement of air in a room than supply terminals so their selection and location is generally less critical. However, there are some factors that need to be taken into account in choosing the locations of exhaust terminals and these are considered in section 5.8.

For more detailed information on the performance of exhaust systems used in local pollution control reference should be made to CIBSE Guide[5] (Section B3) and BSRIA Technical Note 3/85 "Design guidelines for exhaust hoods"[15].

4. TYPES OF DISTRIBUTION EQUIPMENT AND SYSTEMS

4.1 SUPPLY TERMINAL DEVICES

The proper selection of supply air terminal devices is critical to the attainment of the desired thermal and air movement environment in a conditioned space. It is therefore most important that the appropriate type of device is used consistent with the air distribution requirements. Figure 12 illustrates the main types of terminal.

In this section the general characteristics of supply terminal devices are reviewed and guidance is given on typical air change rates based on a supply-to-room air temperature difference of 10 K. This is a typical temperature difference for cooling applications where thermal comfort is important[9].

4.1.1 Nozzles and Drum-Louvres

Nozzles, usually of circular section, are generally used for long throw applications (up to 20 m to 30 m), although some devices allow an alternative mode of operation using a flared diffuser discharge which can reduce the throw. Nozzles have been used predominantly in factory heating systems and auditoria. Their use can with care be extended to other large-building applications where comfort requirements may be quite stringent. It is desirable, though, that flexibility be built into the system to allow some adjustment of the nozzle discharge angle during commissioning. For applications where noise is not a major problem (ie. light industrial premises) then supply velocities up to 10 m/s can be used.

Drum-louvres are also appropriate for long throw applications and where pressure loss is to be minimised. They consist of a rectangular louvre formed along the axial length of a cylinder/drum, which is mounted horizontally so that adjustment of the vertical inclination of the supply jet can be achieved by rotation of the drum. Adjustable blades may be fitted within the louvre for modification of the jet spread.

4.1.2 Grilles

Grilles, which may have fixed or adjustable blades, discharge air in a three dimensional flow pattern normally in a direction perpendicular, or near perpendicular, to the grille face. In the main zone of expansion the velocity of the air reduces in direct proportion to the distance from the outlet. A wider spread of jet can be achieved by adjusting the settings of the blades.

Linear grilles, which are defined as having rectangular frames with aspect ratio 5:1 or greater, produce a two-dimensional (plane) airstream. The velocity decay in the main section of this type of airstream decays in proportion to the square root of the distance from the grille.

Grilles normally handle flow rates up to the equivalent of about 8 to 10 room air changes an hour (ach) and at discharge (face) velocities up to 10 m/s or so.

4.1.3 Linear and slot diffusers

Linear and slot diffusers, either single or multi-slot, produce a two dimensional flow pattern within an included angle of about 20 degrees. They can be used for a vertically downward projection of air or a horizontal projection beneath the ceiling. Where freedom from excessive air movement is important then great care should be taken if used for vertically downward projection of cool air.

A little way from the outlet the velocity starts to decay in proportion to the square-root of the distance from the diffuser. They are appropriate for applications requiring air change rates up to approximately 15 ach, with discharge velocities normally up to 10 m/s.

4.1.4 Circular, square and rectangular diffusers

Circular, square or rectangular diffusers are normally used in ceiling applications and allow a high air flow rate to be introduced into a room. In the main zone of expansion the velocity of the air stream reduces in direct proportion to the distance from the outlet. Air change rates up to approximately 20 ach can be provided using these types of device.

Depending on the detailed design of the diffuser, alternative air flow patterns can be produced. For example, a multi-cone circular diffuser may have the ability to project air horizontally beneath a ceiling or, by adjustment, to discharge the air downwards. Diffusers are available with motorised deflector blades which utilise a swirl action for horizontal projection of cool air beneath a ceiling and vertical discharge of warm air for heating.

Flexibility of flow pattern control can be provided by appropriate selection of square or rectangular ceiling diffusers to give one, two, three or four-way discharge. For normal applications in occupied rooms core/neck velocities up to 6 m/s are suitable.

4.1.5 Perforated face diffusers

These are usually the simplest and cheapest form of device. They are suitable for extract and transfer grille applications and also for supply purposes where air is to be introduced into a room in a significant volume (up to 15 ach) over large areas. The free area of the devices is between 10 and 50%. They are normally applied at face velocities up to approximately 3.5 m/s. Their performance suffers from poor projection of air within the room unless equipped with deflector plates which direct the jet in a horizontal direction beneath a ceiling.

4.1.6 Swirl diffusers

Swirl inducing diffusers can be of circular outlet form fitted with adjustable deflector vanes which impart a strong swirl action to the flow to provide rapid diffusion of supply air into a room. They can be located at ceiling level and set to produce a horizontally projected radial jet, or can be adjusted to project air vertically downwards. They may be located at floor level as part of an appropriately designed system. Other designs of diffuser can provide a swirling air flow by the geometrical form of the aperture but without using deflector blades. These types may be linear or radial flow devices.

Swirl type diffusers provide a rapid diffusion of supply air into a room and are therefore appropriate for high load applications and also where a limited space is available.

4.1.7 Other types

Other types of diffuser can include those utilised as part of a user-controllable local air diffusion system such as a desk or work-station system. In this case it is important that the supply velocity does not exceed approximately 1 - 1.5m/s, and preferably that the user has control over the direction/velocity of the jet.

4.2 SYSTEM CONFIGURATION

The quality of the micro-climate within a room is dependent on the balance of the momentum flow in the supply air and the buoyancy forces generated by thermal transfer within the room. The geometrical configuration of a room and supply system and the magnitude of the thermal load also influence performance. The effects of these influences are very difficult to predict. In many cases the expected performance of a system is well known from past experience and established design procedures can be used. At other times where the expected performance is not known from previous experience then an alternative approach must be considered. Section 6 discusses this latter situation. Here, the main features of the various system configurations is discussed.

4.2.1 Ceiling mounted systems

The use of ceiling-mounted air terminal devices is the most common configuration. This arrangement allows high velocities in the incoming airstreams to be dissipated above the occupied zone and usually allows a regular pattern of terminal layout that provides uniform air distribution. Flat ceiling surfaces also provide opportunity to take advantage of the Coanda effect to maintain horizontal flow even with cool supply air. It is normally convenient to use the ceiling void to accommodate the supply or extract ductwork or as a supply or extract plenum. It is often possible to allow considerable flexibility in the layout of room partitioning while still maintaining satisfactory air distribution.

Generalised design procedures for both circular and linear ceiling diffuser application are presented in reference 16.

4.2.2 Side-wall mounted grilles

The use of side-wall grilles is appropriate where ceiling-mounting is not practicable and where suitable ductwork connections are possible. A typical arrangement is where the supply ductwork is located in a false ceiling over a corridor and branches are connected to grilles situated high in the internal side-walls of the adjoining rooms. With cool air supply, it is important to ensure that the incoming airstream does not drop prematurely into the occupied zone and so cause discomfort. Initial upward deflection of the supply jet can alleviate this problem. The application of linear grilles close to ceiling surfaces will take advantage of the Coanda effect and help to maintain horizontal flow.

Reference 14 includes a generalised design procedure for this type of application.

4.2.3 Sill mounted grilles

Air supplies through sill mounted grilles are usually associated with perimeter-mounted fan-coil and induction units. In such applications, air is discharged vertically upwards from sill level until it reaches the ceiling, where it is deflected inwards across the room. The penetration of the air into the room is critically dependent on the relationship between the velocity and temperature of the air as it meets the ceiling.

Figure 13 shows the trajectories of cool air streams in diagrammatic form in both unsuccessful and successful design. At too low a discharge velocity a negative buoyancy force will overcome the momentum of the supply air and the jet will fail to reach the ceiling and will fall back causing high velocities and low temperatures near the window sill. At too high a discharge velocity the jet will attach to the ceiling along the full length and descend to floor level at the opposite wall resulting in high local air velocities at floor level. If the temperature differential is too great then the supply jet will detach from the ceiling due to the magnitude of the negative buoyancy force and cause high air velocities and low temperatures in the occupied zone. An ideal trajectory is where the jet projects beneath the ceiling and does not drop into the occupied zone until the velocities and temperatures are within the comfort limits. To achieve this, it is necessary to make use of the Coanda effect to maintain a horizontal flow direction as far as possible across the room.

Care is needed to ensure that there is no interference to the supply air stream by curtains or blinds and to avoid the potential for obstruction by books or papers resting on the sill.

In reference 17 there is a generalised design procedure for sill-mounted systems.

4.2.4 Floor mounted systems

Floor systems offer the advantage that they are designed to directly condition the occupied zone of the room. An additional feature is that because the main direction of air movement corresponds to that induced by heat release from occupants and equipment the air movement patterns are significantly more stable than when using other systems. For equivalent stability when using floor mounted systems the air change rate need only be approximately half that required when using a ceiling level system.

Under cooling applications a proportion of heat gains within the space can with appropriate design be allowed to convect into the upper part of the room above head height. At this level higher temperatures can be allowed to prevail without necessarily causing discomfort in the occupied zone. A major benefit for cooling systems is in the energy savings from a reduced effective thermal load on the occupied space. During winter heating conditioned air is supplied directly into the occupied zone thus overcoming the disadvantage of some ceiling mounted supply systems, i.e. an inability to project heated air down into the occupied zone and produce sufficient air movement.

Since air is being supplied directly within the occupied zone it is necessary to minimise the discharge velocity and ensure also that under cooling the temperature of the air is not too low. A discharge velocity of 2 - 4m/s when using swirl outlets is appropriate. This means that local to the terminals the thermal comfort criteria will not be met. This should be recognised when positioning outlets to ensure that staff who will be engaged in a mainly sedentary activity are not likely to be closer than approximately 1.5 m. The lower limit of the supply air temperature should be restricted to 16 to 18°C. Swirl type diffusers are appropriate for floor mounted systems in that they will rapidly diffuse the momentum of the supply air into the room.

4.2.5 Local supply systems

Local supply air systems are those which are integrated into seating arrangements such as in theatres or courtrooms or are incorporated into desks or work-stations. They are becoming quite widely used in Europe, particularly in West Germany. A first requirement though is that the seating or desk arrangement should be pre-determined and ideally fixed. In the theatre application, air can be supplied to the occupants from the backs of the seats in front of them. The angle of discharge of the supply jet is important and should be set to avoid impingement on the back of the head of the occupant in the active supply seat and to provide a comfortable environment for the occupant in the seat immediately to the rear. The discharge velocity should be in the range 1 to 1.5 m/s and the temperature not less than 16-18°C.

As an alternative in a theatre or lecture hall, low velocity air may be supplied at floor level beneath the seat. In this case it is necessary to ensure that the discharge velocity does not exceed 0.5 m/s and the supply temperature is not less than 18°C for the avoidance of draughts.

In an office environment some flexibility can be provided by modular design of systems with desk mounted air supply terminals and an appropriate floor layout using removable floor tiles over a floor plenum. For successful operation the supply air conditions should be similar to the requirements discussed above. In practice a combination of desk outlets and floor grilles will be the appropriate way of conditioning a space. In many cases it is advantageous to provide the occupant with some control of air flow direction and/or velocity. The velocity of air at the outlet of a local supply system at desk level should not exceed 1 m/s.

4.2.6 Ventilated ceilings

The supply of air through suspended ceilings may advantageously be applied particularly for rooms requiring large air flow rates. By discharging air through a large number of perforations or slots covering a wide area, substantial air flow rates can be introduced without the penetration of unacceptably high velocities into the occupied zone. Ventilated ceilings have successfully been used at flow rates of 100 ach or more and applications include computer rooms, laboratories as well as accommodation with sedentary occupation.

5. GOOD DESIGN PRINCIPLES AND PRACTICE

In the design of air distribution systems for human occupancy, the normal requirements are the provision of a thermal environment that meets the comfort criteria with respect to the level and uniformity of air temperatures and air velocities. In addition the effective provision of air flow rates and movement patterns to meet ventilation requirements is important.

In this section attention is drawn to those factors that should be considered in establishing good design practice.

5.1 ZONING ARRANGEMENT

In sub-dividing a space into air distribution zones, the following rules should be applied for ceiling, side-wall and sill mounted air terminal devices:-

each zone should be such that the horizontal distance in the direction of the supply air discharge from the terminal to the zone boundary is no more than 3 times the room height. This is because over greater distances a double rather than a single flow circulation pattern tends to form (see Figure 14).

for air terminal devices with a radial discharge pattern (e.g. circular or square ceiling diffusers), the plan aspect ratio of the zone should be no greater than 1.5, and the zone width to room height ratio should not be greater than 3.

for side-wall mounted grilles, the zone length (in the direction of supply air discharge) to zone width ratio should not be greater than 3.

5.2 PENETRATION OF SUPPLY AIRSTREAM

In selecting an air terminal device to serve a room zone, one of the main criteria is the throw of the supply jet in relation to the zone size. To achieve adequate air diffusion it is normal to design to produce a throw (to 0.5 m/s) equal to 3/4 of the distance in the direction of discharge from the terminal to the wall or zone boundary. Acceptable variation from this norm would be in the range 65 to 85% of the distance to the zone boundary.

Care should be taken to ensure that the drop of a cool supply airstream does not penetrate downwards into the occupied zone.

Manufacturers' data are available to enable appropriate air terminal device selection based on the throw and drop requirements at given air supply flow rates and temperatures.

5.3 VARIABLE FLOW RATE

Variable flow rate (known as variable air volume or VAV) systems allow advantage to be taken of the diversity of cooling loads from zone to zone in a building so economising on installed fan capacity. Additionally, major energy savings follow from reductions in fan power during times when the system is not off-setting maximum room heat gains. Since fan power is proportional to the cube of flow rate considerable savings can be achieved. Unfortunately, reducing flow rate alone may influence the quality of air distribution. This is particularly important if a fixed discharge-area terminal is used, since the impact of the supply air reduces as the square of discharge velocity. The combination of a substantial reduction in supply air velocity and, if reheat is used with a ceiling supply, the influence of buoyancy forces may result in poor air distribution. Under cooling conditions a reduced discharge velocity may result in the supply jet becoming detached from the ceiling (known as dumping) and causing high local velocities within the room. This may occur at supply velocities below 1.5 m/s. If it does occur, a significantly higher discharge velocity may be required to re-establish the normal flow pattern and during the intermediate condition discomfort due to high local velocities may be caused.

A generally preferred approach is to use a variable area air terminal device which is designed to maintain a substantially constant supply velocity at reducing air flow rates. However this type of device is not commonly used.

Holmes[18] has studied the operation of VAV systems and has described a design procedure for ceiling mounted systems used in cooling applications. The design methodology allows a consideration of both fixed and variable area devices. Tables are used to simplify the selection procedure.

5.4 HEATING AND COOLING APPLICATIONS

Due to buoyancy effects, air distribution in a room may well vary considerably depending on whether the incoming air is heated or cooled. For example, warm air from a ceiling diffuser will tend naturally to remain at high level unless the initial velocity is such that the warm air is forced downwards at the zone boundaries. However, if the higher supply velocity were still used when cool air is being introduced, the air would penetrate much further downwards and may cause unacceptably high velocities in the occupied zone. Where both winter and summer operation is required without the possibility of adjustment, a compromise between the optimum heating and cooling design solutions will have to be accepted.

5.5 PATTERN STAINING

Pattern staining is the discolouration of surfaces adjacent to supply terminals. It can result from dust particles in suspension in the room air being entrained in rapidly moving supply air and being deposited on surfaces adjacent to terminals. Equally, poor filtration of air taken from a dusty outdoor environment can also be responsible for accumulations of dirt on or close to the supply terminals. Pattern staining can be evident as bands or streaks of dark stain around ceiling diffusers set in smooth surfaces. Annular anti-smudge rings are effective in preventing staining in many applications. These are circular or rectangular frames attached to and extending 100-300mm beyond the edges of the diffuser. However, where highly textured ceiling surfaces are used, such as rough plaster, some types of ceiling tile or sprayed ceiling finishes then pattern staining can be expected to be more noticeable and difficult to remove.

5.6 INFLUENCE OF OBSTRUCTIONS

Design information applied to the performance of airstreams flowing adjacent to a surface assumes that the surface is smooth. In practice, obstructions due to ceiling beams or light fittings may be present, which, if closer to the supply air terminal than a certain critical distance will cause the airstream to be deflected from the ceiling and down into the occupied part of the room. Holmes and Sachariewicz[19] studied the influence of jet obstructions and defined a method of checking whether there may be a problem in a proposed design. A relationship was established between the height of the obstruction and the critical distance from the source of the jet, both parameters were made dimensionless by relating them to the initial depth of the jet (effective slot height), see Figure 15. For example, for an obstruction height three times the effective slot height the ratio of the critical distance (from the slot to the obstruction) to the slot height was found to be thirty. Relationships were also defined for predicting the maximum separation from the ceiling of the line of maximum velocity, and also the velocity decay of the jet as influenced by the obstruction.

5.7 DUCT ENTRY CONDITIONS

When selecting supply air terminal devices it is assumed that the upstream conditions of air velocity and pressure are uniform. In practice this is often not the case. Non-uniform duct entry conditions can lead to:

i)throw and spread not corresponding to manufacturer's data;

ii)failure to generate a surface-effect at a ceiling;

iii)high noise levels generated at the terminal;

iv)difficulties in obtaining reliable measurements during commissioning.

Recommendations to assist in good design of duct entry conditions for supply air terminal devices are described in detail in the HEVAC 'Air Diffusion Guide'[9].

5.8 POSITION OF EXTRACT GRILLES

The position of extract grilles is not too important in relation to the general circulation of air within a space. However, several factors may influence the choice of location.

For example, there are advantages in locating extract grilles at the part of the room towards which the air with the highest concentration of pollution or, in cooling applications, at the highest temperature, is likely to move. So, in the latter case, the extract locations should preferably be at high level whereas for the removal of pollutants of higher density than air, locations at low level would be preferable.

Extract grilles should not be located in a position that allows a significant amount of the supply air to be evacuated from the space before it has had chance to circulate through it. This 'short-circuiting' can occur if the extract grille is close to the supply terminal and in line with the direction of discharge. If such an alignment is unavoidable, the distance between the supply and extract locations should be sufficient to allow the airstream to room air temperature difference to decay by at least 80% of its original value.

5.9 LOW SUPPLY AIR FLOW RATES

Air movement in a room results from the relative magnitude of the momentum flow (multiple of mass flow rate and velocity) of the supply air and the buoyancy forces generated by energy transfers. Most design information relates to the situation where supply-to-room air temperature differences are in the range 5 K to 10 K, and where the supply air flow dictates the air movement pattern. Here, the case is considered where higher temperature differences up to 15 K are used, and where the supply air flow rates are correspondingly reduced.

Holmes and Caygill[20] studied this problem and identified for a range of heat loads the minimum supply momentum at which satisfactory air movement conditions could still be achieved. It was found that the ratio M/qH should not be less than 0.07 if the internal heat load is not to dominate the air movement, where

M=momentum flow of supply air (N) (= mass flow rate x velocity)

q=heat load (kW)

H=height of room (m)

In the report general comments were made relating to the performance of supply air diffusers. For example, it was found that circular diffusers tended to produce a more uniform velocity field than linear diffusers, although the performance of the latter type was found to be less sensitive to flow rate changes.

5.10 COLD WINDOWS

The use of ceiling mounted supply systems involving the discharge of warm air horizontally beneath the ceiling can result in poor air distribution if appropriate design procedures are not adopted. The design requirements are particularly critical when considering the influence of a cold window; in this case the air distribution system may be unable to overcome or minimise the influence of the downward flow of cold air.

Where it is not feasible to counteract the influence of the cold window by under-sill heating it is necessary to pay close attention to the design of the air distribution system. Holmes[21] has illustrated an appropriate design procedure for this application based on a theoretical extension of experimental work. An aim of the design method was to ensure that the temperature difference between the main body of the room and the area influenced by the cold window did not exceed 2 K. It was confirmed in this application that the throw of a ceiling diffuser to a terminal velocity of 0.5 m/s should be to 3/4 of the distance to the opposing wall.

It was also found beneficial to discharge all or at least some of the heated supply air towards the perimeter.

This work has been updated to cover a wider range of conditions and a new Technical Note[22] has been published by BSRIA.

5.11 COMMISSIONING

Commissioning is defined as "the advancement of an installation from the stage of static completion to full working order to specified requirements". Of most relevance to the air distribution performance is the regulation of the air flow throughout the air supply and extract systems.

The CIBSE (then IHVE) have published a Commissioning Code[23] for air distribution that outlines the principles and procedures to be followed. The BSRIA Manual for Regulating Air Conditioning Installations (Application Guide 3/89[24]) is a comprehensive, updated handbook giving practical guidance on the application of the CIBSE Code. It includes information on suitable instrumentation and its use.

The importance of the proper provision for commissioning in system design and the effective application of the regulation procedures following installation cannot be over emphasised.

6. DESIGN EVALUATION FOR CRITICAL APPLICATIONS

Situations arise where either standard design methods prove inadequate for the task in hand, or due to the critical nature of the application, additional confidence in a design needs to be developed. In these situations modelling the application and assessing the resulting air velocity and temperature distributions can provide a greater assurance of the appropriateness of the design and allow optimisation to take place before the design is finalised. Two options are available for modelling - physical modelling and numerical modelling.

6.1 PHYSICAL MODELLING

BSRIA have been involved in physical modelling of room air movement over many years encompassing various applications. The procedure is usually to build in the laboratory a representative module of the proposed building, and operate the model under design conditions so that a detailed investigation of room air distribution performance can be undertaken.

Where the building comprises individual rooms then a representative size and orientation of room would be selected. Alternatively, in a predominantly open plan building a test module would be identified based on the modularity of layout of the room air distribution system. Usually, aerodynamic boundaries to a module can be identified and represented in the laboratory model as physical walls. Plant and air distribution equipment are incorporated so that specified air supply configurations, flow rates, and temperatures can be provided and the performance of the system monitored. It is important to apply appropriate thermal loads and boundary conditions to represent heat transfer rates, from electrical equipment and across external walls for example.

The size of a building may preclude full-scale laboratory modelling, for example atria, stadia, theatres and concert halls; in these cases reduced-scale investigations may be carried out. The scale factor (length scale of model divided by length scale of building) should be kept as large as possible but scale factors as small as 1/12 have been employed in the past. The data obtained from the reduced-scale model is related to the full-size building according to established scaling laws.

Visualisation of air movement patterns can be undertaken using smoke as a tracer. The detailed measurements of air speeds and temperature distributions obtained using sophisticated instrumentation are interpreted using the appropriate thermal comfort standards and guides and current comfort research findings.

The influence of any refinements to the design which are deemed necessary can readily be studied at the modelling stage.

6.2 NUMERICAL MODELLING

Numerical modelling of air movement in rooms involves the use of computer software in which the fundamental conservation equations of momentum, energy and mass are applied to predict air speeds and temperature distributions [25]. The conservation equations which are of partial differential form, are integrated numerically over the flow field. This is a process of algebraically satisfying the physical laws over a large number of control volumes which together make up the room or enclosure under study. Work is underway at BSRIA and elsewhere in applying and validating these tools and making them more specific to building services. Sufficient progress has been made such that, if used with care, they can provide both qualitative and quantitative information during the design process and allow rapid study of alternative design solutions or various operational conditions. Figure 16 shows an example of the results of an analysis of the velocity and temperature distributions in a naturally ventilated atrium.

Whereas previously the use of numerical modelling software was limited to a few experts, systems are now becoming available[26] that are purposely tailored for routine use by building services designers.

7. EVALUATION OF INSTALLED SYSTEMS - SITE INVESTIGATIONS

It is sometimes necessary to make a detailed assessment of the performance of an air distribution system in a real building using appropriate instrumentation and experimental procedures.

One difficulty of studying air movement in real buildings is the lack of control that the investigator has over the thermal conditions. In the main it is possible only to study the operation of the system under the thermal loads prevailing at the time of the investigation although in some cases it is possible to impose additional heat loads by adding electric heaters or to deceive the plant into providing cooling or heating by, for example, adjusting thermostat settings. Because of the strong influence that thermal load can have on room air distribution it is important to note weather conditions, occupancy and other thermal gains at the time of the investigation together with any manual adjustments made to the control settings. The investigator must also be aware of the dynamics of the systems since the influence of a sudden and substantial changes will propagate through the system in a short time period. Continuous monitoring of conditions throughout the investigation must be maintained. The system air flow rate is often highly relevant to an investigation of the thermal environment although in many installations it can be very difficult to measure accurately. Techniques such as pitot traverses in the ductwork, calibrated air flow hoods, tracer gas injection and monitoring, and other procedures may be used.

Visualisation of air movement patterns is often appropriate using a hand held smoke puffer. This can readily indicate directions of air movement but care should be taken to ensure that the person holding the smoke puffer or anyone standing close by does not influence the local air movement. Smoke bombs or large smoke generators are only appropriate for use if the building or room under investigation is not occupied. They also tend to produce so much smoke that the visual indications of any underlying air movement pattern may be quickly obscured. Any buoyancy influence from smoke produced by thermal reaction should be noted. The use of still photography supported by hand written notes and/or the use of video photography can be a means of providing a permanent record of observations. A sketch of air movement patterns on planes through the flow field are useful means of communicating the essence of flow patterns to others. Smoke as a tracer should be used with caution since it may trigger alarms in an automated fire detecting system.Hot film or hot wire thermal anemometers are suitable for air velocity measurements in the range associated with room air movement. Spherical headed probes are largely unaffected by the direction of flow and hence may be used without specific knowledge of the pattern of air motion. They operate with acceptable resolution and repeatability in the range above 0.05 m/s. Below 0.05 m/s convection due to heat transfer from the sensor can induce spurious air movement signals. It is permissible to use a portable hand-held anemometer for an initial assessment of conditions but where detailed information is required then a more intensive approach should be considered. Air velocity readings at between three to five heights from ankle level (0.1 m) to head level (1.8m) will give adequate coverage of the occupied height. A plan grid spacing of 1 m to 2 m, or greater if the zone studied is large, will give acceptable spatial resolution.

Air movement in a mechanically ventilated room is turbulent, having a large amplitude and a fairly low frequency dominant fluctuation. It is therefore necessary to measure air speeds over a suitable time period, normally not less than 2 minutes to obtain a reliable mean value. If discrete sampling is used, which with a data acquisition system is generally the more convenient approach, then a sampling period increment of 1 or 2 seconds is appropriate. It is often convenient to measure temperature distribution in the space at the same physical locations as air speed. However temperature can be obtained by a single snapshot measurement rather than using the long integration time period necessary for air speed measurements. Thermocouples connected to a data acquisition system can give a fairly quick and accurate (+/-0.2 K) measure of temperature. Other types of sensor are also readily available for this application, sometimes they can be obtained as part of the anemometer system. Air temperature sensors can be protected from radiation by using small shields made from aluminium foil or aluminium coated tape. Dry resultant temperature as used to define comfort temperature in the CIBSE Guide[2] can be measured directly using a 100mm diameter globe thermometer.

Radiant temperature can be obtained by combining the information from a globe thermometer with air temperature and air speed measurements. A globe thermometer has a time constant of several minutes which must be taken into account to minimise errors.

The humidity of the air can be obtained from measurements made with a sling hygrometer. However, where long-term monitoring is being undertaken an installed instrument is likely to be more appropriate. Humidity is a difficult parameter to measure with consistently high accuracy so great care should be taken in the selection and maintenance of instruments for this purpose.

More general information on instruments, their care and use is presented in BSRIA Technical Note 14/86 "Instruments for building services applications"[27]. That publication includes information on the suitability of various types of instrument particularly for site use. Among others, instruments for the measurement of air velocity, volume flow rate, temperature and humidity are covered.

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Table 1 - Rise or drop of horizontally projected air jet from a 50mm nozzle*

Diameter of supply nozzle	Supply velocity	Supply room air temperature	Rise or drop (mm) of air jet at selected distances from air terminal device(m)							
(mm)	(m/s)	(K)	0.5	1	1.5	2	2.5	3	4	5
50	1.5	2	0	30	100	250	480	830	1970	
	1.5	4	10	60	210	490	980	1660		
	1.5	6	10	90	310	740	1440	2490		
	1.5	8	20	120	420	980	1920			
	1.5	10	20	150	520	1230	2400			
	1.5	12	20	180	620	1480	2880			
	1.5	15	30	230	780	1840				
	1.5	20	40	310	1040	2460				
50	2	2	0	20	60	140	270	470	1110	2160
	2	4	0	30	120	280	540	930	2210	
	2	6	10	50	180	420	810	1400		
	2	8	10	70	230	550	1080	1870		
	2	10	10	90	290	690	1350	2330		
	2	12	10	100	350	830	1620	2800		
	2	15	20	130	440	1040	2030			
	2	20	20	170	580	1380	2700			
50	3	2	0	10	30	60	120	210	490	960
	3	4	0	20	50	120	240	420	980	1920
	3	6	0	20	80	180	360	620	1480	2880
	3	8	0	30	100	250	480	830	1970	
	3	10	0	40	130	310	600	1040	2460	
	3	12	10	50	160	370	720	1250	2950	
	3	15	10	60	190	460	900	1560		
	3	20	10	80	260	610	1200	2080		
50	4	2	0	0	10	30	70	120	280	540
	4	4	0	10	30	70	140	230	550	1080
	4	6	0	10	40	100	200	350	830	1620
	4	8	0	20	60	140	270	470	1110	2160
	4	10	0	20	70	170	340	580	1380	2700
	4	12	0	30	90	210	410	700	1660	
	4	15	0	30	110	260	510	880	2080	
	4	20	10	40	150	350	680	1170	2770	
50	5	2	0	0	10	20	40	70	180	350
	5	4	0	10	20	40	90	150	350	690
	5	6	0	10	30	70	130	220	530	1040
	5	8	0	10	40	90	170	300	710	1380
	5	10	0	10	50	110	220	370	890	1730
	5	12	0	20	60	130	260	450	1060	2080
	5	15	0	20	70	170	320	560	1330	2590
	5	20	0	30	90	220	430	750	1770	

* The data in this table also applies to a square or near square grille with an equivalent free area

Table 2 - Rise or drop of horizontally projected air jet from a 100mm nozzle*

Diameter of supply nozzle (mm)	Supply velocity (m/s)	Supply room air temperature (K)	Rise or drop (mm) of air jet at selected distances from air terminal device (m) 0.5	1	1.5	2	2.5	3	4	5
100	1.5	2	0	20	50	120	240	420	980	, 1920
	1.5	4	0	30	100	250	480	830	1970	
	1.5	6	10	50	160	370	720	1250	2950	
	1.5	8	10	60	210	490	960	1660		
	1.5	10	10	80	260	610	1200	2080		
	1.5	12	10	90	310	740	1440	2490		
	1.5	15	10	120	390	920	1800			
	1.5	20	20	150	520	1230	2400			
100	2	2	0	10	30	70	140	230	550	1080
	2	4	0	20	60	140	270	470	1110	2160
	2	6	0	30	90	210	410	700	1660	
	2	8	0	30	120	280	540	930	2210	
	2	10	10	40	150	350	680	1170	2770	
	2	12	10	50	180	420	810	1400		
	2	15	10	60	220	520	1010	1750		
	2	20	10	90	290	690	1350	2330		
100	3	2	0	0	10	30	60	100	250	480
	3	4	0	10	30	60	120	210	490	960
	3	6	0	10	40	90	180	310	740	1440
	3	8	0	20	50	120	240	420	980	1920
	3	10	0	20	60	150	300	520	1230	2400
	3	12	0	20	80	180	360	620	1480	2880
	3	15	0	30	100	230	450	780	1840	
	3	20	0	40	130	310	600	1040	2460	
100	4	2	0	0	10	20	30	60	140	270
	4	4	0	0	10	30	70	120	280	540
	4	6	0	10	20	50	100	180	420	810
	4	8	0	10	30	70	140	230	550	1080
	4	10	0	10	40	90	170	290	690	1350
	4	12	0	10	40	100	200	350	830	1620
	4	15	0	20	50	130	250	440	1040	2030
	4	20	0	20	70	170	340	580	1380	2700
100	5	2	0	0	0	10	20	40	90	170
	5	4	0	0	10	20	40	70	180	350
	5	6	0	0	10	30	60	110	270	520
	5	8	0	10	20	40	90	150	350	690
	5	10	0	10	20	60	110	190	440	860
	5	12	0	10	30	70	130	220	530	1040
	5	15	0	10	40	80	160	280	660	1300
	5	20	0	10	50	110	220	370	890	1730

* The data in this table also applies to a square or near square grille with an equivalent free area

Table 3 - Rise or drop of horizontally projected air jet from a 200mm nozzle*

Diameter of supply nozzle (mm)	Supply velocity (m/s)	Supply room air temperature (K)	Rise or drop (mm) of air jet at selected distances from air terminal device (m) 0.5	1	1.5	2	2.5	3	4	5
200	1.5	2	0	10	30	60	120	210	490	960
	1.5	4	0	20	50	120	240	420	980	1920
	1.5	6	0	20	80	180	360	620	1480	2880
	1.5	8	0	30	100	250	480	830	1970	
	1.5	10	0	40	130	310	600	1040	2480	
	1.5	12	10	50	160	370	720	1250	2950	
	1.5	15	10	60	190	460	900	1560		
	1.5	20	10	80	260	610	1200	2080		
200	2	2	0	0	10	30	70	120	280	540
	2	4	0	10	30	70	140	230	550	1080
	2	6	0	10	40	100	200	350	830	1620
	2	8	0	20	60	140	270	470	1110	2160
	2	10	0	20	70	170	340	580	1380	2700
	2	12	0	30	90	210	410	700	1660	
	2	15	0	30	110	260	510	880	2080	
	2	20	10	40	150	350	680	1170	2770	
200	3	2	0	0	10	20	30	50	120	240
	3	4	0	0	10	30	60	100	250	480
	3	6	0	10	20	50	90	160	370	720
	3	8	0	10	30	60	120	210	490	960
	3	10	0	10	30	80	150	260	610	1200
	3	12	0	10	40	90	180	310	740	1440
	3	15	0	10	50	120	230	390	920	1800
	3	20	0	20	60	150	300	520	1230	2400
200	4	2	0	0	0	10	20	30	70	140
	4	4	0	0	10	20	30	60	140	270
	4	6	0	0	10	30	50	90	210	410
	4	8	0	0	10	30	70	120	280	540
	4	10	0	10	20	40	80	150	350	680
	4	12	0	10	20	50	100	180	420	810
	4	15	0	10	30	60	130	220	520	1010
	4	20	0	10	40	90	170	290	690	1350
200	5	2	0	0	0	10	10	20	40	90
	5	4	0	0	0	10	20	40	90	170
	5	6	0	0	10	20	30	60	130	260
	5	8	0	0	10	20	40	70	180	350
	5	10	0	0	10	30	50	90	220	430
	5	12	0	0	10	30	60	110	270	520
	5	15	0	10	20	40	80	140	330	650
	5	20	0	10	20	60	110	190	440	860

* The data in this table also applies to a square or near square grille with an equivalent free area

Table 4 - Rise or drop of horizontally projected air jet from a 300mm nozzle*

Diameter of supply nozzle (mm)	Supply velocity (m/s)	Supply room air temperature (K)	Rise or drop (mm) of air jet at selected distances from air terminal device (m) 0.5	1	1.5	2	2.5	3	4	5
300	1.5	2	0	10.	20	40	80	140	330	640
	1.5	4	0	10	30	80	160	280	660	1280
	1.5	6	0	20	50	120	240	420	980	1920
	1.5	8	0	20	70	160	320	550	1310	2560
	1.5	10	0	30	90	200	400	690	1640	
	1.5	12	0	30	100	250	480	830	1970	
	1.5	15	0	40	130	310	600	1040	2460	
	1.5	20	10	50	170	410	800	1380		
300	2	2	0	0	10	20	50	80	180	360
	2	4	0	10	20	50	90	160	370	720
	2	6	0	10	30	70	140	230	550	1080
	2	8	0	10	40	90	180	310	740	1440
	2	10	0	10	50	120	230	390	920	1800
	2	12	0	20	60	140	270	470	1110	2160
	2	15	0	20	70	170	340	580	1380	2700
	2	20	0	30	100	230	450	780	1840	
300	3	2	0	0	0	10	20	30	80	160
	3	4	0	0	10	20	40	70	160	320
	3	6	0	0	10	30	60	100	250	480
	3	8	0	10	20	40	80	140	330	640
	3	10	0	10	20	50	100	170	410	800
	3	12	0	10	30	60	120	210	490	960
	3	15	0	10	30	80	150	260	610	1200
	3	20	0	10	40	100	200	350	820	1600
300	4	2	0	0	0	10	10	20	50	90
	4	4	0	0	0	10	20	40	90	180
	4	6	0	0	10	20	30	60	140	270
	4	8	0	0	10	20	50	80	180	360
	4	10		0	10	30	60	100	230	450
	4	12	0	0	10	30	70	120	280	540
	4	15	0	10	20	40	80	150	350	680
	4	20	0	10	20	60	110	190	460	900
300	5	2	0	0	0	0	10	10	30	60
	5	4	0	0	0	10	10	20	60	120
	5	6	0	0	0	10	20	40	90	170
	5	8	0	0	10	10	30	50	120	230
	5	10	0	0	10	20	40	60	150	290
	5	12	0	0	10		40	70	180	350
	5	15	0	0	10		50	90	220	430
	5	20	0	0	20		70	120	300	580

* The data in this table also applies to a square or near square grille with an equivalent free area

Table 5 - Rise or drop of horizontally projected air jet from a 500mm nozzle*

Diameter of supply nozzle (mm)	Supply velocity (m/s)	Supply room air temper-ature (K)	Rise or drop (mm) of air jet at selected distances from air terminal device (m) 0.5	1	1.5	2	2.5	3	4	5
500	1.5	2	0	0	10	20	50	80	200	380
	1.5	4	0	10	20	50	100	170	390	770
	1.5	6	0	10	30	70	140	250	590	1150
	1.5	8	0	10	40	100	190	330	790	1540
	1.5	10	0	20	50	120	240	420	980	1920
	1.5	12	0	20	60	150	290	500	1180	2310
	1.5	15	0	20	80	180	360	620	1480	2880
	1.5	20	0	30	100	250	480	830	1970	
										220
500	2	2	0	0	10	10	30	50	110	430
	2	4	0	0	10	30	50	90	220	650
	2	6	0	10	20	40	80	140	330	860
	2	8	0	10	20	60	110	190	440	1080
	2	10	0	10	30	70	140	230	550	
										1300
	2	12	0	10	40	80	160	280	660	1620
	2	15	0	10	40	100	200	350	830	2160
	2	20	0	20	60	140	270	470	1110	
500	3	2	0	0	0	10	10	20	50	100
	3	4	0	0	10	10	20	40	100	190
	3	6	0	0	10	20	40	60	150	290
	3	8	0	0	10	20	50	80	200	380
	3	10	0	0	10	30	60	100	250	480
	3	12	0	0	20	40	70	120	300	580
	3	15	0	10	20	50	90	160	370	720
	3	20	0	10	30	60	120	210	490	960
500	4	2	0	0	0	0	10	10	30	50
	4	4	0	0	0	10	10	20	60	110
	4	6	0	0	0	10	20	40	80	160
	4	8	0	0	10	10	30	50	110	220
	4	10	0	0	10	20	30	60	140	270
	4	12	0	0	10	20	40	70	170	320
	4	15	0	0	10	30	50	90	210	410
	4	20	0	0	10	30	70	120	280	540
500	5	2	0	0	0	0	0	10	20	30
	5	4	0	0	0	0	10	10	40	70
	5	6	0	0	0	10	10	20	50	100
	5	8	0	0	0	10	10	30	70	140
	5	10	0	0	0	10	20	40	90	170
	5	12	0	0	10	10	30	40	110	210
	5	15	0	0	10	20	30	60	130	260
	5	20	0	0	10	20	40	70	180	350

* The data in this table also applies to a square or near square grille with an equivalent free area

Table 6 - Penetration of a CIRCULAR heated air jet projected vertically downward (to a terminal velocity of zero)

Initial temperature difference	Initial velocity	Penetration (m) of jet of specified initial diameter (mm)				
°C	m/s	100	200	300	500	1000
5	2.5	3.4	4.7	5.7	7.3	10.0
5	5.0	6.8	9.6	11.7	15.0	21.0
5	10.0	13.8	19.4	23.7	30.5	42.5
5	15.0	20.7	29.2	36.4	46.6	-
5	20.0	27.6	40.0	48.8	-	-
10	2.5	2.4	3.3	4.0	5.0	6.8
10	5.0	4.8	6.7	8.2	10.5	14.5
10	10.0	9.7	13.7	16.7	21.4	30.0
10	15.0	14.6	20.6	25.2	32.4	45.2
10	20.0	19.5	27.5	33.7	43.9	-
15	2.5	1.8	2.6	3.2	4.0	5.4
15	5.0	3.9	5.5	6.6	8.5	11.7
15	10.0	7.9	11.1	13.6	17.4	24.4
15	15.0	11.9	16.8	20.5	26.4	36.4
15	20.0	15.9	22.4	27.4	35.6	49.5
20	2.5	1.5	2.3	2.7	3.4	4.5
20	5.0	3.4	4.7	5.7	7.3	10.0
20	10.0	6.8	9.6	11.7	15.0	21.0
20	15.0	10.3	14.5	17.7	22.8	31.9
20	20.0	13.8	19.4	23.7	30.5	42.5
30	2.5	1.2	2.1	2.2	2.7	3.5
30	5.0	2.7	3.8	4.6	5.9	8.0
30	10.0	5.6	7.8	9.5	12.2	16.9
30	15.0	8.4	11.8	14.4	18.5	25.9
30	20.0	11.2	15.8	19.3	24.8	34.1

Note - indicates a penetration greater than 50m.

Table 7 - Penetration of a LINEAR heated air jet projected vertically downwards (to a terminal velocity of zero)

Initial temperature differential °C	Initial velocity m/s	Penetration (m) of jet of specified initial thickness (mm) 5	7.5	10	15	20	30
5	2.5	2.4	2.8	3.1	3.5	3.9	4.4
5	5.0	6.1	7.0	7.7	8.8	9.7	11.1
5	10.0	15.4	17.6	19.4	22.2	24.4	28.0
5	15.0	26.4	30.2	33.3	38.1	41.9	48.0
5	20.0	38.7	44.3	48.8	-	-	-
10	2.5	1.5	1.7	1.9	2.2	2.4	2.8
10	5.0	3.8	4.4	4.8	5.5	6.1	7.0
10	10.0	9.6	11.0	12.2	13.9	15.3	17.5
10	15.0	16.5	18.9	20.9	23.9	26.3	30.1
10	20.0	24.3	27.8	30.6	35.0	38.6	44.1
15	2.5	1.2	1.3	1.5	1.7	1.8	2.1
15	5.0	2.9	3.3	3.7	4.2	4.6	5.3
15	10.0	7.3	8.4	9.2	10.6	11.6	13.3
15	15.0	12.6	14.4	15.8	18.1	20.0	22.8
15	20.0	18.4	21.1	23.2	26.6	29.3	33.5
20	2.5	1.0	1.1	1.2	1.4	1.5	1.7
20	5.0	2.4	2.7	3.0	3.4	3.8	4.3
20	10.0	6.0	6.9	7.6	8.7	9.6	10.9
20	15.0	10.3	11.8	13.0	14.9	16.4	18.8
20	20.0	15.1	17.3	19.0	21.8	24.0	27.5
30	2.5	0.7	0.8	0.9	1.0	1.1	1.3
30	5.0	1.8	2.1	2.3	2.6	2.9	3.3
30	10.0	4.5	5.2	5.7	6.1	7.2	8.3
30	15.0	7.8	8.9	9.8	11.2	12.4	14.2
30	20.0	11.4	13.1	14.4	16.5	18.2	20.8

Note - indicates a penetration greater than 50m.

Table 8 - Throw of CIRCULAR chilled air jet projected vertically downwards

Initial temperature differential °C	Initial velocity m/s	Throw (m) of jet of specified initial diameter(mm) to terminal velocities of 0.5 and 0.3 m/s							
		100mm		200mm		300mm		500mm	
		0.5	0.3	0.5	0.3	0.5	0.3	0.5	0.3
0	1.0	1.3	2.1	2.4	4.1	3.7	6.2	6.2	10.3
0	2.0	2.6	4.2	4.9	8.1	7.4	12.2	12.3	20.4
0	3.0	3.8	6.3	7.4	12.2	11.1	18.3	18.5	30.5
0	4.0	5.1	8.3	9.8	16.2	14.8	24.5	24.6	40.7
0	5.0	6.3	10.4	12.3	20.4	18.4	30.5	30.8	-
2	1.0	1.6	2.9	3.3	6.1	5.2	9.8	9.2	25.6
2	2.0	3.1	5.8	6.5	12.2	10.4	19.4	18.4	-
2	3.0	4.6	8.7	9.9	18.5	15.6	29.2	27.6	-
2	4.0	6.2	11.5	13.1	24.5	20.7	38.9	36.8	-
2	5.0	7.7	14.4	16.4	30.8	25.9	48.5	46.0	-
4	1.0	1.7	3.2	3.6	8.2	5.7	18.8	11.4	-
4	4.0	3.4	6.3	7.1	16.4	11.3	37.3	22.9	-
4	3.0	5.1	8.7	10.8	24.9	17.0	-	34.4	-
4	4.0	6.8	12.6	14.3	33.2	22.7	-	45.9	-
4	5.0	8.4	15.8	18.0	41.7	28.3	-	-	-
6	1.0	1.8	3.4	3.8	2.5	6.3	28.5	17.4	-
6	2.0	3.6	6.7	7.5	25.0	12.0	-	34.8	-
6	3.0	5.4	10.0	11.4	37.9	18.0	-	-	-
6	4.0	7.1	13.4	15.1	-	24.0	-	-	-
6	5.0	8.9	16.7	19.0	-	29.9	-	-	-
8	1.0	1.9	4.4	3.9	16.9	8.5	38.6	23.4	-
8	2.0	3.7	8.7	7.8	33.8	16.9	-	47.1	-
8	3.0	5.6	13.1	11.9	-	25.4	-	-	-
8	4.0	7.4	17.5	15.8	-	34.0	-	-	-
8	5.0	9.4	21.9	19.8	-	42.4	-	-	-

Note - indicates that the terminal velocity specified is not reached within a throw of 50m or the jet becomes buoyant before reaching the terminal velocity.

Table 9 - Throw of LINEAR chilled air jet projected vertically downwards

Initial temperature difference °C	Initial velocity m/s	Throw (m) of jet of specified initial thickness (mm) to terminal velocities of 0.5 and 0.3 m/s											
		5mm		7.5mm		10mm		15m		20mm		30mm	
		0.5	0.3	0.5	0.3	0.5	0.3	0.5	0.3	0.5	0.3	0.5	0.3
0	1.0	0.1	0.3	0.2	0.4	0.2	0.6	0.4	0.9	0.5	1.3	0.7	1.9
0	2.0	0.5	1.3	0.7	1.9	0.9	2.5	1.4	3.8	1.9	5.0	2.8	7.5
0	3.0	1.1	2.8	1.6	4.2	2.1	5.6	3.0	8.4	4.1	11.2	6.1	16.8
0	4.0	1.9	5.0	2.8	7.5	3.7	10.0	5.5	14.9	7.3	20.0	10.9	29.8
0	5.0	2.9	7.8	4.3	11.7	5.7	15.0	8.6	23.3	11.3	31.0	17.0	-
2	1.0	0.2	0.3	0.2	0.4	0.2	0.6	0.4	1.2	0.5	1.9	0.7	3.7
2	2.0	0.5	1.3	0.7	2.3	0.9	3.7	1.4	7.3	1.9	11.8	3.1	23.1
2	3.0	1.0	3.5	1.6	6.8	2.1	10.9	3.1	21.5	4.7	34.6	9.1	-
2	4.0	2.9	7.4	2.8	14.5	3.7	23.5	6.2	46.1	10.0	-	19.5	-
2	5.0	2.9	13.0	4.3	26.3	5.7	42.5	11.2	-	18.0	-	35.3	-
4	1.0	0.1	0.3	0.2	0.6	0.2	1.0	0.4	1.9	0.5	3.0	0.8	5.9
4	2.0	0.5	1.9	0.7	3.7	0.9	6.0	1.6	11.8	2.6	-	5.0	-
4	3.0	1.1	5.6	1.6	10.9	2.5	17.6	4.6	-	7.5	-	14.7	-
4	4.0	1.9	11.9	3.2	23.4	5.1	-	9.9	-	16.0	-	31.5	-
4	5.0	2.9	21.6	5.7	-	9.2	-	18.0	-	29.0	-	-	-

Note - indicates that the terminal velocity specified is not reached within a throw of 50m or the jet becomes buoyant before reaching the terminal velocity.

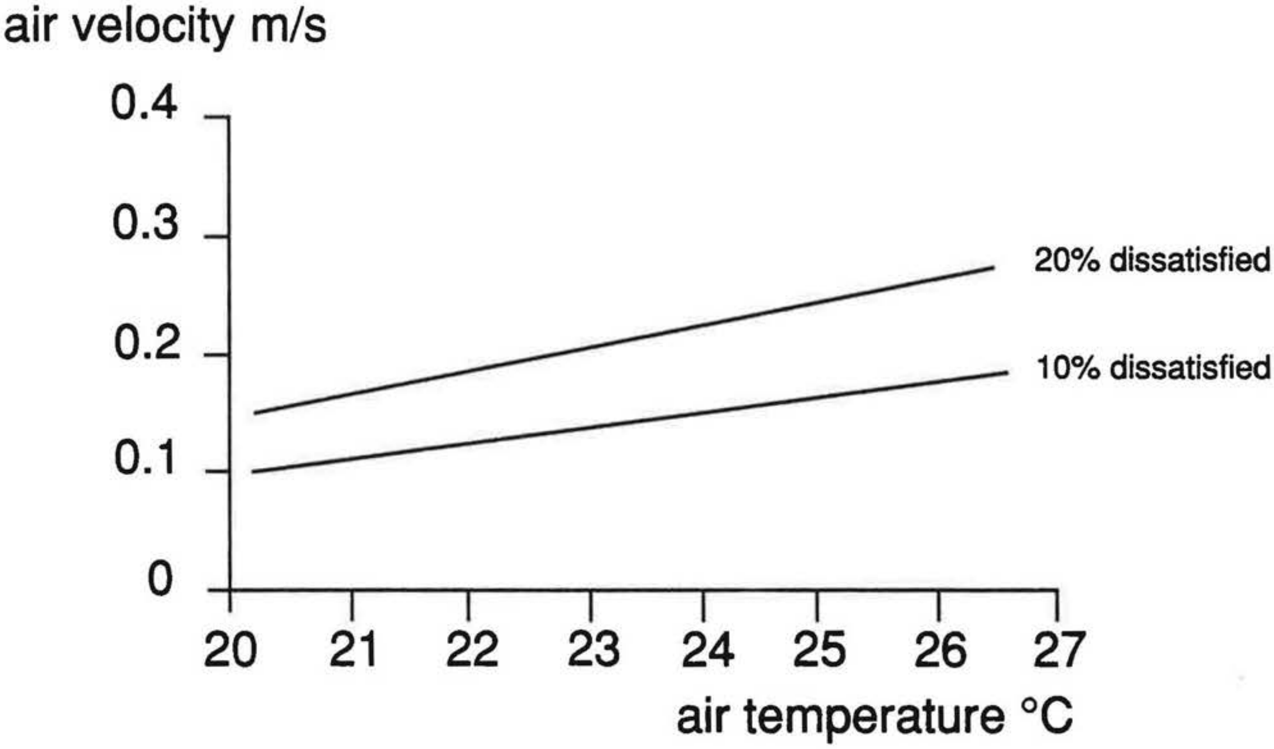


Figure 1 Velocities above which discomfort is likely at the indicated temperatures

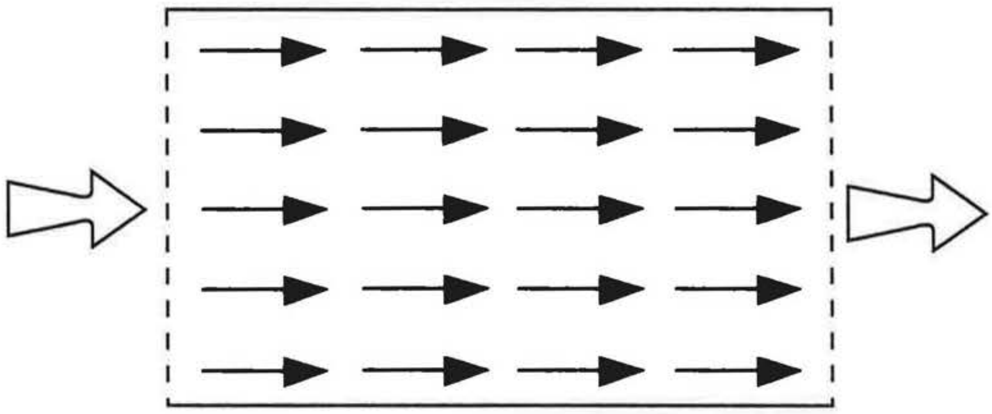


Figure 2 Displacement flow

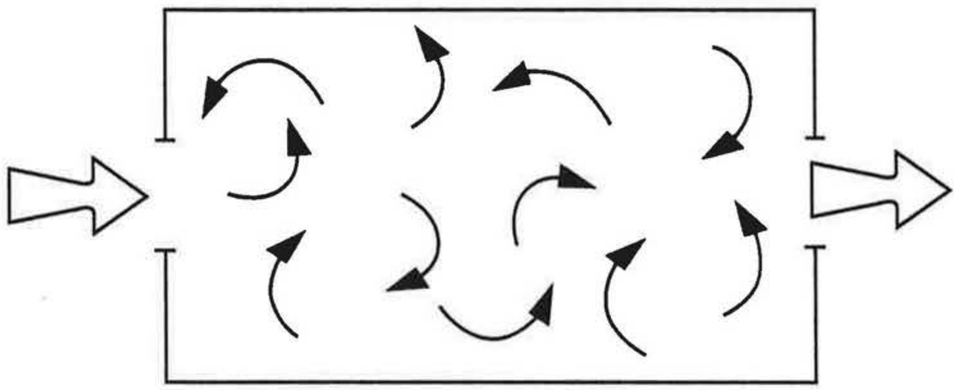


Figure 3 Mixed flow

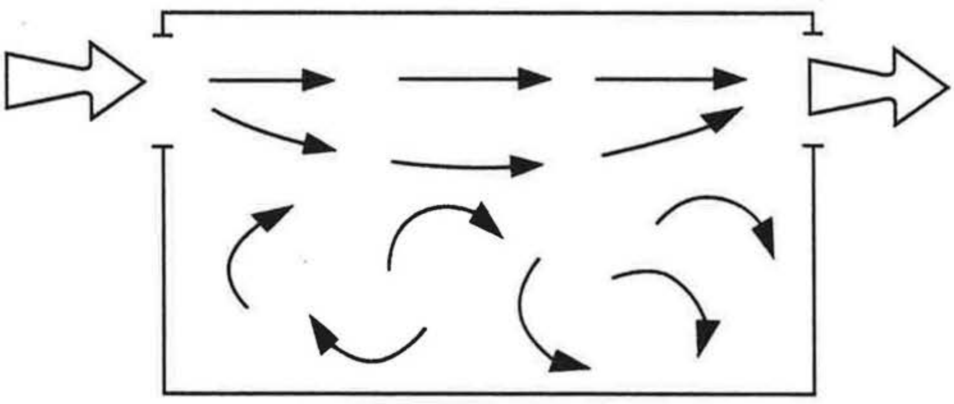


Figure 4 Short circuiting

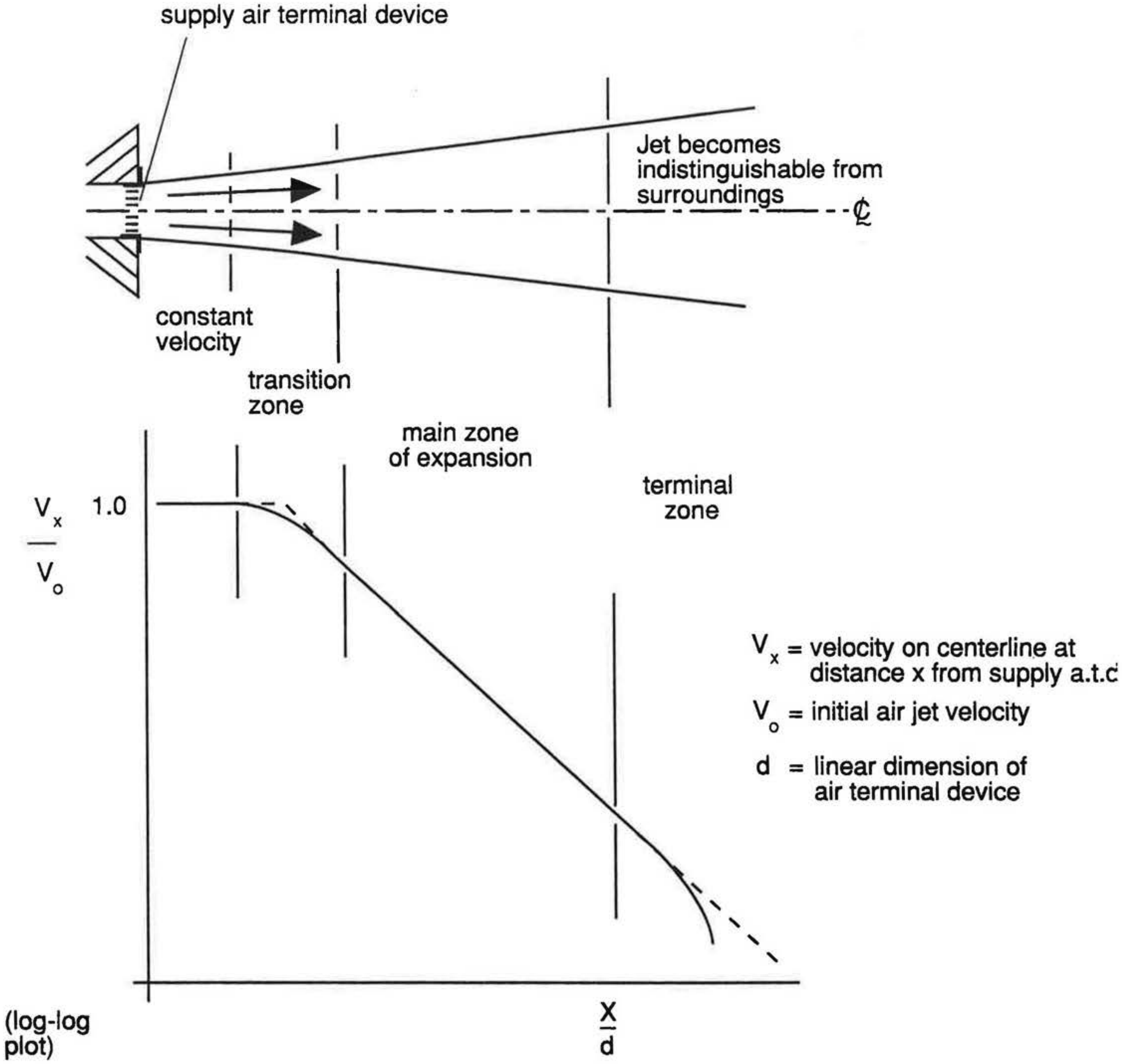


Figure 5 Velocity decay of a free jet

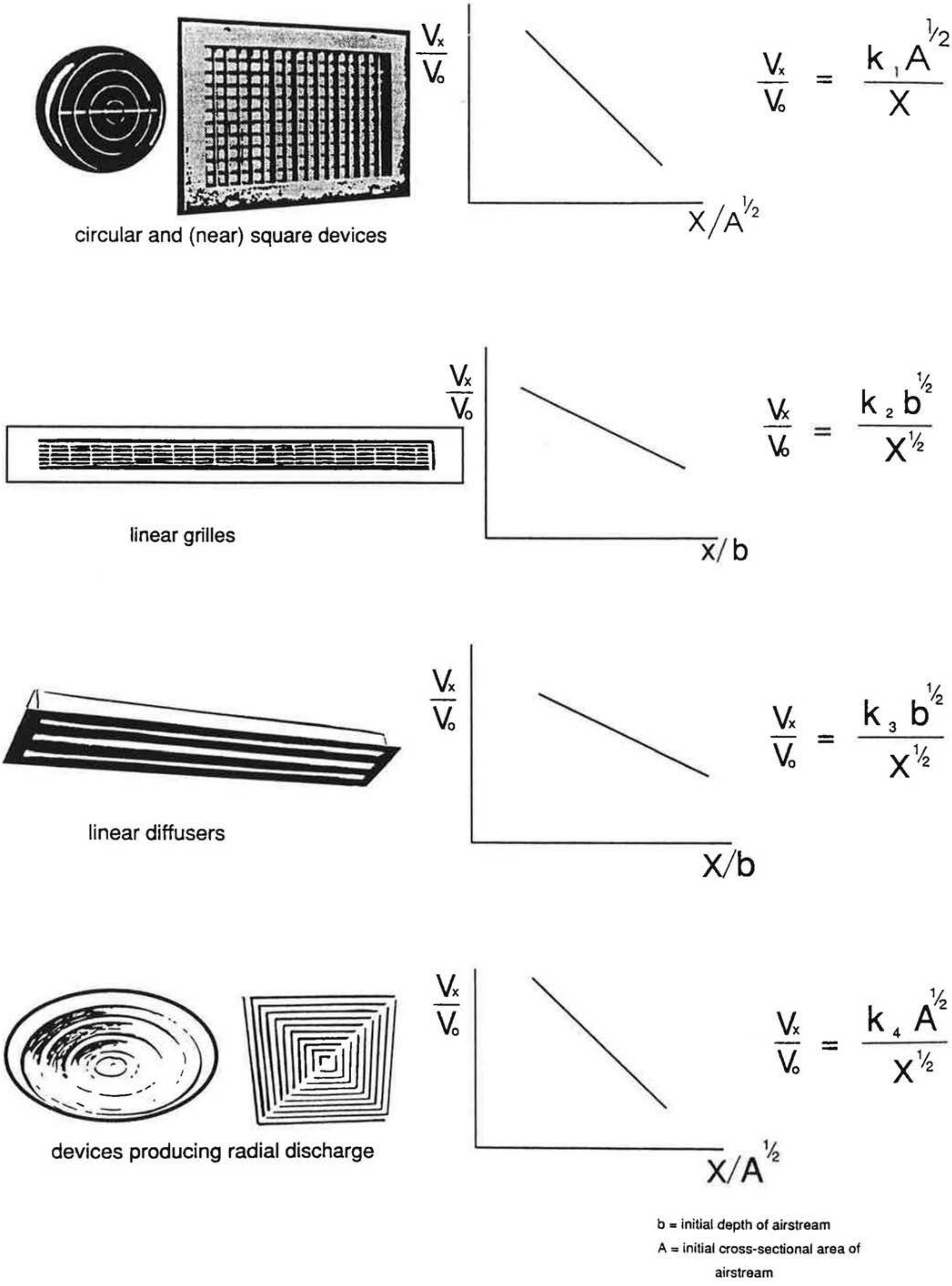


Figure 6 Velocity decay characteristics for different types of supply air terminal device

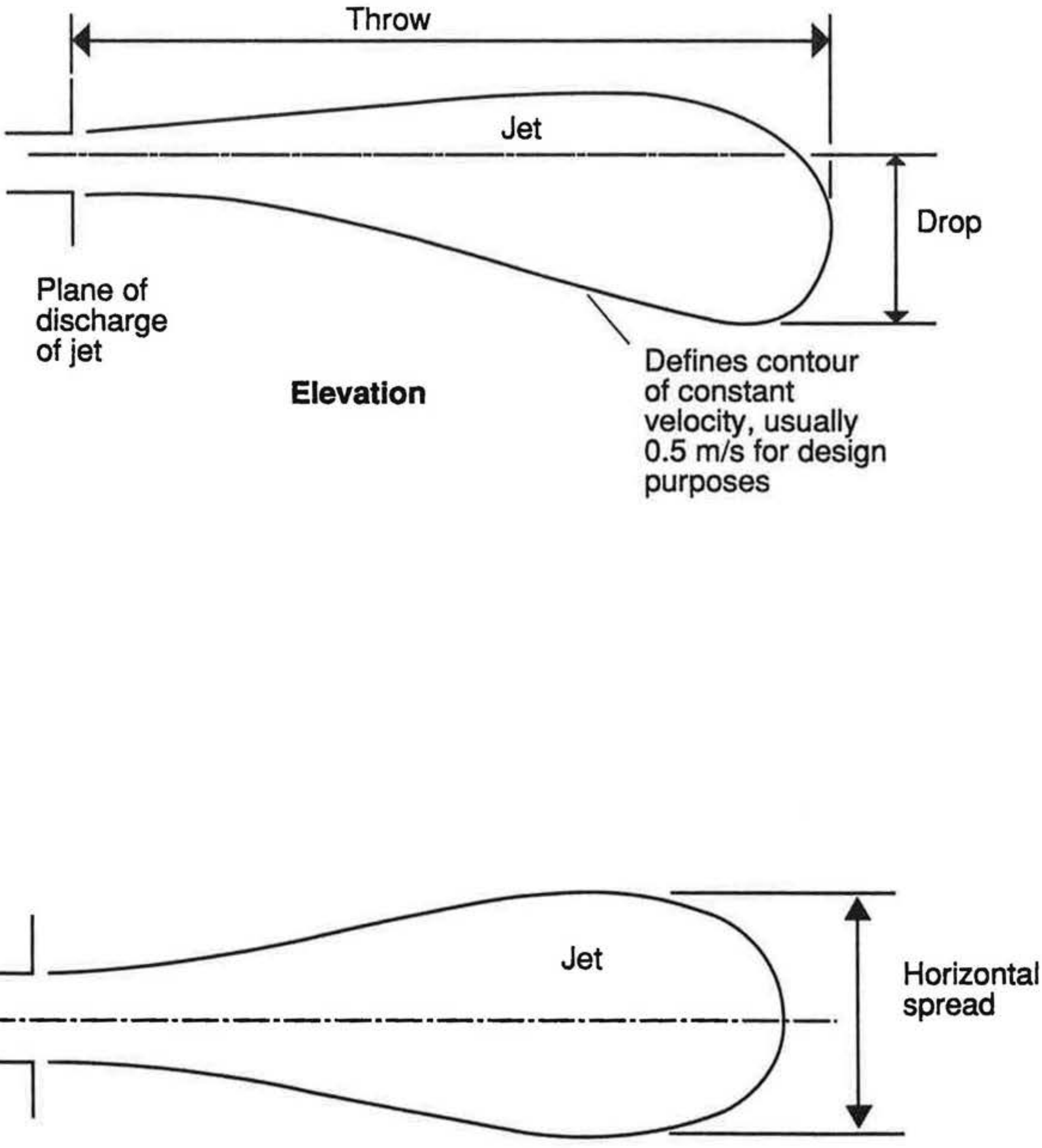


Figure 7 Throw, drop and spread of a cooled jet

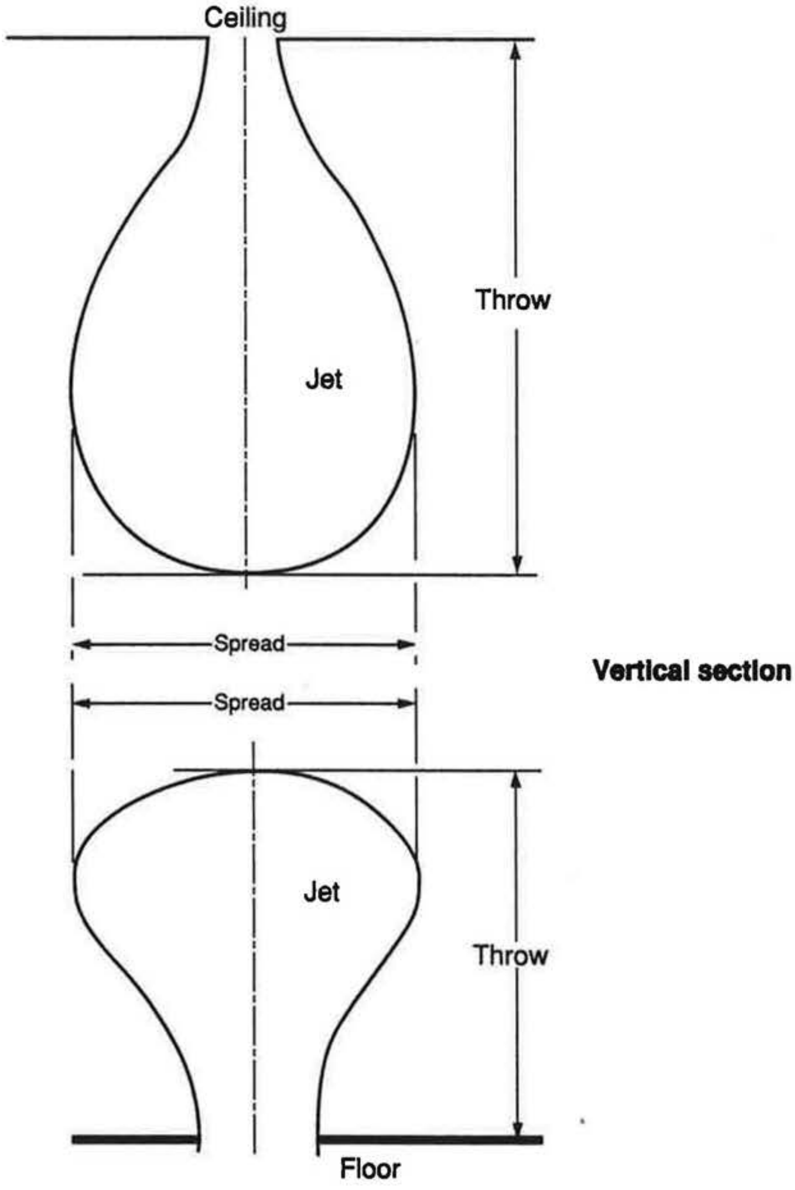


Figure 8 Throw and spread of a vertical jet

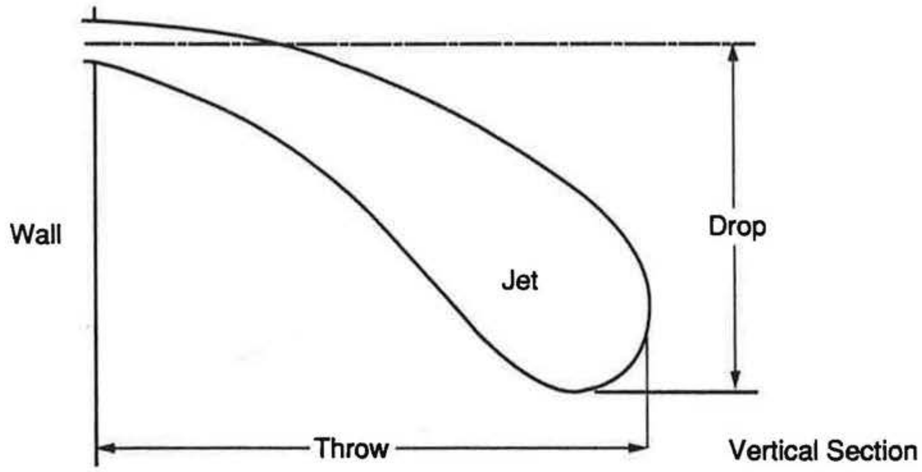


Figure 9 Throw and drop of a horizontally projected cool jet

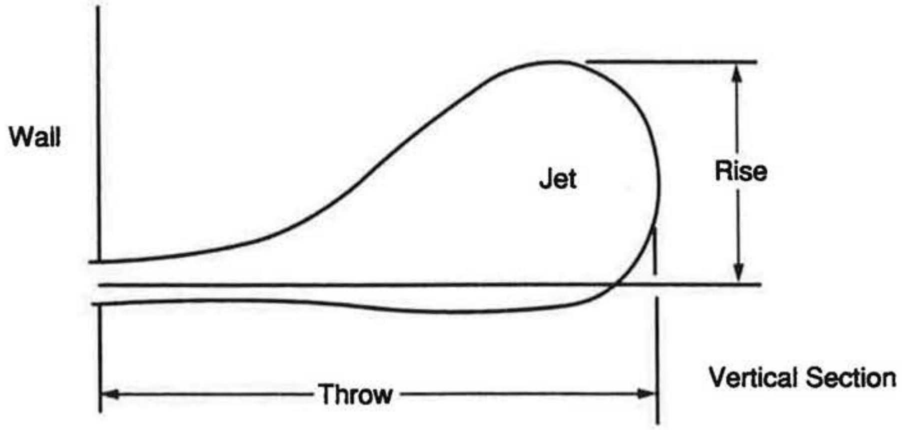


Figure 10 Throw and rise of a horizontally projected warm jet

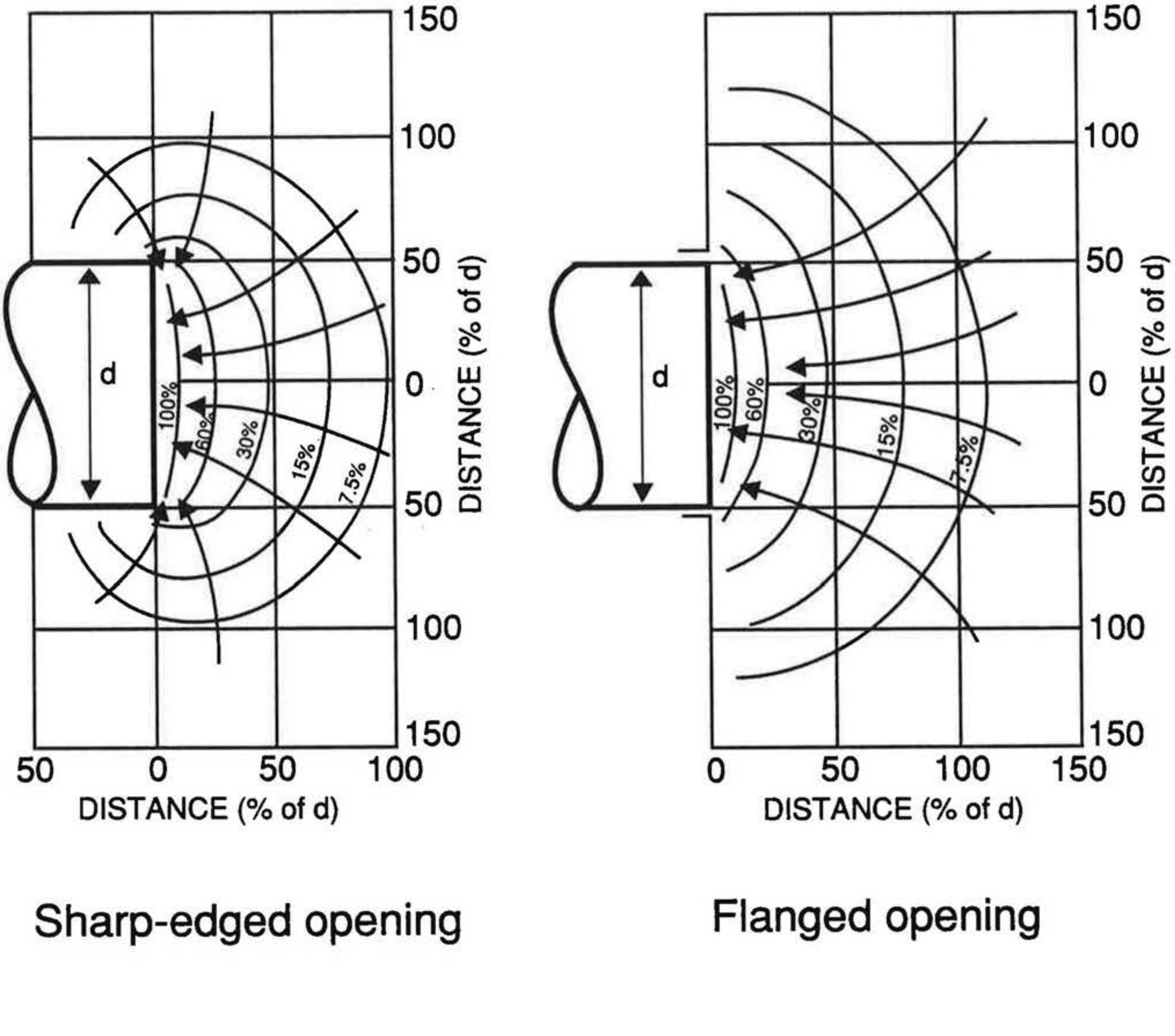


figure 11 Velocity contours for circular/exhaust opening (reproduced from CIBSE Guide 1986)

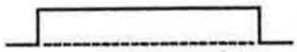
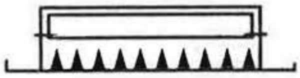
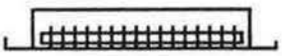
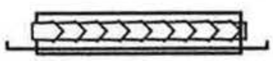
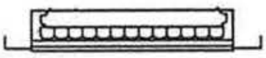
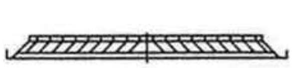
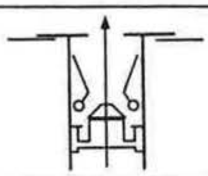
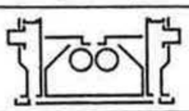
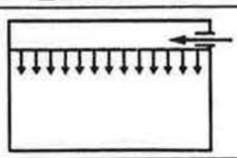
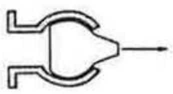
Type	Diagram	Application and location	Core velocity / (m/s) Quiet	Core velocity / (m/s) Commercially Quiet	Description and remarks
1. Perforated or stamped lattice		Supply Extract Transfer Ceiling Sidewall Floor	Up to 4	Up to 6	Simple form of grille with small free area. Alternatively can be used as supply diffuser with high air entrainment allowing large quantities to be diffused. For low level "laminar flow" panels to give displacement ventilation. velocity of 0-25 m/s is used.
2. Aerofoil blades - one row adjustable		Supply Extract Ceiling Sidewall Desk top	7	10	Frequently used grille with large free area. Directional control in one plane only for supply applications.
3. Aerofoil blades - two rows adjustable		Supply Sidewall	7	10	As 2 but with directional control in two planes.
4. Fixed blade		Supply Extract Ceiling Sidewall Floor Seat back	6	9	Robust grille with limited free area. Some directional control possible using profiled blades.
5. Non-vision		Extract Transfer Sidewall	7	10	Low free area. Designed to prevent through vision.
6. Egg crate		Extract Ceiling Sidewall	7	10	Generally largest free area grille available.
7. Fixed geometry diffusers		Supply Extract Ceiling Floor Desk top	7	10	Radial discharge diffusers offer good air entrainment allowing diffusion of large air quantities. Square or rectanuglar diffusers can provide 1, 2 or 3 way diffusion. Angled blades or slots can be used to apply twisting motion to supply air.
8. Adjustable diffusers		Supply Ceiling	4	6	As 7 but offers horizontal or vertical discharge. Can be thermostatically controlled.
9. Slot and linear diffusers		Supply Extract Ceiling Sidewall Desk top Under window	6	9	Offers vertical or horizontal discharge; single or multiple slots. Care must be taken with design of plenum box. Desk top units may incorporate induction of room air.
10. Air handling luminaires		Supply- Extract Ceiling	7	10	As 9 but single slot only. Normally used in conjunction with extract through luminaire.
11. Ventilated ceiling		Supply or Extract			Void above ceiling is pressurised to introduce air at low velocity through many single holes or through porous panels. Air entrainment is restricted and natural air currents may affect room air diffusion.
12. Nozzles, drum and punkah louvres		Supply Ceiling Sidewall Under window Seat back			Adjustable type shown can be rotating drum or swivelling ball, with or without nozzle jet for long throws and personal air supply/spot cooling. For high induction applications fixed multiple nozzles are used. Velocities depend on throw, noise and induction requirements.

Figure 12 Main types of air terminal devices (reproduced from CIBSE Guide 1986)

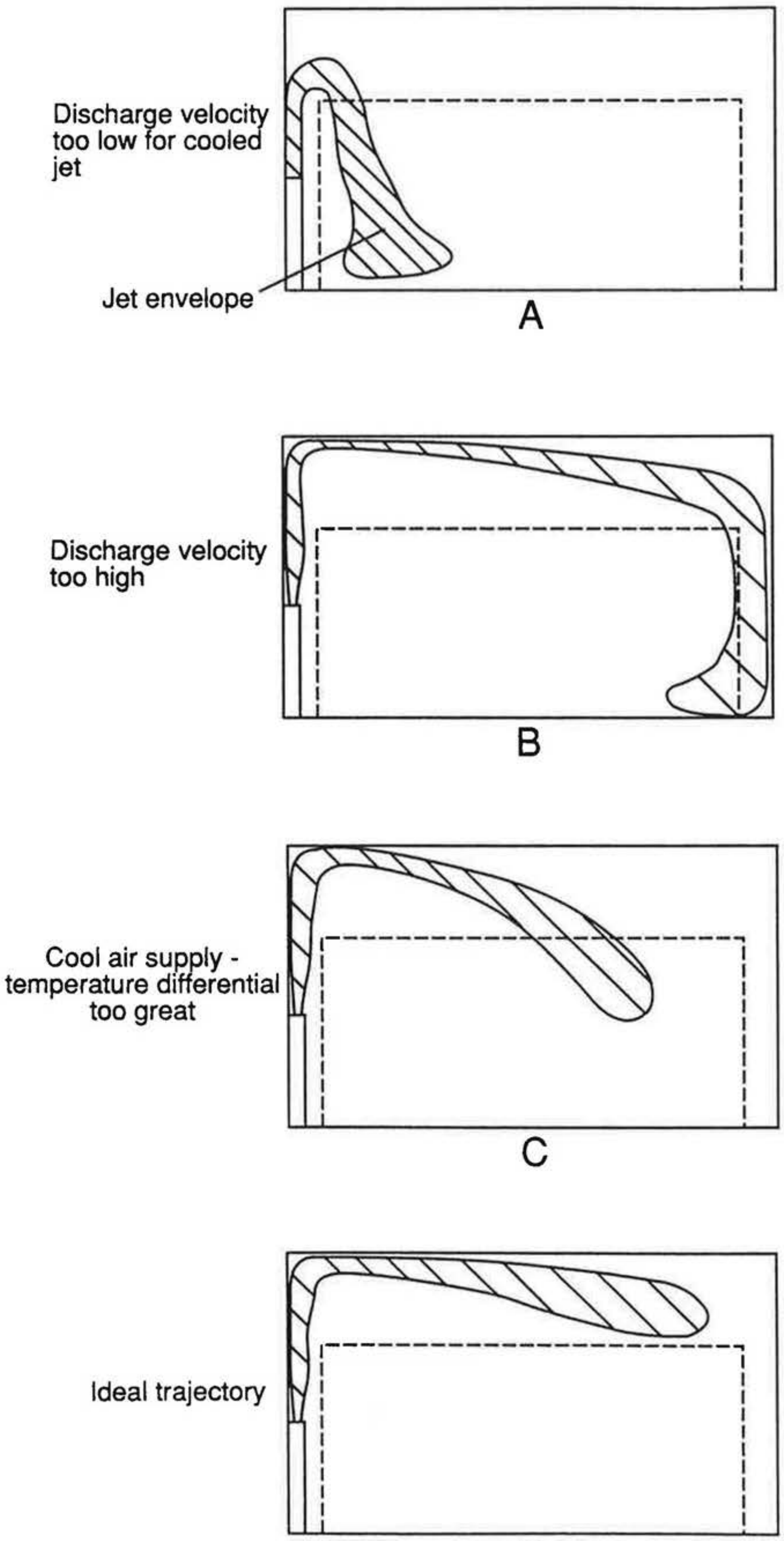
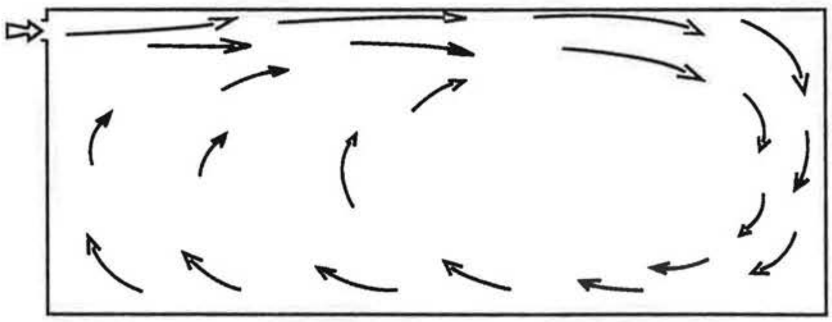
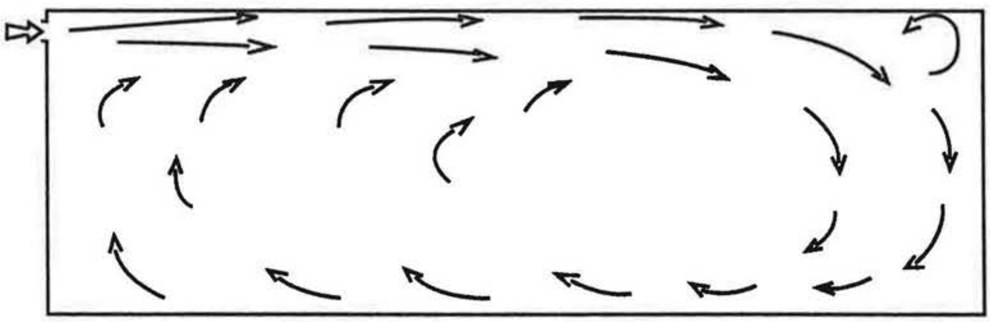


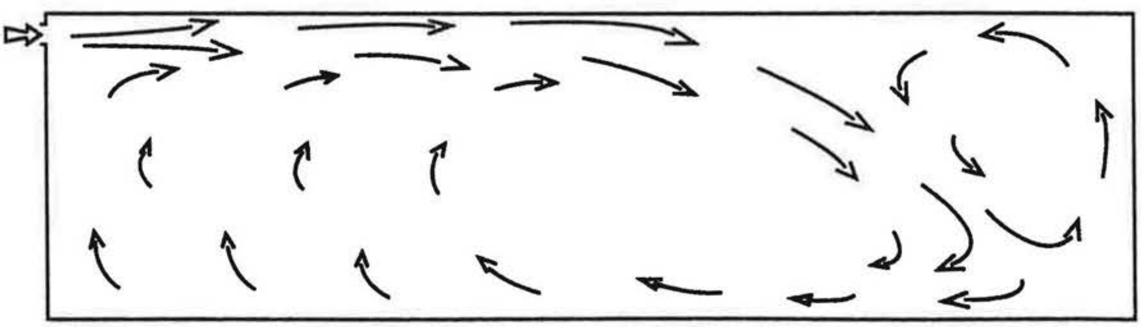
Figure 13 Trajectories of cool air streams from sill mounted grille



length: height ratio 2.5:1



length: height ratio 3:1



length: height ratio 3.5:1

Vertical sections

Figure 14 Flow patterns in zones of various length: height ratios

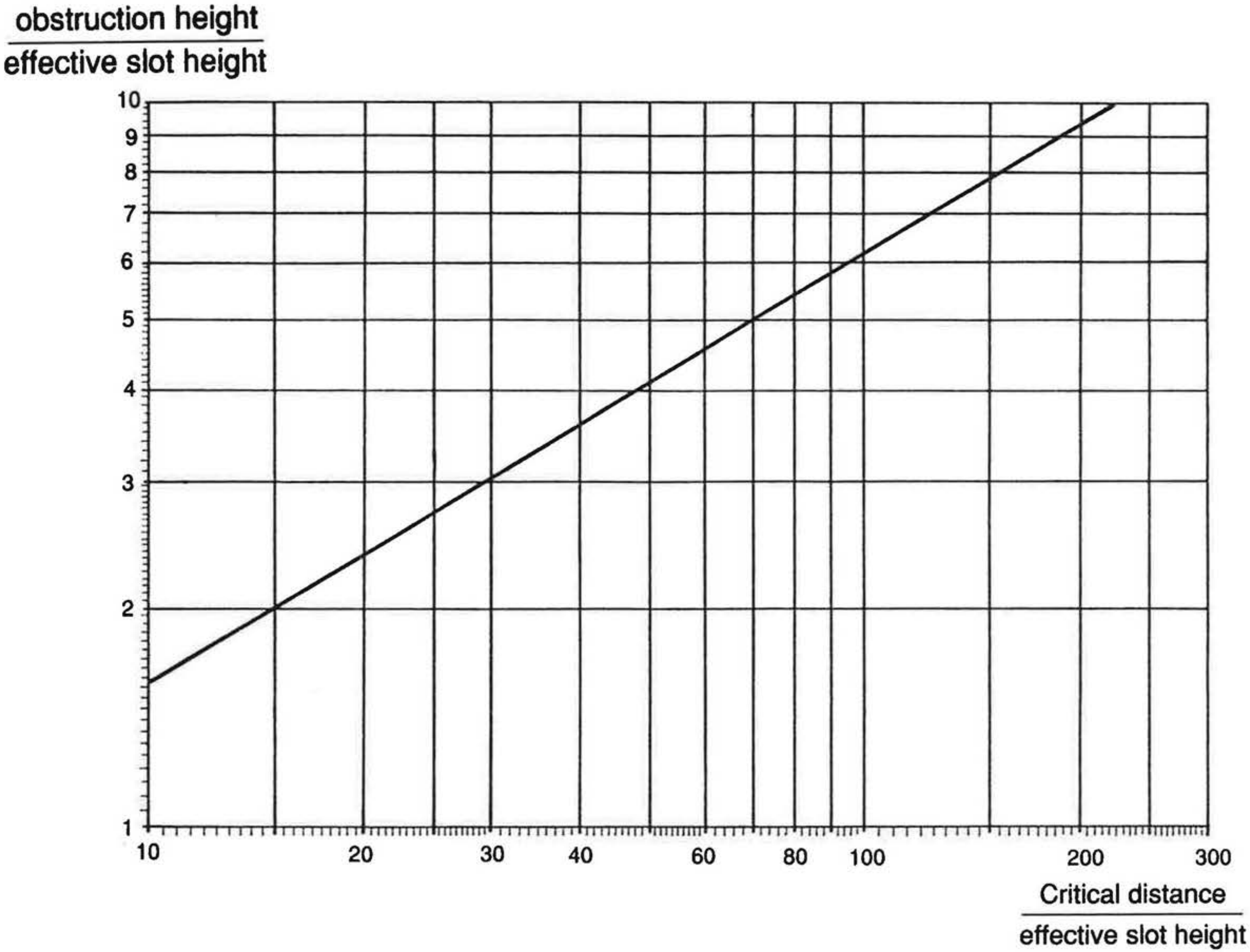
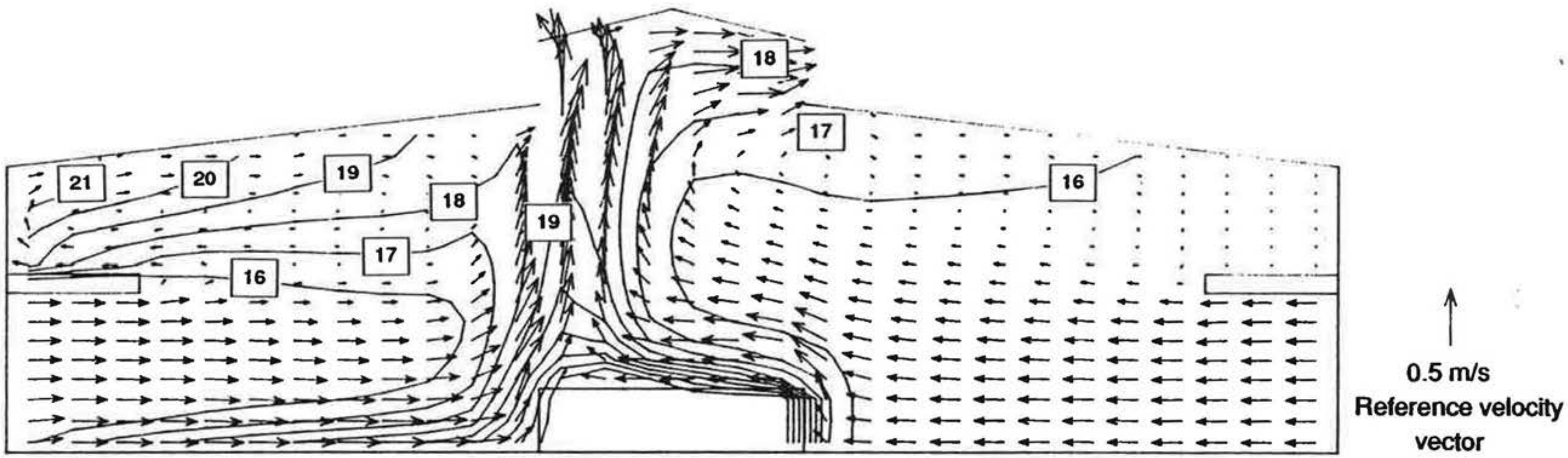


Figure 15 Graph for determining critical distance below which jet deflection will be caused by obstruction



Note: Figures represent temperature in °C

Figure 16. Example of numerical modelling output

APPENDIX A

A.1 Archimedes number

Archimedes number is the ratio of buoyancy to inertia forces and in relation to a supply air jet is defined as:-

$$Ar = \frac{g \Delta t d}{TV^2}$$

where Ar = Archimedes number (non-dimensional)

g = gravitational acceleration (m/s^2)

d = characteristic dimension of supply opening (m)

Δt = temperature difference between supply air temperature and the surrounding (room) air (K)

T = air temperature (K)

V = supply air velocity (m/s)

A.2 Jet trajectory - horizontal projection

The trajectory of a horizontally projected heated or cooled jet has been found by experiment [A1] to be related to the Archimedes number. The relationship is:-

$$\frac{1}{Ar} \cdot \frac{y}{d} = 0.052 \left[\frac{x}{d}\right]^3$$

where y = vertical deflection (m) of jet at distance x (m) from the supply opening, and where Ar is as defined above and d is taken to be the initial diameter of the supply air jet.

Rearranging this equation gives a relationship to enable the deflection, y, to be determined for various distances, x, from knowledge of Ar and d:-

$$y = 0.052\, Ar.d \left[\frac{x}{d}\right]^3$$

Tables 1-5 of the Technical Note have been compiled using this equation, assuming normal room air temperature.

A.3 Reference

A1 Abramovich GM "The theory of turbulent jets" The MIT Press, Cambridge, Massachusetts 1963.

APPENDIX B

B.1 Jet penetration - vertical projection, counteracting forces

In downward projected warm air jets, buoyancy forces reduce the effect of inertia forces and so reduce the throw and penetration of the airstream. Here penetration is defined as the maximum vertical distance travelled from the supply opening to a terminal velocity of zero. An upwardly projected cool air jet is affected in the same way.

Koestel[B1] emperically derived (with some experimental backup by other authors) an expression to predict the penetration of circular air jets. The equation given is as follows:-

$$\left[\frac{x_o}{d} + 2.85\right]^2 = \frac{3.4}{Ar}$$

where x_o = penetration (m)

d = initial diameter of supply air jet (m)

Ar = is as defined in Appendix A

Table 6 is based on this relationship, assuming normal room temperature.

For the case of a linear (plane) jet, Danishevski deduced an equation (Baturin[B2]) which, when related to the vertical direction, reduces to:-

$$x_o = 1.28\, b\left[\frac{1}{Ar}\sqrt{\frac{T_i}{T_o}}\right]^{2/3}$$

where x_o = penetration (m)

b = initial width of supply air jet i.e. slot width (m)

T_i = surrounding air temperature (K)

T_o = supply air temperature (K)

Ar is as defined in Appendix A with b used as the characteristic dimension.

This equation has been used to derive the penetration values shown in Table 7, assuming normal room air temperature.

B.2 References

B1 Koestel A. "Computing temperatures and velocities in vertical jets of hot or cold air". ASHVE Trans Vol 60, 1954

B2 Baturin V.V. "Fundamentals of industrial ventilation". Pergamon Press, Oxford 1972.

APPENDIX C

C.1 Jet behaviour, vertical projection, combined forces

In downward projected cool air jets and in the upward discharge of warm air jets, the buoyancy forces act in the same direction as the inertia forces. These are known as buoyant jets and experimental data for such jets discharged into calm environments have been reviewed by Chen and Rodi[C1]. It has been shown that a buoyant jet initially decays in velocity until it eventually behaves as a pure plume. As a pure plume, it is driven by buoyancy forces only and moves at a constant velocity. Table C1 shows the equations derived for the centre line velocity and temperature in the non-buoyant, intermediate and buoyant plume regions for both circular and linear (plane) jets.

Based on these equations, Tables 8 and 9 show the throws for terminal velocities of 0.5 and 0.3 m/s, of chilled air jets projected vertically downwards.

C.2 Reference

C1 Chen C.V. and Rodi W. "Vertical turbulent buoyant jets - a review of experimental data". Pergammon Press, Oxford 1980.

Table C1 - Equations relating to non-isothermal jets

	Axisymmetric jet	Plane jet
Non-boyant region	$\frac{U_{\not{c}}}{U_0} = 6.2\left(\frac{\rho_0}{\rho_a}\right)^{\frac{1}{2}}\left(\frac{X}{D}\right)^{-1}$	$\frac{U_{\not{c}}}{U_0} = 2.4\left(\frac{\rho_0}{\rho_a}\right)^{\frac{1}{2}}\left(\frac{X}{D}\right)^{-\frac{1}{2}}$
	$T_c = \left[\left\{5\left(\frac{\rho_0}{\rho_a}\right)^{-\frac{1}{2}}\left(\frac{X}{D}\right)^{-1}\right\}(T_0 - T_a)\right] + T_a$	$T_{\not{c}} = \left[\left\{2\left(\frac{\rho_0}{\rho_a}\right)^{-\frac{1}{2}}\left(\frac{X}{D}\right)^{-\frac{1}{2}}\right\}(T_0 - T_a)\right] + T_a$
Intermediate region	$\frac{U_{\not{c}}}{U_0} = 7.26F^{-1/10}\left(\frac{\rho_0}{\rho_a}\right)^{9/20}\left(\frac{X}{D}\right)^{-4/5}$	$\frac{U_{\not{c}}}{U_0} = 2.85F^{-1/6}\left(\frac{\rho_0}{\rho_a}\right)^{5/12}\left(\frac{X}{D}\right)^{-\frac{1}{2}}$
	$T_{\not{c}} = \left[\left\{5.59\,F^{1/8}\left(\frac{\rho_0}{\rho_a}\right)^{-7/16}\left(\frac{X}{D}\right)^{-5/4}\right\}(T_0 - T_a)\right] + T_a$	$T_{\not{c}} = \left[\left\{1.65F^{1/6}\left(\frac{\rho_0}{\rho_a}\right)^{-5/12}\left(\frac{X}{D}\right)^{-3/4}\right\}(T_0 - T_a)\right] + T_a$
Boyant plume region	$\frac{U_{\not{c}}}{U_0} = 3.5F^{-1/3}\left(\frac{\rho_0}{\rho_a}\right)^{1/3}\left(\frac{X}{D}\right)^{-1/3}$	$\frac{U_{\not{c}}}{U_0} = 1.9F^{-1/3}\left(\frac{\rho_0}{\rho_a}\right)^{1/3}$
	$T_{\not{c}} = \left[\left\{11F^{1/3}\left(\frac{\rho_0}{\rho_a}\right)^{-1/3}\left(\frac{X}{D}\right)^{-5/3}\right\}(T_0 - T_a)\right] + T_a$	$T_{\not{c}} = \left[\left\{2.4F^{1/3}\left(\frac{\rho_0}{\rho_a}\right)^{-1/3}\left(\frac{X}{D}\right)^{-1}\right\}(T_0 - T_a)\right] + T_a$

Where

$U_{\not{c}}$=centre line velocity (ms^{-1})
U_0=velocity of discharge (ms^{-1})
ρ_0=discharge density (kgm^{-3})
ρ_a=ambient density(kgm^{-3})
D=jet width(plane jet) or diameter (circular jet) at source (m)
X=throw (m)

F=Froude number $\left(\frac{U_0^2\,\rho_0}{(\rho_a-\rho_0)\,gD}\right)$
$T_{\not{c}}$=centre line temperature (°C)
T_0=supply air temperature (°C)
T_a=ambient air temperature (°C)