Guide to Air Distribution Technology for the Internal Environment

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HEVAC

Contents

Chapter One

1 INTRODUCTION TO AIR DISTRIBUTION TECHNOLOGY

This booklet is intended as a guide to designers or the contractors who install air diffusion equipment and is complementary to BS EN12238 - Air terminal devices. Aerodynamic testing and rating for mixed flow application, BS EN12239 - Air terminal devices. Aerodynamic testing and rating for displacement flow application.

Although not specifically related to the final air diffusion within a conditioned space, the following European standards will serve as a useful reference for air distribution systems in general:

1 Air terminal device/unit performance
BS EN 16445 Aerodynamic testing and rating for mixed flow application: non isothermal procedure for cold jet
BS EN ISO 5135 Acoustics - Determination of sound power levels of noise from air terminal devices, air terminal unit, dampers and valves in a reverberant room

2 Design criteria
BS EN 13779 Ventilation for non-residential buildings. Performance requirements for ventilation and room-conditioning
BS EN ISO 7730 Ergonomics of the thermal environment

3 Room air distribution measurement
BS EN 13182 Instrumentation requirements for air velocity measurements in the in ventilated spaces
BS EN 15726 Air diffusion. Measurements in the occupied zone of air-conditioned/ventilated rooms to evaluate thermal and acoustic conditions

4 Other standards which serve as a useful reference
BS EN 1751 Air terminal devices. Aerodynamic testing of dampers and valves
BS EN 13642 Floor mounted air terminal devices. Tests for structural classification

Another useful document to consult when listing air terminal devices or equipment is HEVAC's current General Specification and Product Directory for Air Distribution Equipment Issue Level 2.

The ultimate purpose of installing the extensive equipment and controls involved in an air conditioning or ventilation system is to provide adequate, controlled air movement at the specified conditions in the treated space. If the air distribution/system is inadequate or inefficient then this reflects on the efficiency as a whole. The selection of air terminal devices is therefore of paramount importance and it is hoped that the sections throughout this guide will provide current best practice for reference purposes.

Although the sections that follow give a generic overview of the various designs that may be considered, manufacturers' data should be followed in order to obtain the best results from their particular products.

The performance data published in the manufacturers' catalogues are the results of extensive research and operational studies. Where actual performance on site falls short of the design criteria, this can generally be attributed to incorrect interpretation of the catalogue information or to faulty installations.

It should be understood that the conditions which apply in the laboratories may be impossible to achieve on site, since structural or architectural features may determine how an air terminal device is installed. However, if principles of air distribution are applied, a satisfactory solution to site problems can be reached.

The use of the following standards used early in the design stage will assist both designers and contractors in obtaining a satisfactory air distribution system.

Chapter Two

2 CONVENTIONAL AIR TERMINAL DEVICES TOGETHER WITH THEIR NORMAL APPLICATIONS

2.1 INTRODUCTION

This section will explore and list the current range of air terminal devices. It is as comprehensive as possible and lists each product under two main air distribution categories (a) mixed flow air distribution, (b) displacement flow air distribution.

In order to cover a wide range of applications, supply, return, air transfer and air terminal devices are manufactured in a number of specific categories. These main categories are as follows:-

Grilles (square, rectangular and linear)
adjustable, single or double deflection
fixed bar
eggcrate
non-vision
perforated, stamped or mesh
high security

Circular or Rectangular Diffusers (non-adjustable)
multi-cone
perforated plate
pan type

Circular or Rectangular Diffusers (adjustable)
multi-cone
perforated plate
pan type

Linear Diffusers
Slot; non-adjustable, adjustable
Multi-blade; non-adjustable, adjustable

Specialist Air Terminal Devices
nozzles/drums
disc valves
air terminal luminaires
ventilated ceilings
laminar flow panels
displacement ventilation diffusers
linear high induction diffusers

Swirl Diffusers (non-adjustable)
square type
circular type

Swirl Diffusers (adjustable)
square type
circular type
floor swirl diffusers

Details of these various types of devices together with the systems normally used within are further explained and detailed in table 2.1 pages 8 to 10.

Systems & Applications
In order to select the correct air terminal device for (a) mixed flow air distribution systems and (b) displacement/laminar flow air distribution system, the air distribution categories will further be explained. Suitable products for these systems together with the supply positions are also given in table 2.1. Unit positions have therefore been split as follows:-

(a) mixed flow - supply positions
1. ceiling
2. wall
3. floor

(b) displacement/laminar flow - supply positions
1. ceiling
2. wall
3. floor

(a) Mixed Flow Air Distribution Systems

The principle of this type of air distribution system design is to inject air from the various types of air terminal devices into the conditioned space, thus generating high induction and effective mixing with the existing room air to provide comfort conditions within the occupied zone. The effectiveness of the system will normally be judged by the occupants and their perception of the room conditions, measured generally as room air movement and temperature differentials within the occupied zone. High air change rates throughout the space make the selection and positioning of air terminal devices more demanding, with usually this selection taken from the range of high induction devices. A differential between the supply and room air temperatures is normally limited to 10^{o}C for cooling, but will depend upon the devices used and their positions. Reference to table 2.1 will indicate the current type of air terminal devices together with their application and correct positioning throughout the conditioned space. The philosophy of mixed flow systems is based upon continual dilution of stale warm air by a fresh supply. A feature of this system is that conditions within the enclosure are considered to be predominantly uniform. A simple illustration of the air patterns from such a system can be seen in figure 2.1.

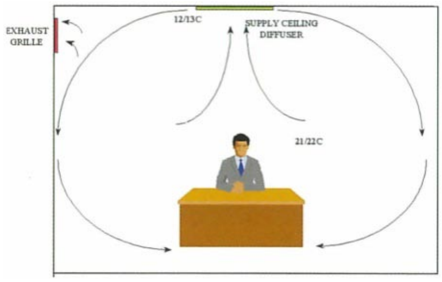


Figure 2.1 Mixed flow

(b) Displacement/laminar Flow Air Distribution Systems

With these types of air distribution systems air is introduced into a space in an even pattern in order to generate gradual displacement of existing room air. This can either be from a lower level which is normally referred to as displacement ventilation, or from a high level which is normally described as laminar flow.

Displacement Flow

In this type of flow air enters through the air terminal units and spreads across the floor forming a reservoir of fresh cool air. Any sources of heat will generate a thermal plume rising upwards and thus entraining the surrounding air as illustrated in figure 2.2. The fresh cool air at floor level will flow to replace that which is warmed and lifted into the convective plumes. By this means the fresh air continually replaces the air as it is used and contaminated air is lifted to a higher level from which it is removed. This gives rise to stratification of temperature and contaminant levels within the enclosure. Displacement ventilation works on lower temperature differentials than mixed flow systems (normally 3-6^{o}C cooling) and uses buoyancy to distribute the air throughout the space. This has the effect of potentially being more energy efficient and at the same time new fresh air is kept separate from stale air and subsequently the breathing air quality is substantially uncontaminated. Air can be introduced at this lower level through either side wall displacement units or floor grille/swirl outlets.

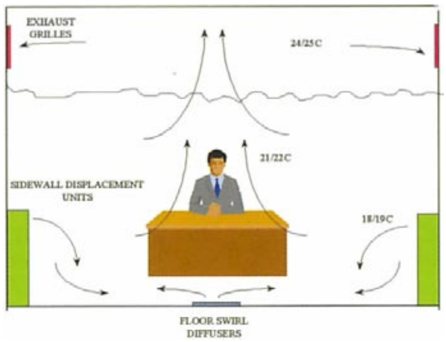


Figure 2.2 Displacement Flow

Laminar Flow

This type of system introduces fresh air at generally a ceiling or side wall position and is used to remove contaminants from working areas by cascading cooler fresh air down over the conditioned space. Contaminated air is then removed from lower areas and therefore a continual flow of air migrates over the working surface and avoids high mixing conditions. General applications for this type of system are industrial working areas, clean rooms and hospital theatre spaces. Air is normally introduced at a lower temperature than the surrounding room air and thus falls through the conditioned zone to a lower level. A typical illustration of such a system can be seen in figure 2.3. This type of system introduces air using proprietary laminar flow panels or a full ventilated ceiling or wall.

Note: Unless total wall or ceiling surfaces are used for air supply, only local laminar flow areas will be achieved.

It is important to ensure that the supply air flow rate is at least equal or greater than the estimated air flow rate in the heat source convective plumes; otherwise the system can change into mixed flow in the upper areas of the occupied zone.

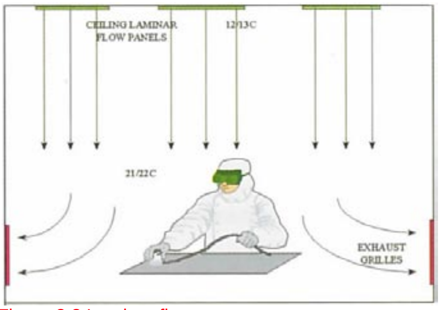
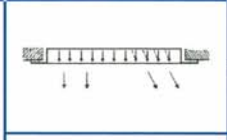
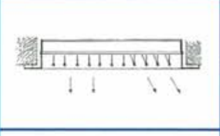
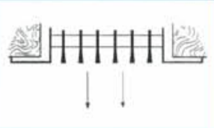
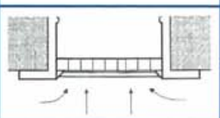
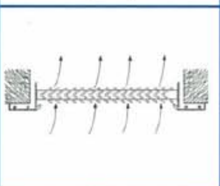
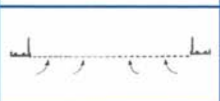
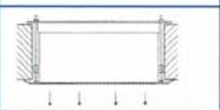
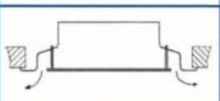
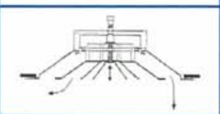
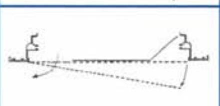
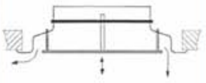
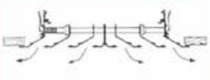
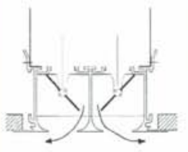
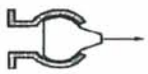
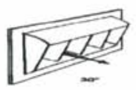
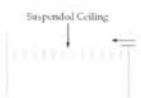


Figure 2.3 Laminar flow

Table 2.1 MAIN AIR TERMINAL DEVICES, SYSTEMS AND APPLICATIONS

MODEL	DIAGRAM	TYPE	SYSTEM M,D	APPLICATION S,E,T	POSITION C,W,F	RECOMMENDED CORE VELOCITIES*m/s		DESCRIPTION
						Quiet upto	Commercially Quiet upto	
Adjustable single or double deflection		Square/ Rectangular/ Linear	M	S, E	C, W	4	8	Frequently used grille, large free area, in supply application has directional control in one plane only by adjustment of aerofoil blades.
		Square/ Rectangular/ Linear	M	S, E	C, W	4	8	As above, but gives direction control in two planes.
Fixed bar		Square/ Rectangular/ Linear	M	S, E	C, W, F	4	8	General robust grille with free area dependent upon application. Some directional control can be achieved by use of profiled blades or by using additional rear adjustable blades which are perpendicular
Eggcrate		Square/ Rectangular/ Linear	M	E	C, W	4	8	Generally the largest free area grille available.
Non-Vision		Square/ Rectangular	M,D	E,T	W or Door	3	6	Low free area designed to provide no through vision. Grilles which permit the predetermined passage of air from one treated space to another. Grilles normally have one set of non vision type fixed angle or chevron shaped blades which obstruct direct line of sight through the grille core.
Perforated, stamped or mesh		Square/ Rectangular/ Linear	M,D	S, E,T	C, W, F	3	6	Simple form of grille relatively small free area.
High Security		Square/ Rectangular/	M	S, E,T	C, W	3	6	Heavy and robust grille with generally low free area. Vandal proof grids or plates as part of the construction with additional balancing blades if required.
Multi-cone		circular or rectangular diffusers (non-adjustable)	M	S, E	C	4	8	Radial discharge diffuser offering good air entrainment and allowing large quantities of air to be diffused into a room. In square or rectangular form can provide 1, 2, 3 or 4 way discharge.
Perforated face		circular or rectangular diffusers (non-adjustable)	M	S, E	C	3	6	Radial discharge diffuser using deflectrol vanes or baffles behind perforated face for directional control. 1, 2, 3 or 4 way discharge.
Pan		circular or rectangular diffusers (non-adjustable)	M	S, E	C	4	8	Central baffle plate design, fixed in position to generate horizontal discharge of supply air.
Multi-cone		circular or rectangular diffusers (adjustable)	M	S, E	C	4	6	As multi-cone fixed geometry units, but offers facility for horizontal or vertical air discharge.
Perforated face		circular or rectangular diffusers (adjustable)	M	S, E	C	3	6	As perforated face fixed geometry units with additional adjustable control for horizontal or vertical air discharge.

SYMBOLS		
SYSTEM	APPLICATION	POSITION
M = Mixed Flow Air Distribution Systems D = Displacement Air Distribution Systems	S = Used to supply air to conditioned space E = Used to extract air from conditioned space T = Transfer of air from one conditioned space to another *based on any local control damper being fully open*	C = Ceiling W = Wall F = Floor

MODEL	DIAGRAM	TYPE	SYSTEM M,D	APPLICATION S,E,T	POSITION C,W,F	RECOMMENDED CORE VELOCITIES*m/s		DESCRIPTION
						Quiet	Commercially Quiet	
						upto	upto	
Pan		circular or rectangular diffusers (adjustable)	M	S, E	C	4	8	As pan fixed geometry units but offering vertical positioning of central baffle plate for horizontal or vertical air discharge control.
Linear		circular or rectangular diffusers (adjustable)	M	S, E	C,W	4	8	Fixed blade design offering either horizontal or vertical discharge, single or multiple slot design.
Linear Slot Diffusers		linear diffusers (adjustable)	M	S, E	C,W	4	8	Offers vertical or horizontal discharge single or multiple slots available. Care must be taken with associated plenum box design. Components normally used as ceiling diffusers having a continuous slot appearance and generally having an aspect ratio greater than 10:1. Adjustable single or multi slot diffusers shall incorporate flow control blades to set the air discharge pattern for one of the following combinations. Discharge characteristics can be set as follows:- a) horizontal one way b) horizontal two way c) horizontal alternating d) alternating between horizontal and vertical per slot e) vertical
Nozzles/ Drums A B C	  	Specialist air terminal devices	M	S	C,W Free Space	6	10	Nozzles/drums are air terminal devices designed to give low energy loss and thus to produce a maximum throw with minimum air entrainment. These are generally divided into three separate groups related to function, (a) small nozzles for personal ventilation/spot cooling, (b) small nozzles in arrays for mixed flow application, (c) large nozzles for projecting air over long distances. Nozzles shall either be circular or rectangular, using a single unit or multiple array and shall have the following facilities as specified. Fixed or adjustable air pattern, fixed or adjustable air discharge direction, and air flow rate control.
Disc valves		Specialist air terminal devices	M	S, E	C, W	6	10	Disc valves have an adjustable circular core complete with mounting sub-frame. The core position is adjustable for flow rate control purposes and suitable normally for exhaust air. Special designs are necessary for supply air conditions.
Air handling luminaires		Specialist air terminal devices	M	S	C	4	8	Air handling luminaires are complete with adjustable or fixed linear slot diffuser within integral plenum box, incorporating flow control blades to set the pattern for horizontal one way or vertical air diffusion. Fitted to either one or more sides of a luminaire.
Ventilated ceiling		Specialist air terminal devices	M	S, E	C	3	6	The void above the ceiling is pressurised and air is introduced at low velocity through many single holes or porous panels forming the ceiling. Entrainment of room air is restricted and natural currents in the room can seriously affect room air distribution.

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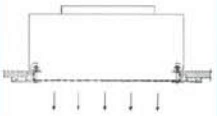
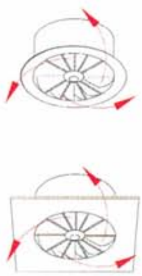
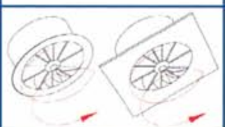
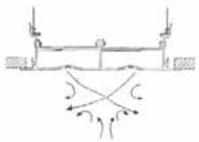
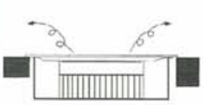
MODEL	DIAGRAM	TYPE	SYSTEM M,D	APPLICATION S,E,T	POSITION C,W,F	RECOMMENDED CORE VELOCITIES*m/s		DESCRIPTION
						Quiet	Commercially Quiet	
						upto	upto	
Laminar flow panels		Specialist air terminal devices	M,D	S, E	C, W	3	6	Laminar flow panels have a large perforated face plate complete with rear plenum box and spigot entry. Generally incorporating a damper for flow control purposes and all designed to provide uniform air discharge at 90° to the panel face. Normally used for supply applications but can be matched for appearance purposes for extract applications.
Displacement ventilation diffusers		Specialist air terminal devices	D	S	W,F	3	6	Displacement ventilation panels are constructed with a large face area normally using a perforated material. The remaining construction normally incorporates a rear plenum box with spigot entry in which an air inlet damper can be fitted for flow rate control if required. Units are designed to provide uniform air discharge at low velocities over the face area, and to supply air directly into the occupied zone of a conditioned space. Note: Can be supplied in addition to flat diffusers in 90°, 180° and 360° configurations.
Square And circular		Swirl diffusers (non-adjustable)	M	S	C	4	8	The construction of this type of diffuser causes a swirling or highly turbulent discharge which accelerates the mixing of the supply air into the surrounding space. This is generally greater than the mixing effect from conventional square or circular diffusers. The swirling motion of the discharge air is imparted by either a device behind the diffuser face and/or the configuration of the face itself. The air direction can be fixed generally for a horizontal discharge, but may be vertical in special cases. Designs of this type of device vary from open face vanes as illustrated to multiple slot arrays or perforated face units.
Square And Circular		Swirl diffusers (adjustable)	M	S	C	4	8	As square or circular fixed geometry swirl diffusers but offering the additional control of air discharge from horizontal to vertical by manual or motorised means of control.
Linear		High induction design (fixed or adjustable)	M	S	C,W	4	8	These units offer a similar high turbulent discharge to the square or circular swirl diffusers which accelerates the mixing of the supply and room air over and above that offered by conventional linear slot diffusers. Units can be fixed or adjustable if required.
Square/ Circular		Floor Swirl diffusers (fixed or adjustable)	M,D	S, E	F	4	8	These are specifically designed swirl diffusers for floor discharge application where the unit shall impart a swirl motion of the discharge air either by a device by the diffuser face and/or by the configuration of the face itself. The discharge shall be vertical, horizontal or adjustable between the two. Units should be of a robust design to take required floor loading requirements.

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Chapter Three

3 SELECTION OF AIR TERMINAL DEVICES

3.1 INTRODUCTION

The function of an Air Terminal Device (ATD) is to direct the incoming or exhaust air, in such a way that comfortable and clean conditions are maintained in the occupied zone of the treated space. Failure to choose a suitable ATD, especially one used for supply purpose, may well nullify all other efforts to achieve such conditions in the treated space. At the same time, the ATD invariably needs to suit aesthetic requirements.

Acceptable conditions in any occupied zone will depend upon the use of that zone, but since it is the purpose of the supplied air (possibly together with other means) to maintain these conditions, this will lead to definite requirements for supply and exhaust air flow rates, noise level, and air movement in the occupied zone.

From these parameters, it is possible to put forward several solutions which, whilst economical in themselves, might not lead to a satisfactory overall scheme. Therefore, it is necessary at the selection stage, to choose the position (wall, ceiling, floor, or sill) of the ATD's together with the number, form and type.

Before full technical proposals can be made, consideration must be given to the occupancy and function of the space, and to internal features, such as irregularities of surface, position of furniture, and any sources of heat loss or gain.

It is important to note that duct entry conditions to the ATD's must be correctly and carefully considered to ensure ATD will work as designed.

3.2 CHARACTERISTICS OF AIR JETS

3.2.1 Mixed flow applications

Many air terminal devices are designed to supply air in a predominantly unidirectional sense and with significant throw; hence it is important to understand the characteristics of air jets.

The flow profile of an air jet depends upon the temperature difference between supply and room air as well as the proximity of adjacent surfaces, however, the discharge characteristics of an isothermal air jet, as shown in figure 3.1, can generally be described as follows:

Zone 1

A short zone extending to a length of approximately four outlet diameters from the discharge point where the centre line airstream velocity remains practically constant.

Zone 2

In most cases a transitional zone where the centre line velocity decays inversely as the square root of the distance from the outlet. However, for high aspect ratio terminals this section can extend up to a distance of approximately the width of the terminal multiplied by four times the aspect ratio.

Zone 3

This is a long zone where fully established turbulent flow exists and usually extends from 25 to 100 outlet diameters from the outlet. This zone is of great importance and where most ATD selection takes place. The centre line velocity decays inversely with increasing distance from the outlet.

Zone 4

The final zone where the centre line velocity decays rapidly to a value below 0.25 m/s.

Symbols

A = Discharge Area
X = Distance from Discharge Point
Vx = Centre Line Air Velocity
Vo = Air Velocity at Discharge Point

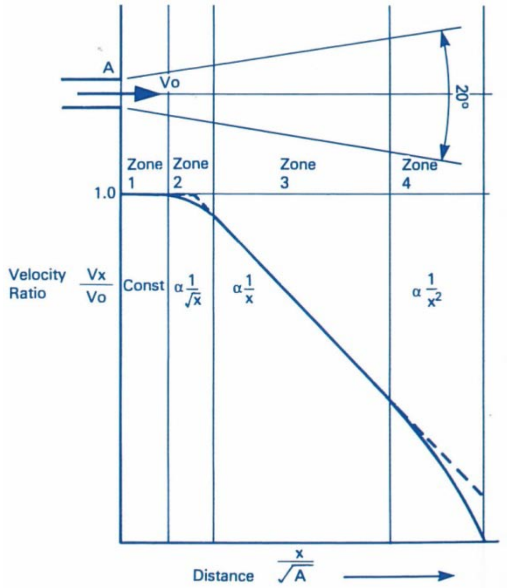


Figure 3.1 Profile of a free air jet

3.2.1.1 Throw

The distance measured from the face of the ATD to the point at which the maximum axial velocity reduces to a specific value, normally 0.25 to 0.5m/s, is known as the throw. Values of throw for a range of ATD sizes and appropriate flow rates are given in manufacturers' catalogue data, see figure 3.2 for a typical characteristic.

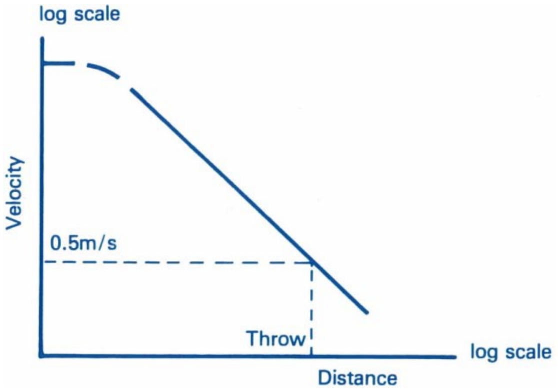


Figure 3.2 Typical Characteristic for an ATD

3.2.1.2 Effect of temperature difference

Vertical jets

Incoming air with a higher or lower temperature than room air will have a lower or higher density respectively. This will have the effect of increasing the throw of cool downward or warm upward projected air. Conversely, the throw of a warm downward or a cool upward projected air stream will be reduced (See Figure 3.3).

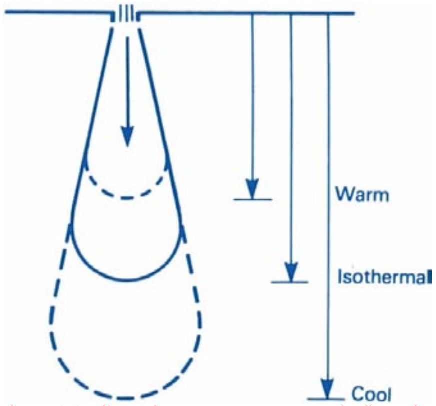


Figure 3.3 Effect of temperature on a vertically projected air stream

Horizontal jets

The trajectory of a horizontally projected air stream will curve upwards if the incoming air is warmer than room air and downwards if cooler. The vertical deflection from the horizontal plane is defined as the 'drop' or 'rise' as shown in figure 3.4.

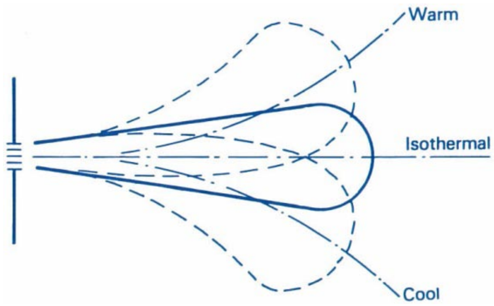


Figure 3.4 Effect of temperature on a horizontally projected air stream

Note: The magnitude of these effects for vertical and horizontal jets is a function of the temperature differential and the initial air stream velocity. Many manufacturers include information on ATD performance at specific temperature differentials, which vary depending on type and function of the ATD.

3.2.1.3 Spread

The spread is defined as the divergence (or 'widening') of the air stream in a horizontal and/or vertical plane after it leaves the supply air terminal.

3.2.1.4 Effect of adjacent surfaces

When an ATD is located in or close to a surface so that the discharged air stream passes along and adjacent to that surface the characteristics of the air stream will be affected. The entrainment in the region close to the surface will be inhibited and the reduction in velocity of the air jet will be less rapid.

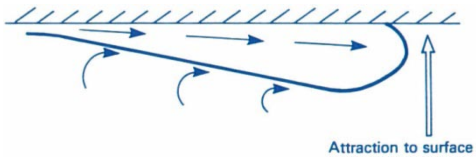


Figure 3.5 Attraction to an adjacent surface

There will also be a tendency for the air stream to 'cling' to the surface and this reduces the influence of temperature differences which normally cause the trajectory of a horizontally projected air stream to curve downwards (in the cooling mode).

This effect, often called 'ceiling effect' or 'Coanda effect', is more pronounced in radial air streams and in those produced from linear devices because of the large perimeter area in proximity to the surface. Thus by mounting circular and linear diffusers in the ceiling and setting them to discharge horizontally, maximum advantage of this effect can be gained, allowing the throw of the air stream to be somewhat larger than an equivalent unbounded air stream. Similar effects will be evident with wall-mounted linear grilles located just below the ceiling level. To this end, wall-jets are the normal means of air distribution in rooms with mixing flow systems.

3.2.2 Displacement flow applications

In displacement flow applications, the air terminals are located at low level in the occupied zone. In order to avoid local discomfort due to draught, the supply air temperature should only be slightly lower than the design temperature and with significantly lower momentum compared to mixed air flow systems.

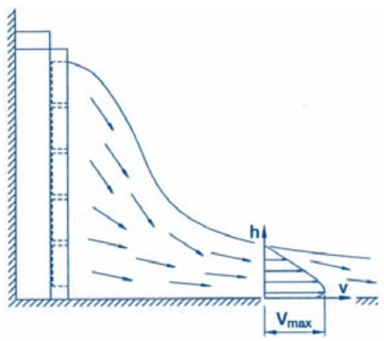


Figure 3.6 Schematic showing typical profile from a displacement air terminal

As the air leaves the diffuser, it falls to the floor under the action of gravitational forces and because the cooler supply air is denser than surrounding air, see figure 3.6. The air spreads across the floor forming a reservoir of fresh cool air. Any sources of heat in the occupied zone generate thermal plumes rising upwards entraining the surrounding air. The cool air at floor level flows to replace that which was warmed and lifted into the convective plumes. This process is continuous and gives rise to vertical temperature and contaminant gradients, also known as 'stratification'.

3.2.2.1 Near Zone

Since the air is supplied directly into the occupied zone there will always be a 'zone of discomfort' in the immediate proximity of any supply diffuser. This region is known as the "near zone" and is defined as the area around the air terminal unit in which the velocity exceeds a given value that will cause discomfort to occupants (0.15m/s in winter and 0.25m/s in summer, with reference to BS EN ISO 7730).

The extent of the near zone is a function of air flow rate, the temperature of the supply air relative to the room temperature (known as the "under-temperature"), the air terminal type and the location/proximity of the unit. It should also be noted that some units have adjustable internal elements specifically designed to alter the size and shape of the near zone. Generally, air terminals will differ from manufacturer to manufacturer, hence it is important to contact them for up-to-date information. The extent of the near zone will also be increased if units are placed too close together. To keep the near zone small it is better to use more air terminals with low flow rates rather than have fewer terminals with higher air flow rates.

3.3 TYPES OF ATD

3.3.1 Mixed flow applications

Supply ATD's fall within four main categories each with their own particular characteristics which may be used to advantage in specific applications. These categories are:

1. Sidewall mounted square and rectangular grilles and nozzles
2. Sidewall mounted linear grilles
3. Ceiling or sidewall mounted slot and linear diffusers
4. Ceiling mounted circular, rectangular and swirl diffusers

The characteristics of the air streams discharged from a typical grille are outlined below:

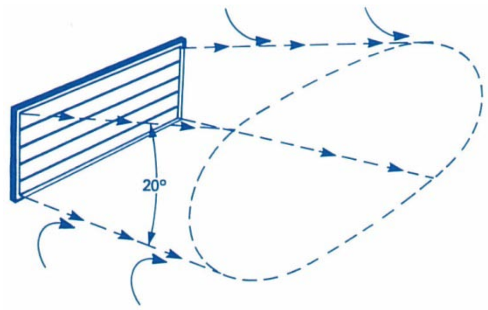


Figure 3.7 Air stream from a grille

3.3.1.1 Side wall mounted square and rectangular grilles and nozzles

From these ATD's the air discharges in a three-dimensional stream which normally flows in a direction perpendicular or near perpendicular to the face of the device. After a short distance from the ATD the velocity at the axis of the air stream starts reducing as the surrounding air is entrained into and mixes with it. This velocity reduces in direct proportion to the distance from the ATD which corresponds to zone 3 in figure 3.1.

The cross-sectional area of the air stream increases so that its boundary diverges at an included angle of about 20˚ as shown in figure 3.7.

Deflection characteristics of grilles

The most effective use of grilles incorporating air stream deflection using adjustable blades is through good spacing design. The spacing between adjacent terminals depends upon the throw and spread characteristics of these ATD's.

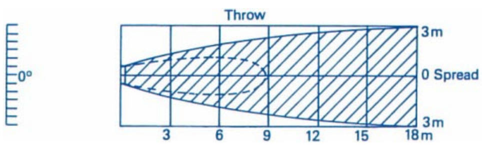


Figure 3.8 With 0° deflection, the terminal spacing interval is equal to 1/3rd of the throw (based on 0.5m/s envelope)

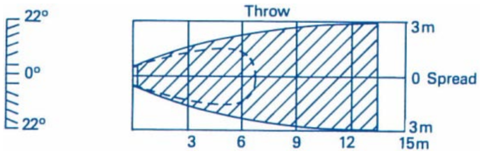


Figure 3.9 With 22° deflection, the terminal spacing interval is equal to 1/2 of the throw (based on 0.5m/s envelope)

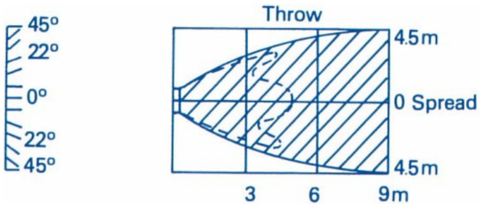


Figure 3.10 With 45° deflection, the terminal spacing interval is equal to the throw (based on 0.5m/s envelope)

3.3.1.2 Sidewall mounted linear grilles

These ATD's have a definition of length to width aspect ratio of 10:1 or greater, but can have varied geometry. The air discharge from these terminals is similar to grilles, however it forms a wider stream which can be taken as a two-dimensional jet. The primary velocity reduction is in proportion to the square root of the distance from the ATD. Expansion of the air stream takes place mainly across the thinner section, again at an included angle of about 20°. There is little significant increase in the width of the air stream (see figure 3.11).

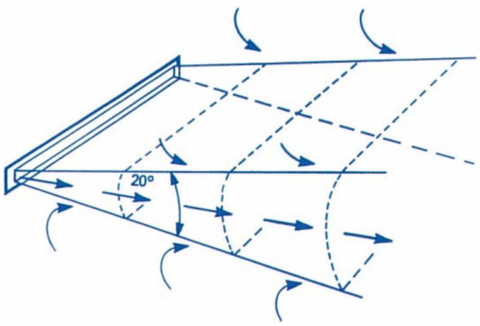


Figure 3.11 Typical linear grille

3.3.1.3 Ceiling or sidewall mounted slot and linear diffusers

These air terminals are mainly used in conjunction with attaining the ceiling effect. Air from these ATD's discharge in a wide air stream which can be regarded as two dimensional. The profile that is emitted from these diffusers are known as plane wall-jets. (See figure 3.12).

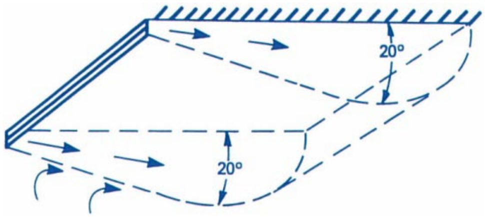


Figure 3.12 Profile of a wall jet

3.3.1.4 Ceiling mounted circular, rectangular and swirl diffusers

Air is discharged from these ATD's in a thin stream. Entrainment and expansion takes place rapidly, and as the stream progresses outwards from the diffuser its velocity reduces in direct proportion to the distance from the diffuser. These diffusers are mainly used in conjunction with attaining the ceiling coanda effect (see figure 3.13).

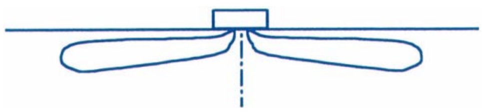


Figure 3.13 Section through a circular or rectangular ceiling mounted diffuser

Swirl diffuser (figure 3.14) units usually have a circular pattern of radial vanes which generate a swirling air motion when used in supply mode. This highly turbulent swirl effect allows the unit to introduce high volumes of air into the conditioned space, taking advantage of the rapid entrainment and intermixing characteristics. As a result the unit can deliver high room air change rates as compared to conventional diffusers.

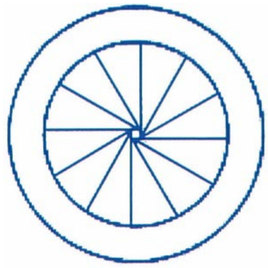


Figure 3.14 Schematic of a typical swirl

3.3.2 Displacement ventilation systems

Displacement air terminals, which are located in the occupied zone, should provide air at low velocities and at a temperature only slightly lower than room temperature so as to avoid local discomfort. In practice, there will always be a ‘zone of discomfort’ in the immediate proximity of any supply diffuser placed in the occupied zone. This zone is kept to a minimum through appropriate terminal selection, in other words one having large facial area with uniform velocity profile, and by dividing the total air flow demand over several well placed terminals. The total air flow delivered to the space need not be more than 6 air changes per hour since conditions above the occupied zone are not critical. The essence of displacement ventilation is to provide acceptable conditions in the occupied zone with considerably lower energy consumption due to lower flow rates and higher supply air temperatures. The terminal types used for displacement ventilation fall into the following categories:

3.3.2.1 Wall mounted

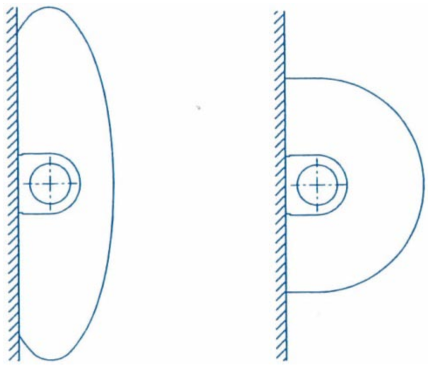


Figure 3.15 Typical profiles from wall mounted air terminal

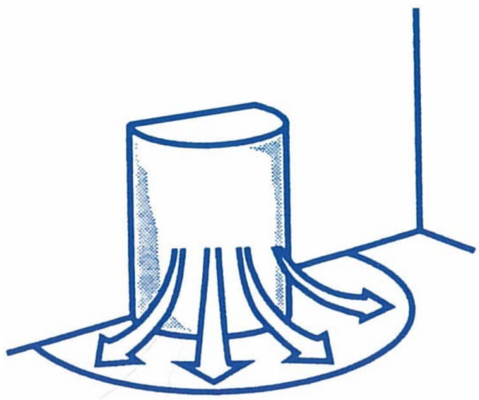


Figure 3.16 Wall mounted displacement air terminal

Figure 3.16 shows a view from a wall mounted displacement air terminal, whereas Figure 3.15 highlights the possibilities in varying the emitted airflow profile (check with the manufacturer for specific details).

3.3.2.2 Corner mounted

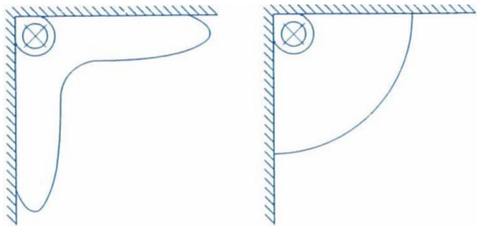


Figure 3.17 Typical profiles from corner air terminals having adjustable internal elements

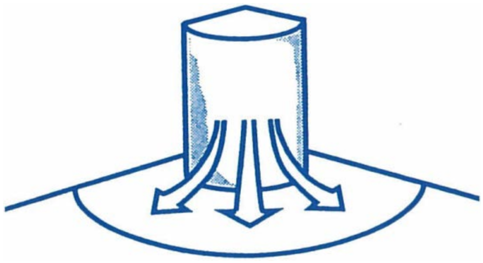


Figure 3.18 Corner mounted displacement air terminal

Figures 3.18 and 3.17 show schematic and typical profiles for corner mounted units.

3.3.2.3 Free standing
Figures 3.20 and 3.19 show the typical profiles for free standing units.

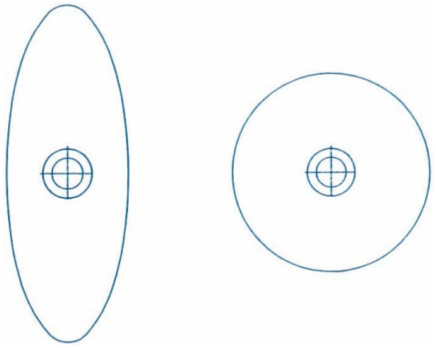


Figure 3.19 Typical profile for a free standing air terminal

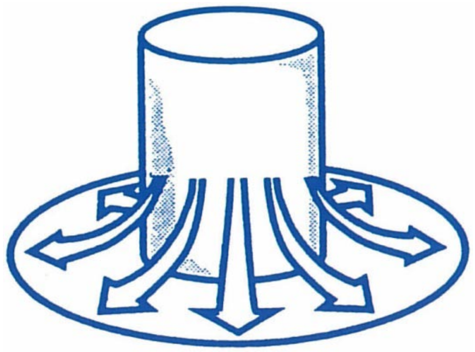


Figure 3.20 Free standing displacement air terminal

3.3.2.4 Raised access floor mounted swirl diffusers

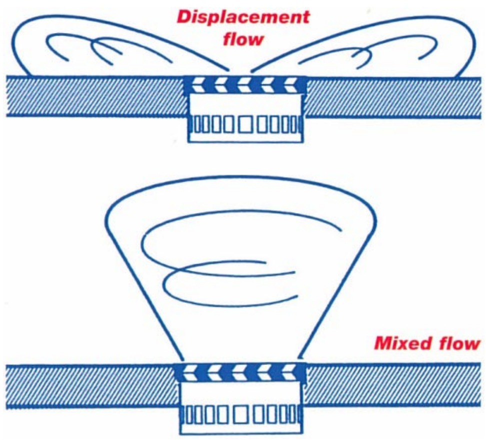


Figure 3.21 Floor mounted swirl diffuser

This type of air terminal is suitable for areas of high internal heat gain and is often used in large open plan offices (typically as a supplement to perimeter units). As can be seen from figure 3.21 these units can usually be modified to offer alternative air profiles and introduce air with a high degree of swirl which generates rapid mixing with room air so as to reduce the extent of the near zone.

3.4 COMFORT CRITERIA

The main objective for room air distribution design is to provide a suitable thermal environment in terms of comfort and ventilation effectiveness for the occupants and processes to be undertaken inside. Acceptance of thermal environment and the perception of comfort are related to metabolic heat production, its transfer to the environment, and the resulting physiological adjustments and body temperatures. The heat transfer is influenced by physical activity and clothing, as well as the environmental parameters; air speed, air temperature, mean radiant temperature and humidity. Of these, air speed and air temperature are significantly dependent on the air distribution system.

Many attempts have been made to devise indices for the specification and determination of the conditions for thermal comfort. Laboratory research has led to the production of the predicted mean vote (PMV) thermal comfort index. The PMV has been incorporated into international standard BS EN ISO 7730.

3.4.1 Air temperature

Air temperature in a room generally increases from floor to ceiling. If this vertical temperature gradient is large, local warm discomfort can occur at the head and / or cold discomfort at ankle height. Research has shown that the temperature difference between head and ankle height should be less than 3K. Thus the limiting temperature gradient depends on whether the occupants are considered to be standing or seated and gradients of 3K/m or 1.8K/m may be used respectively. It is also recommended that the limits of variation of temperature across the occupied zone of an enclosure should be within ±1.5K about the mean room air temperature.

3.4.2 Air Speed

Air speed may cause unwanted local cooling of the body, known as draught. The risk of draught depends on the local air speed, the turbulence intensity, and the air temperature. The draught risk may be expressed as the percentage of people predicted to be bothered by draughts. For a temperate climate it is recommended that the air speed be kept within the range of 0.13 to 0.18m/s.

3.5 GENERAL GUIDE ON PRELIMINARY SELECTION OF CLASS OF AIR TERMINAL DEVICE.

The following sections indicate guidelines to good practice.

3.5.1 Mixed flow applications

For a cooling temperature differential within the range of 10K (10^{o}C), Table 3.1 gives typical air change rates, and air flow rates per square metre of floor area. These should not be taken as Max. or Min. but will generally result in average velocities within the above range. The figures in Table 3.1 are for use as a general guide and have been assessed from a number of published manufacturers' data. They have been based on two room sizes of 3.6 x 3.6 x 2.5m high and 7.2 x 3.6 x 3.5m high, together with the mounting positions shown in Table 3.2. Consideration must be given to the following which may result in deviations from the values given in Table 3.1.

(a) Manufacturer.

(b) Mounting position
(c) Type of ATD.
(d) Blade divergence.
(e) Acceptable local velocities within the room.

Class of ATD	Air change rate per hour	Air flow rate (litres/sec m² floor area)	
		3.6 x 3.6 x 2.5m high	7.2 x 3.6 x 3.5m high
Grilles	8	5.6	8
Linear Grilles	10	7	10
Slot and Linear Diffusers	15	10.5	15
Rectangular Diffusers	15	10.5	15
Perforated Diffusers	15	10.5	15
Circular Diffusers	20	14	20
Swirl Diffusers	20-30	14-21	20-30

Table 3.1: General guide for mixed flow - See section 3.5.1

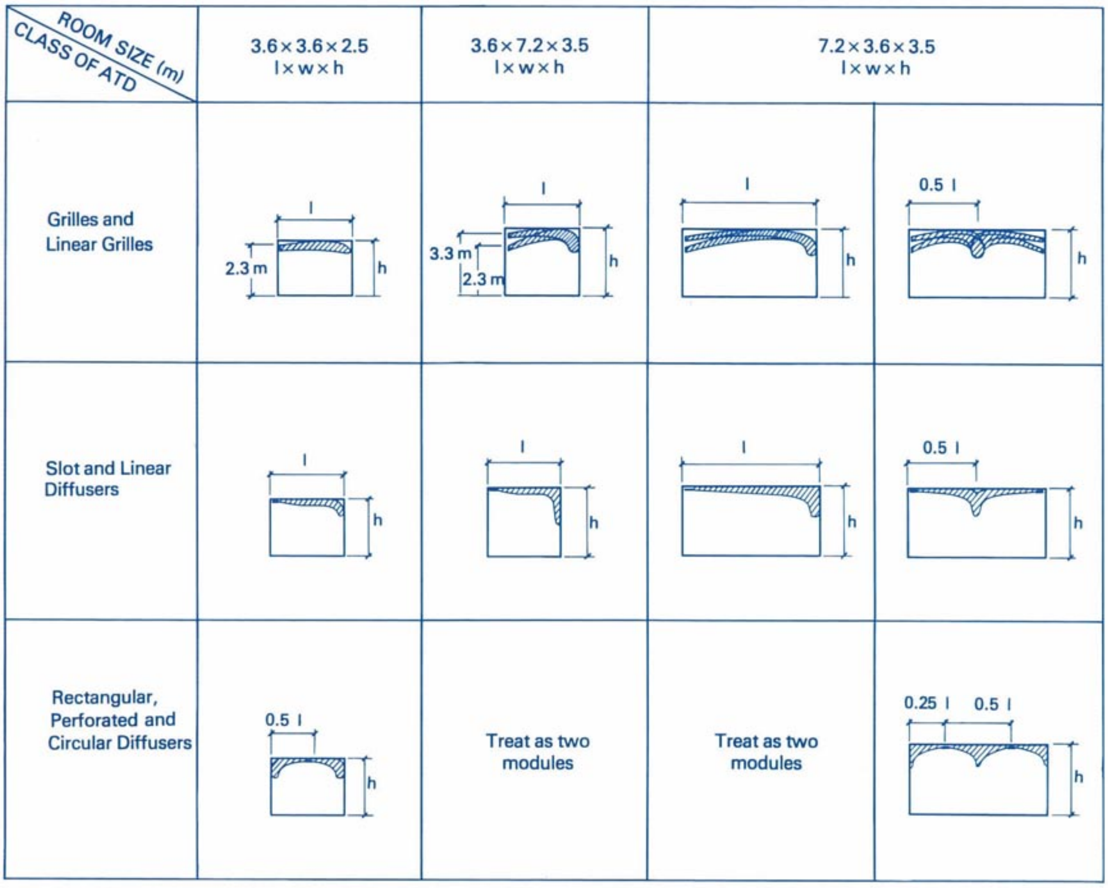


Table 3.2 Typical applications

3.5.1.1 Step by step procedures for selection and positioning of ATD's.

3.5.1.1.1 Sidewall grilles (method I).

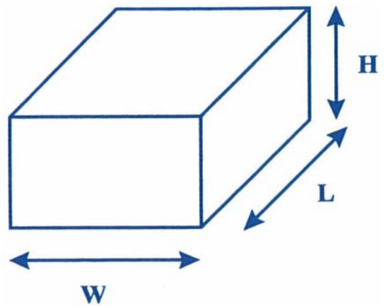


Figure 3.22 Room dimensions

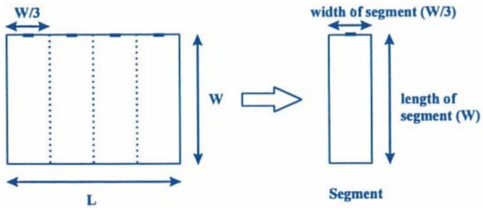


Figure 3.23 Plan view and segment size

Given the total air flow rate, q_t and room dimensions (figure 3.22) determine the number of supply ATD's (with blades set at 0^o deflection) that are needed.

Solution:

1) It was stated in section 3.3.1.1, figure 3.8, that the spacing between two adjacent air terminals should be 1/3rd of the throw of the air terminal when the grille blades are set at 0^o deflection.

2) The room is then divided along length L into equal sections such that the width/length of each segment has an aspect ratio of not less than 1:3, see figure 3.23.

3) In this example, the number of ATD's is 4.

4) Then the air flow rate through each ATD can be evaluated by the following expression:

Air flow rate per ATD = $q_t/4$

5) Once the flowrate for each ATD is known, the exact throw can be determined relative to the segment length (W) from the manufacturers catalogue for selected ATD's. This relationship is specified differently by individual manufacturers. Reference should therefore be made to the data published by the particular manufacturer under consideration.

6) Next, determine the allowable drop such that the air flow envelope at a specified velocity does not enter the occupied zone (normally up to 1.8m) causing draught, see figure 3.24, section 3.4 and table 3.2.

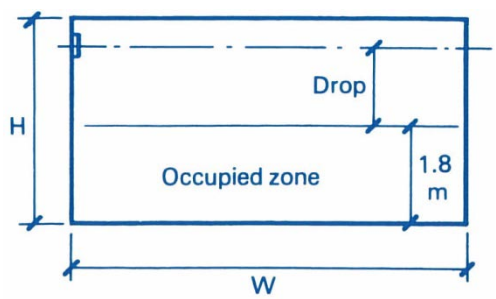


Figure 3.24 Allowable drop

7) Referring to the manufacturers' data select a suitable ATD size taking into consideration air flow, throw, temperature differential and position of the ATD in relation to the ceiling. It should be noted that drop is made up of two components

a) drop due to temperature differential

b) drop due to vertical spread.

8) Note should also be taken of the resultant pressure and noise characteristics to check that they meet the environment being served. If the above selection does not meet either the noise or pressure requirement the following alternative procedures can be adopted (Methods II and III). Both of these methods alleviate this problem

by dividing the room space into more segments, increasing the number of ATD's such that each ATD has a smaller duty.

3.5.1.1.2 Sidewall grilles (Method II)

Divide the room area in two about the longitudinal centre line in order to increase the number of ATD's and then proceed as in method I (section 3.5.1.1.1).

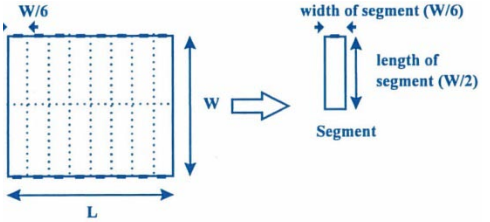


Figure 3.25 Same example, as method I, but with 16 ATD'S

3.5.1.1.3 Sidewall grilles (Method III)

1. Divide the room area into two halves about the longitudinal centreline maintaining the same segment length (W/2).

2. Set the ATD's blades at 22^o and then proceed as before.

3. Note that the segment width to segment length ratio must not be less than 1:2 according to the deflection characteristics outlined in section 3.3.1.1 which would correspond to a segment width of no less than W/4. In this example we are maintaining a segment length of W/3 which is acceptable.

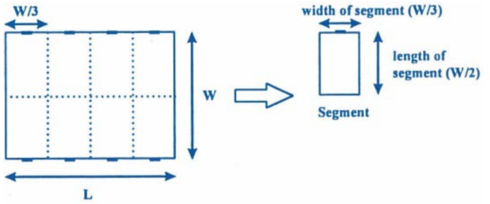


Figure 3.26 Same example as method I but with 8 ATD'S

Important note
Should the top of the core of the ATD be mounted within 200mm of the ceiling, an extension of the throw can be anticipated (as described in section 3.2.1.4) Information regarding this effect is normally given by all manufacturers of ATD's.

3.5.1.1.4 Linear grilles and registers

Given the total air flow rate (q_t) and room dimensions (as in figure 3.22) determine the number of supply air terminals and their spacing.

Solution :

1. The throw (Xm) relative to the sectional length (W) or (L) is specified differently by individual manufacturers. Reference should therefore be made to the data published by the particular manufacturer under consideration

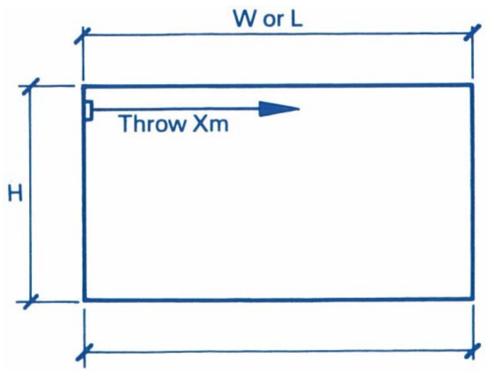


Figure 3.27 Throw

2. Determine the allowable drop such that the air jet does not enter the occupied zone. This zone is normally taken as 1.8 m from floor level. (See figure 3.24 and Table 3.2)

3. Referring to the manufacturer's data, taking into consideration throw, drop, temperature differential relative to the ceiling, establish the air flowrate per unit length q_m and the pressure and noise characteristics

4. Then determine the total active length (l) of the ATD by dividing the total air flow rate by air flow rate per unit length

Total active length (l) = q_t/q_m

5. This total active length can be divided to meet the physical constraints and/or architectural requirements (see figure 3.28). Each divided section should have a minimum aspect ratio of 1:5.

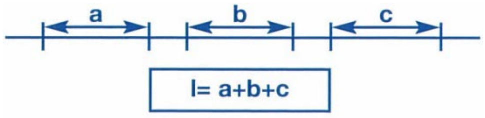


Figure 3.28 Total active length (l) divided into three sections

6) Check that the noise characteristics of the selected ATD meet the requirements of the environment being served. Note should be taken of the manufacturer's corrections for

a) Total unit lengths
b) Number of individual active outlets

Note
Should the noise characteristics exceed the environmental requirement a reduction in the velocity through the ATD may well result in premature drop of the air envelope into the occupied zone; it is therefore questionable whether the type of ATD selected is suitable for the application (See chapter 2, table 2.1).

3.5.1.1.5 Slot diffusers

From the given air flow rate, q_t and room dimensions, as in figure 3.22, determine:

1) The throw (Xm) relative to the sectional distance to the boundary of the area to be treated (W or L) see figure 3.29. This relationship is specified differently by individual manufacturers. Reference should be made to the data published by the particular manufacturer under consideration.

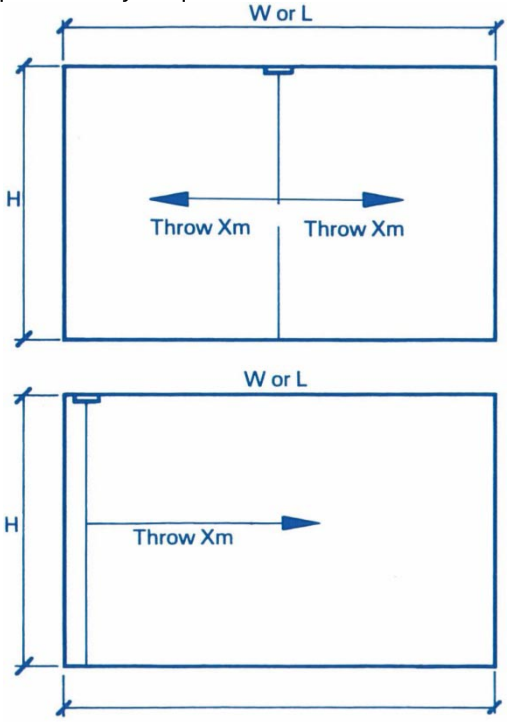


Figure 3.29 Position of slot diffuser

2) The allowable drop such that the air flow envelope at a specified velocity does not enter the occupied zone, see figure 3.24 This zone is normally taken as 1.8m from floor level. In the event of the room height being less than 2.5m reference should be made to individual manufacturers' published data.

3) Referring to the manufacturer's data and taking into account throw, drop (if applicable) and position of ATD in ceiling, establish air flow rate per unit length and width (number of slots) and slot width of the ATD to meet the throw and drop requirements. Note should also be taken of the pressure and noise characteristics.

4) The total active length of the ATD by dividing total air flow rate by air flow rate per unit length. This total length can then be divided to suit the physical limitations and architectural requirements.

5) From the manufacturer's data and the ATD resulting from the procedure in the previous paragraphs, select suitable ATD width (qualified by the number of slots) noting the noise characteristics.

6) Should architectural features indicate different design requirements select suitable width (qualified by slot widths) to meet throw requirements noting noise characteristics.

Note This type of ATD performs to the manufacturers' published data with a flat ceiling application to ensure surface attachment (see Section 3.2.1.4). With other than flat ceiling conditions it is advisable to check with the particular manufacturer of the ATD under consideration.

3.5.1.1.6 Circular, square and swirl diffusers (mounted at ceiling level)
(First selection)

Given the total air flow rate, q_t and room dimensions, as in figure 3.22, determine:

1) Number of supply ATD's by dividing the plan area into sections such that their dimensions do not exceed three times the mounting height

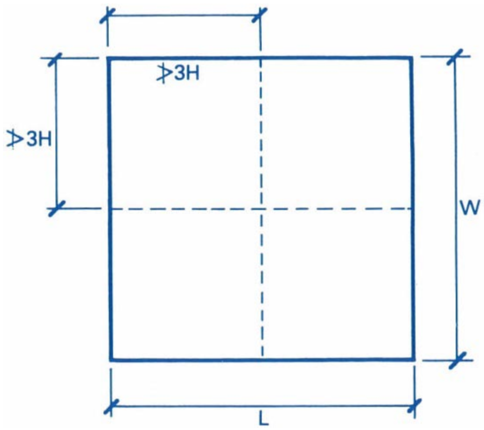


Figure 3.30 Plan room layout

2) Should rectangular sections be necessary their long dimensions should not exceed 1.5 times the short dimension, this short dimension not exceeding three times the mounting height.

3) Example: Number of ATD's = 4

4) Air flow rate for ATD by dividing the ATD's selected.

q_t = Air flow rate / number of ATD's

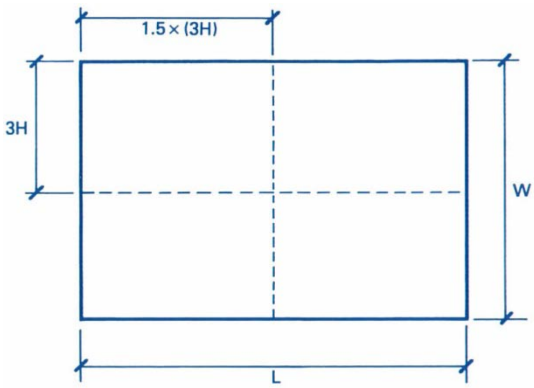


Figure 3.31 Plan room layout

5) Determine radius of diffusion (Xm) relative to the section dimensions. This relationship is specified differently by individual manufacturers. Reference should therefore be made to the data published by the particular manufacturer under consideration.

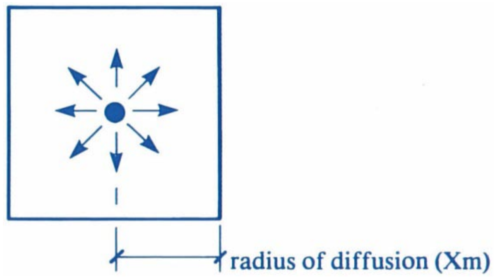


Figure 3.32 Radius of diffusion

6) Determine the allowable drop such that the air flow envelope at a specified velocity does not enter the occupied zone, see figure 3.24. This zone is normally taken as 1.8m from floor level. (See Section 3.4.2 and Table 3.2)

7) Referring to manufacturer's data select a suitable ATD from the preferred range based on air flow rate, radius of diffusion drop and temperature differential. (It should be noted that drop is made up of two components):

(a) Drop due to temperature differential
(b) Drop due to vertical spread

8) Note should also be taken of the resultant pressure and noise characteristics.

9) If the above selection does not meet either the noise or pressure requirements, review the implications in terms of using the next larger size or alternative model. Should this not meet the environment requirement, the following alternative procedures can be adopted.

Alternative selection

Divide each square or rectangular section (see section 3.5.1.1.6) into 2 or 4 smaller sections within the guide lines set out in 3.5.1.1.6. step 1 in order to increase the number of ATD's (see figure 3.33) and thus reduce the noise, throw and radius of diffusion requirements of each ATD and then proceed as in Sections 3.5.1.1.6 steps 2 to 9.

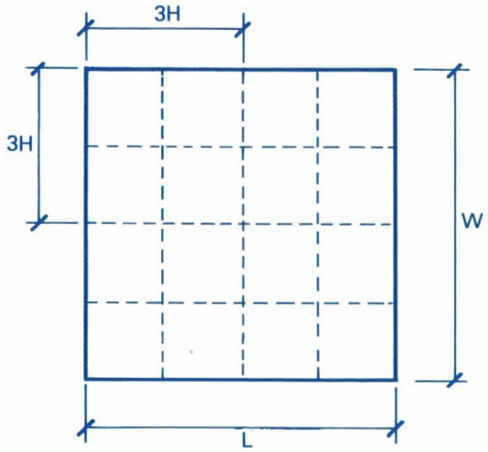


Figure 3.33 Number of ATD's= 16

Alternative Selection

For use with directional ATD's only.

1) Position one additional ATD centrally in each set of four ATD's, as shown in figure 3.34, in order to similarly reduce the radius of diffusion, the air flow rate per ATD ($q_t/5$) and the noise characteristics of each ATD.

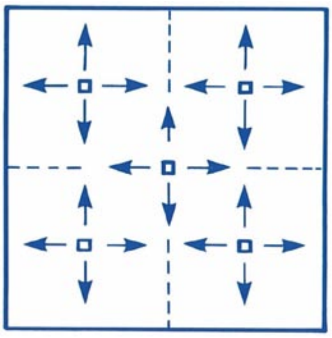


Figure 3.34 Adding one diffuser

2) Example: Number of ATD's =5

Note
Where opposing air streams meet the air will be diverted downwards and this feature should be taken into account, see figure 3.35. For further information refer to manufacturer's data

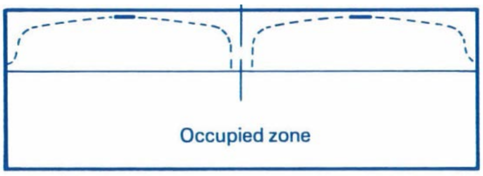


Figure 3.35 Opposing airstreams

Equalising grids and dampers where fitted in the approach duct immediately above the ceiling diffuser, can materially affect the total pressure loss and noise level. Allowances must be made for such accessories and manufacturer's data should be consulted.

3.5.2 Displacement flow applications

Air terminals should not be placed too close together since the combined effects will generate a near zone greater than the extent of a single unit near zone.

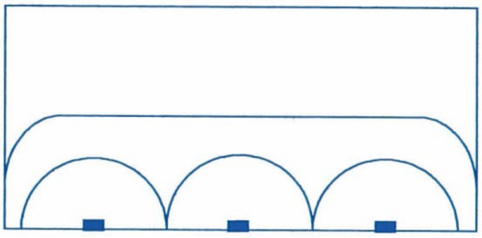


Figure 3.36 Near zone profiles

There should be a uniformly spaced layout so that the near zone profiles do not collide see figure 3.36. Reference should always be made to the manufacturer's literature. It is particularly important with displacement flow applications to locate the exhausts at high level.

3.5.3 Exhaust air terminal devices

The selection and location of exhaust ATD's is much less critical than that of supply ATD's. This is because the air flow through an exhaust ATD has very little influence on the movement of air in a treated space. The air approaching an exhaust ATD does so from many directions so that significant velocities only occur close to the device.

For example, with a square exhaust grille mounted in a flat surface, the approaching air velocity at a distance equivalent to the length of one of its sides is approximately 10% of the core velocity. Thus it is only where occupants are likely to be situated close to exhaust ATD's that consideration needs to be given to the air movement generated.

For typical core velocity see Table 3.3.

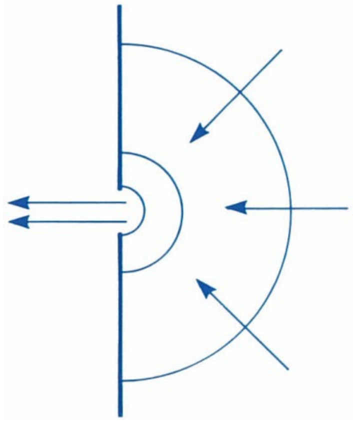


Figure 3.37 Exhaust air velocity profile

Location	Core Velocity
Above occupied zone	4m/s and above
Within occupied zone away from occupants	3-4m/s
Within occupied zone near occupants	1-3m/s
Door Transfer	1-2m/s

Table 3.3 Typical core velocities

3.5.3.1 Pressure drop

Refer to manufacturer's data to select maximum allowable, whilst also taking into account damper setting requirements, etc.

3.5.3.2 Location

There are other factors, however, in addition to those mentioned above which influence the choice of location of exhaust ATD's. There are advantages in locating exhaust ATD's in the zone of the warmest room air for cooling applications or of the coolest air for heating applications. For similar reasons locations close to areas of heat gain or loss may be beneficial. To avoid short circuiting, the exhaust ATD should not be positioned too close to the supply ATD's in the direction of air discharge.

For displacement ventilation systems extracts should be placed at the highest level in order to extract the warmest air. For low ceiling areas, it is also preferable to position extracts above large heat sources. This will minimise contaminant descent from any recirculating air.

3.5.3.3 Grilles and Diffusers

Given the total air flow rate (q_t), room dimensions and the air flow pattern in the treated space determine:

The number of ATD's by consideration of paragraphs 3.5.3.1. to 3.5.3.2. However, at least one ATD should be provided per treated space.

Air flow rate per ATD by dividing the total air flow rate (q_t) by the number of ATD's selected.

Referring to manufacturer's data, select suitable size of ATD by using the flow rate per ATD and one or more of the parameters below:

(a) Noise
(b) Core Velocity
(c) Pressure Drop

Note
Short circuiting occurs when the supply air is exhausted from the treated space before, either reaching design room temperature, or providing sufficient energy to induce satisfactory room air movement.

3.5.3.4 Transfer grilles

Where exhaust ATD's are used in order that air can pass from one treated space to another, such devices are termed transfer grilles and are normally mounted in doors.

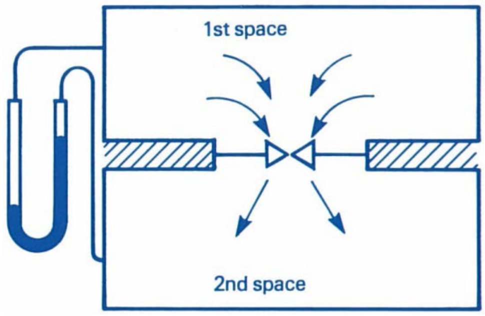


Figure 3.38 Transfer grille

The selection and positioning of transfer grilles, in relation to the first treated space can be dealt with generally in accordance with paragraph 3.5.3.

Chapter Four

4 SOUND CHARACTERISTICS

4.1 INTRODUCTION

The intention of this chapter is to cover sufficient acoustic technology to help engineers in the selection of terminal devices and the interpretation of manufacturers' literature. There are a number of text books giving introductory information concerning sound power, sound pressure level, frequency and rating system.

- Noise Control in Building services. A Fry. Sound research laboratories Ltd
- Noise Control in Industry. Sound Research Laboratories Ltd
- Acoustics :An Introduction . H Kuttruff. CRC Press
- Room Acoustics. H Kuttruff . CRC Press
- Handbook of Noise Control. C M Harris. McGraw Hill
- Engineering Noise Control Theory and Practice. D A Bies C H Hanson. Spon Press
- Acoustics . L L Beranek. American Institute of Physics Rev Sub Edition 31/12/1986
- Noise Reduction. L L Beraneck. McGraw Hill
- Noise and Vibration Data. Trade and Technical Press
- Handbook of Noise and Vibration Control. Trade and Technical Press

In building design acceptable noise levels rank equally with other major environmental requirements in defining an acceptable space or room criteria.

In designing a ventilation or air conditioning system consideration is normally given to the control of noise within the ductwork system, if necessary, by the use of duct silencers. In addition, air terminal devices such as grilles and diffusers, can themselves generate some degree of noise related to the velocity of the air passing through them. However, being the component directly connected to the room or treated space, the air terminal device must be selected not only to provide the desired aerodynamic performance but also to ensure compatibility with the environmental noise requirements. Failure to do this can result in an unacceptable noise level with the resulting necessity of changing the air terminal device to one of different size or shape which can entail obvious penalties at a later stage in the building project.

The success or failure of an installation can depend as much on acoustic performance as on achieving good room air diffusion. Thus it is important that the engineer selecting air terminal devices should have an understanding of the basic acoustic principles affecting their installation, their size and their type.

4.2 SOUND IN ENCLOSED SPACE

Given a knowledge of the physical nature of sound and of the subjective rating of sound pressure levels, one can consider the sound power level generated by an air terminal device and radiating from it into a given space.

Three main factors are involved in the conversion of sound power to sound pressure.

i)Distance from the ATD - see Figure 4.1. This is based on spherical radiation of sound waves and thus as the distance from the source increases then the sound pressure level decreases.

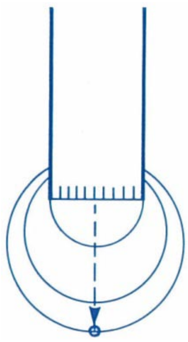


Figure 4.1 Spherical Radiation from a source in free space

ii) Directivity - see Figure 4.2. This is related to the size of the ATD and the number of adjacent boundary surfaces which cause a restriction in the arc over which sound waves can radiate. This locally increases the sound pressure level at a fixed point in space. This characteristic is a function of frequency.

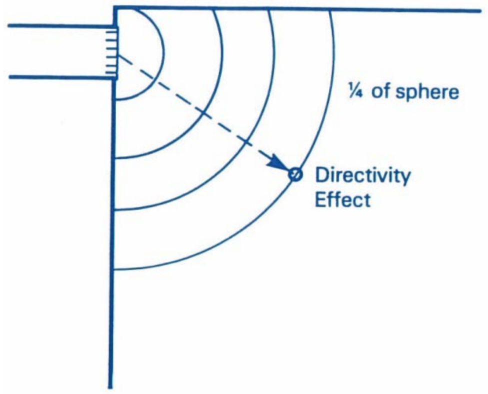


Figure 4.2 Directivity

iii) Room constant - see Figure 4.3. This describes the capability of the boundary of the room and furnishings to absorb or reflect sound waves. It is frequency dependent and related to the average acoustic absorption coefficient of all surfaces. For the same source a hard surfaced room will have a higher sound pressure level than a room with carpet and furnishings.

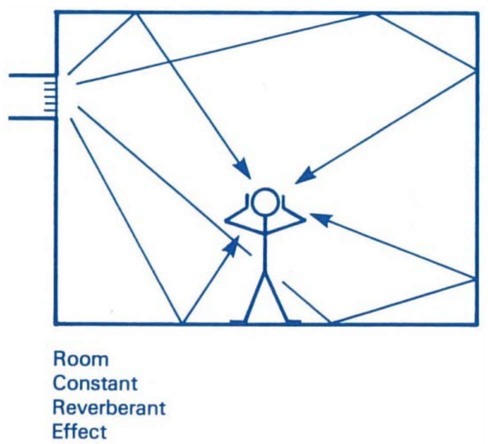


Figure 4.3 Room Constant Reverberant Effects

Factors (i) and (ii) are usually referred to as the direct component of the sound pressure level and factor (iii) as the reverberant component. The resultant sound pressure level at a point in space is the logarithmic summation at each octave band of these two components.

Figure 4.4 shows a typical relationship of the above as a function of distance from the source.

The above phenomena can be described in mathematical terms enabling calculations of the resultant sound pressure level as a function of frequency. This is beyond the scope of this booklet, and in cases where noise level requirements are very low or critical a specialist in this field should be consulted. However, there are approximate methods that can be employed in the initial design stages and these are described in paragraph 4.4.

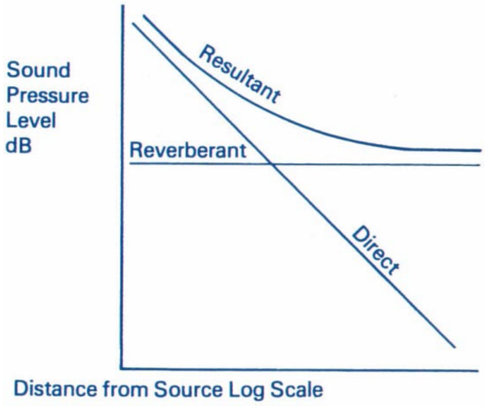


Figure 4.4 Variation of Sound Pressure Level in Room as a function of Distance from the Source

4.3 NOISE GENERATION CHARACTERISTICS OF AIR TERMINAL DEVICES

Noise generation is caused by the increase in air velocity and turbulence as the air flow passes around solid obstructions such as deflecting vanes, etc. (See Figure 4.5). The sound power level generated by these mechanisms can be approximately expressed as follows:

$L_w = K_x \text{ Log } V_v + 10 \text{ Log } A + K_y$

Where
V_v = Discharge velocity in m/s

A = Aerodynamic free area of ATD in m^2

L_w = Overall sound power level in dB re 10^{-12} watts

K_x and K_y are constants, being a function of the sound, frequency, ATD type and design.

Sound power levels can be evaluated using a number of subjective rating systems for example NR, NC or dBA. This is particularly useful when determining room noise levels. See section 4.4.

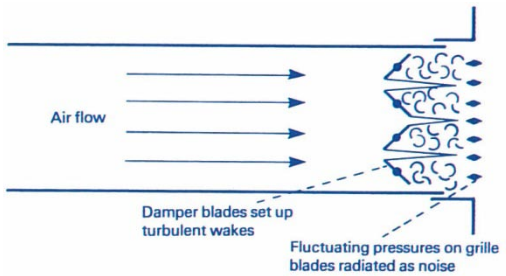


Figure 4.5 Mechanism of Noise Generation by Dampers and Grilles

The previous equation may then be rewritten as:

$L_wNR = K_1 \log V_v + 10 \log A + K_2$

Where K_1 and K_2 are constants, similar to K_x and K_Y, but relate to the subjective NR rating (see table 4.1).

L_wNR = NR rating of sound power levels

L_wNC = NC rating of sound power level

L_wdBA = dBA rating of sound power level

Typically –

L_wNR approximately equal to L_wNC

L_wNR approximately equal to L_wdBA - 5

Also all the foregoing data are based on air flow dampers being fully open.

As has already been indicated, control dampers on the back of ATD's are only intended to provide a fine adjustment of the air flow quantities by imposing additional pressure drop, which results in additional turbulence and, hence, noise. See Figure 4.5.

The increase in noise level due to closure of opposed blade, flap or radial vane dampers can be approximated from the following equation:

For supply ATD's $\Delta dB = 30 \log P_1/P_0$

For extract ATD's $\Delta dB = 16 \log P_1/P_0$

Where ΔdB = increase in overall sound power level or NR sound power level.

P_0 = total pressure drop through the damper plus grille - damper fully open.

P_1 = total pressure drop through damper plus grille - damper partially closed.

Figures 4.6.1 and 4.6.2 gives plots of Log functions for velocity, area and damper pressure drop ratio, this in conjunction with the constants given in Table 4.1 provides a simplified means of noise prediction.

From examination of Figures 4.6.1 and 4.6.2 the following should be noted:

i) With the same ATD doubling of discharge velocity increases the noise level by approximately 18dB from 38 to 56dB in the example.
ii) With the same discharge velocity doubling the ATD area the noise level increases by approximately 3dB, i.e. this is very insensitive in comparison with the velocity characteristic.
iii) A doubling of total pressure drop through an ATD damper combination by means of partial closure of the damper results in (a) supply + 9dB, (b) extract + 5dB.

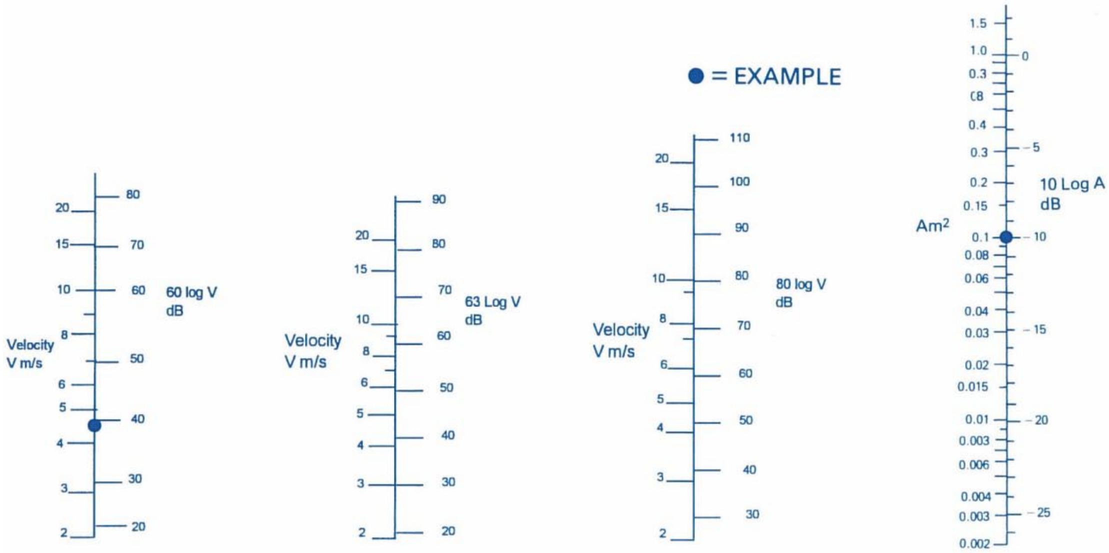


Figure 4.6.1 Data for Basic ATD Noise Prediction

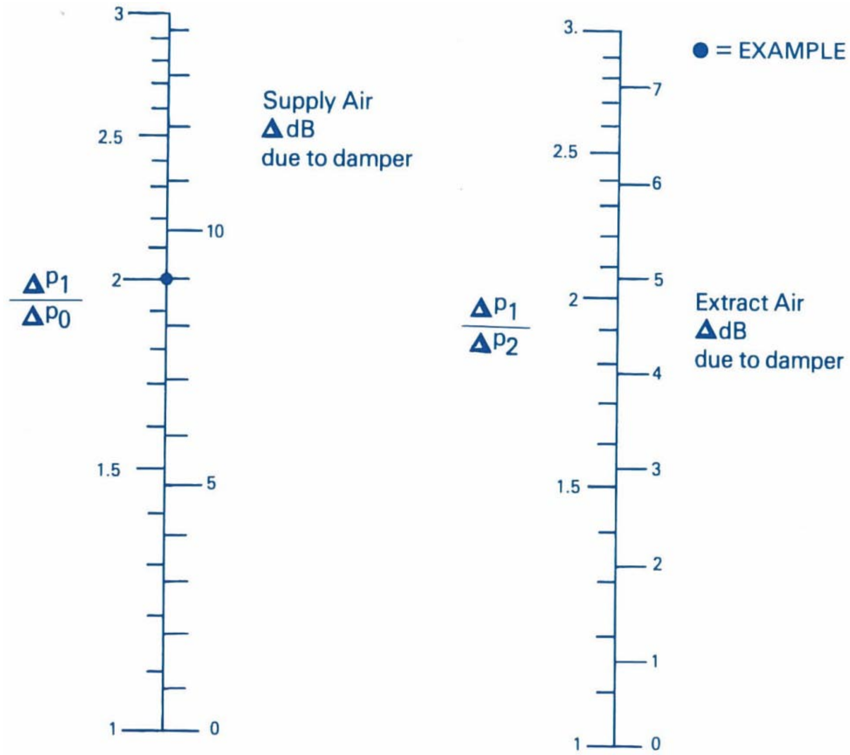


Figure 4.6.2 Increase in ATD noise due to closing Damper

When supply grilles with movable blades are employed, there is additional increase in noise due to blade deflection, shown typically in Figure 4.7.

BLADE DIVERGENCE 1st Row	2nd Row	INCREASE IN NOISE LEVEL dB
0	0	0
0	22.5°	1
0	45°	3
22.5°	22.5°	2
45°	45°	6

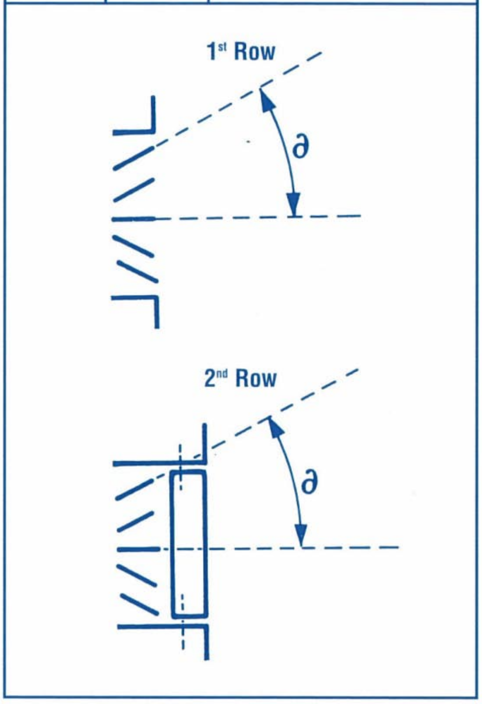


Figure 4.7 Effect of Blade Divergence

4.3.1 Example of Subjective Sound Power Level Prediction

Given:
Supply air grille
Flow rate= 0.45m^3/sec
Discharge velocity= 4.5 m/s
Aerodynamic free area= 0.1m^2

Opposed blade damper fully open P_0= 15 Pa

Required pressure drop at 0.45 m^3/s P_1= 30 Pa

From Table 4.1:
L_wNR = 60 log V + 10 log A - 3

From Figure 4.6:
L_wNR = 39 - 10 - 3 = 26dB

P_1 / P_0 = 30/15 = 2

ΔdB = 9dB

L_wNR = 26 + 9
= 35dB

Type of ATD	K_1	K_2
Grilles and Linear Grilles (Supply Air)	60	-3
Grilles and Linear Grilles (Extract Air)	60	-3
Circular & Square Diffusers	60	-3
Linear Diffusers	60	0
Slot Diffuser	60	+10
Perforated Plate Diffuser	80	-8
Swirl Diffusers	63	+3
Floor Diffusers - vertical Discharge	60	+6
Floor diffusers - Horizontal Discharge	60	+9
Displacement Diffusers For sidewall or free standing units the noise characteristics are a strong function of internal diffuser design. This varies significantly from manufacturer to manufacturer and hence no unique values for K_1 and K_2, may be determined.		
NR Sound Power Level: $L_w NR = K_1 \log V + 10 \log A + K_2$ (Damper fully open) $L_w NC$ approximately equal to $L_w NR$ $L_w dBA$ approximately equal to $L_w NR$ +5		

Table 4.1 Constants for the prediction of subjective ratings of sound power levels

4.4 DETERMINATION OF SPACE NOISE LEVELS DUE TO ATD SELECTION

Having determined the sound power characteristics either from manufacturers' data or from the approximate method given in Section 4.3., it is now necessary to take account of the room characteristics to determine sound pressure levels at a given distance from the ATD. The principles involved have been described in Section 4.2.

Method 1 - Specialist Approach - Within 3dB

A detailed analysis of the direct and reverberant components as a function of octave bands of frequency can be carried out. This is normally required in cases where noise level requirements are critical. Reference should be made to an acoustics engineer for this service.

Method 2 - Graphical - Within 6dB

This is based on a graphical solution and can be undertaken by evaluating the following relationships.

Room NR level = $L_w NR + C_1 + C_2 + C_3$

Assume an office block with a room 10m x 5m x 3m carpeted floor, acoustic tiled ceiling. 3 off ATD's serve the space positioned in the walls at ceiling/wall junction. Take the conditions given in the example in 4.3.1., i.e.

Subjective Rating of Sound Power Level $L_w NR$ = 35dB

Where C_1, C_2 and C_3 are corrections based on room volume and acoustic characteristics, the distance, from ATD location and number of ATD's and can be determined as follows:

Determine NR level at say 2.0m from ATD's.

1. Acoustic rating of space - see Table 4.2.

Due to carpet and acoustic tiled ceiling take room as Medium Dead.

2. Room Volume.

Volume = 10 x 5 x 3 = 150m^3

3. C_1 from Figure 4.8 using space rating room volume and Distance from ATD.

C_1 = -8dB

4. Directivity Correction C_2 which depends on ATD location. See Table 4.3.

C_2 = +6dB (at wall/ceiling junction)

5. C_3 from table 4.4 based on number of ATD's

C_3 = + 5dB (3 off ATD's). See table 4.4

NR Rating = 35 - 8 + 6 + 5 = 38

NC Rating approximately equal to 38
dBA Rating approximately equal to 38 + 5 = 43

RATING	USE OF ROOM
Dead Room	Radio and TV Studios audiometric rooms
Medium Dead	Restaurants, offices and boardrooms with absorbent ceiling and floor covering, hotel bedrooms
Average Room	Standard offices, libraries, hospital wards (rooms with no special acoustic treatment)
Medium Live Room	School rooms, lecture theatres, art galleries and public houses
Live Room	Churches, swimming baths, factories, operating theatres, large canteens and gymnasiums

Table 4.2: Typical room characteristics

MOUNTING POSITION	C_2
Free space	0dB
Flush with one surface	+3dB
Junction with two surfaces	+6dB
Junction with three surfaces	+9dB

Table 4.3: Directivity index

Method 3

The final method is one which is quite often used in manufacturers' literature to give a very approximate idea of room subjective ratings as follows:

Room subjective rating = Sound Power Level subjective rating - 8dB

Where 8dB is referred to as the room correction.

Care must be taken when using this technique as when more than one ATD is involved in large spaces the actual room correction can be in the region of 2 to 3dB. If noise is considered important then either methods 1 or 2 should be adopted.

A general indication is given in Table 2.1 of the relationship between various types of ATD and what, subject to check by the above procedures, is normally considered to be ‘quiet’ and ‘commercially quiet’.

If after calculating the space noise level, this should prove to be too high, then the following possible courses are open to improve the situation.

i) Increase the sizes of the ATD- whilst a greater area will cause a slight increase in noise, the resultant velocity reduction for the same air flow will give an overall noise reduction. If this course is followed, the aerodynamic performance should then be rechecked.

ii) Increase the number of ATD’s of the same size - whilst the increased numbers will result in a small increase in noise level, the resultant reduction in velocity through each individual ATD will again predominate to give an overall noise reduction. Again the aerodynamic performance should be rechecked.

iii) Use an ATD with a greater free area - again the reduction in air velocity will predominate to give an overall noise reduction.

Where large number of ATD’s of similar type are to be checked for noise characteristics, it is usually more convenient to establish a maximum discharge velocity assuming the siting and number of terminals serving a specific space. This is acceptable because the velocity effect on noise is dominant. If when using this method the selection approaches the limiting air velocity, then the case should be subjected to the more rigorous analysis in method 2 above.

When using a damper in conjunction with an ATD, it is good practice to allow an additional 2 to 3dB before finalising selection, but should be subjected to the more rigorous analysis as in method 2 if greater accuracy is required.

4.5 EFFECT OF SYSTEM DESIGN

It is important to realise that the noise apparently radiated from an ATD can either be as a result of:

a) Noise generated at the ATD itself, or

b) Noise propagated down the ductwork from upstream sources such as main control dampers, constant and variable flow rate devices, or

c) Noise generated from poor plenum box design.

d) Another influencing noise source

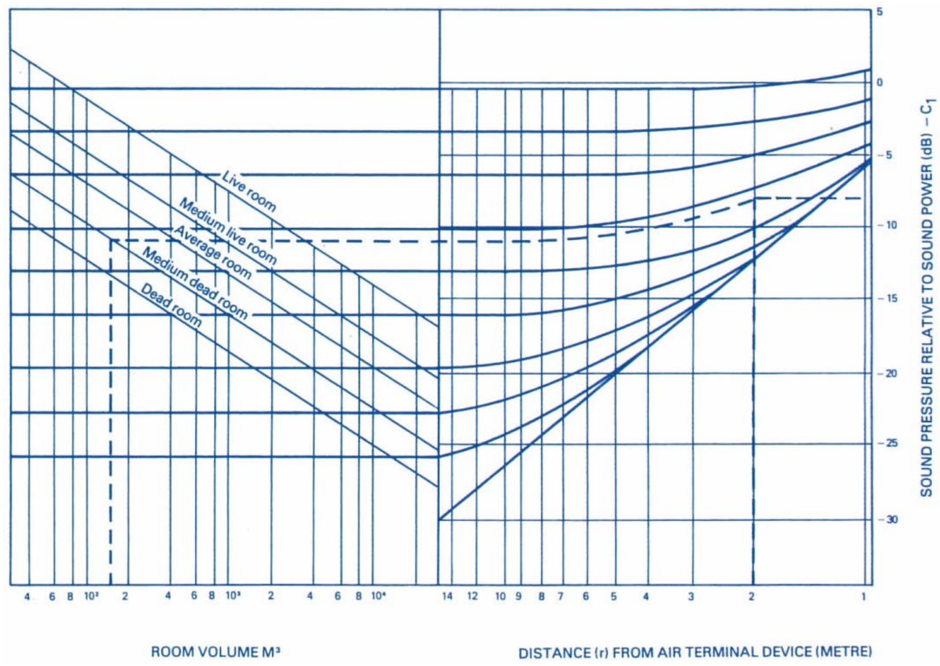


Figure 4.8 Correction for receiving space

Number of ATDs	2	3	4	5	6	7	8	9	10
dB addition	3	5	6	7	8	8	9	10	10

Table 4.4 Correction for number of ATDs C_3

4.5.1 Duct and Plenum Design

As already indicated, air velocity through an ATD provides the dominant contribution to the noise level generated at that device. With a poorly designed plenum or approach ductwork connected to a supply ATD, it is quite possible, for example, for the bulk of the air flow to pass through half the ATD area, thereby doubling the discharge velocity. Not only would this be undesirable from a room air diffusion viewpoint but also the effective noise generation could, depending on exact product calculation, increase by approximately 15 to 20dB.

In cases where plenum boxes are fitted to the back of ATD's, linear grilles, slot diffusers etc., care must be taken to ensure that noise generated in the plenum box, due to flow discharge from the spigot, does not make a significant contribution to the overall noise generation.

Typical methods of avoiding this type of problem are:

i) Air velocity in the spigot should not exceed the ATD discharge velocity.

ii) Maximum spigot velocities for NC/NR 35-40 should not exceed 4 m/s, but are typically selected at 3 m/s at approx NR30.

iii) If there is doubt concerning plenum noise generation, the plenum box should be lined acoustically on the inside with an adequate thickness of fibrous material, suitably covered and installed to prevent particle migration.

In cases where very low noise levels are required, i.e. below NR30, then it is wise to locate the dampers not adjacent to the ATD but some distance upstream followed by secondary attenuation. Care should be taken in the design of the attenuator and the duct connecting the attenuator outlet to the ATD to ensure that local velocity high spots do not occur at the ATD which would again result in increased noise level. Figure 4.9 illustrates a number of points made above.

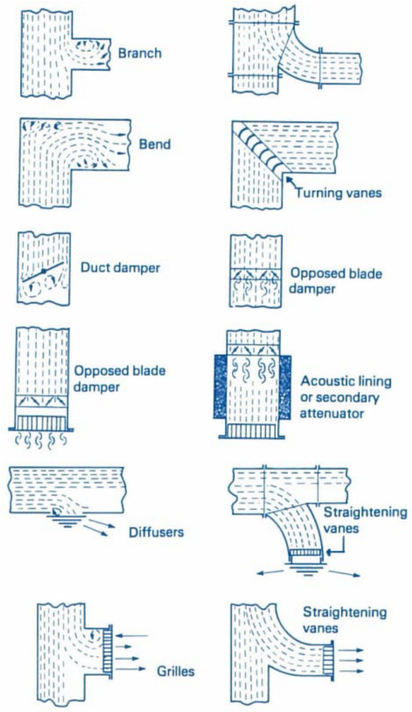


Figure 4.9 Uneven Velocity Profiles

4.6 OTHER NOISE SOURCES

If a noise level problem in a space has been identified when a project has reached commissioning stage, it is recommended that the following basic procedures are employed as a process of elimination in order to determine whether or not the problem relates to the ATD.

1) Shut down in rotation the various plants serving the space in question to identify which system is creating the noise. Take care not to be misled by the possible air imbalance created within the space, i.e. supply only creating noise due to air escaping through doors, etc.

2) On the noisy system, check position of ATD dampers. If these are well closed, open dampers. If the noise level is reduced then damper generated noise is the probable cause of the problem. Solution - consider introduction of dampers as far as possible upstream of the ATD to provide required pressure drop for balancing. This may also involve the use of secondary silencers downstream of the new damper.

3) If opening and closing the ATD damper does not result in any significant variation in noise level then either the noise is entering the space through some other path, i.e. structure borne vibration, direct transmission through walls, ceiling or floor, duct breakout, etc., or is propagating down the ductwork from another noise source.

4) Remove ATD plus associated damper from duct. If the noise level reduces then the sources of the problem may well be the ATD and damper. However, if the noise level increases significantly, then again, it is likely that duct borne noise is propagating down the ductwork system from some upstream source such as main control damper, primary fans, poorly designed ductwork junctions etc.

Should the above investigation indicate that the noise problem is associated with the ATD damper plenum box combination, then consideration must be given to such remedial measures typically as:

i) Replacing ATD with one of greater free area.

ii) Replacing ATD with a larger device.

iii) Redesign the plenum box (possibly increasing the number of spigots).

iv) Increase the number of ATD’s.

As can be seen from the above, the cost implications of remedial action should the noise problem occur with an ATD, emphasises the importance of adequate consideration being given in the design stages to correct selection of ATD’s to ensure compliance with noise specification.

Chapter Five

5. DUCT ENTRY CONDITIONS

5.1 GENERAL

Air terminal devices are selected on the assumption that there are no irregularities of the velocity profile at the face of the air terminal device. It follows that duct entry conditions are equally as important as the size and type of air terminal device.

Unequal velocity profiles at the face of an air terminal device can give rise to the following:

- Excessive turbulence
- Unpredictable throw and spread.
- Breakdown of wall/ceiling attachment.
- High noise levels.
- Extremely difficult balancing procedures

Recommendations to assist in the correct entry condition design to air supply devices are shown in the subsequent sections.

The following designs can be used even in situations where there is limited duct space availability.

Each design illustrated employs the principle of the Total Pressure to Velocity Pressure ratio P_t/P_v, this principle being based on simple Bernoulli theorem. This allows the correct balance of P_v to P_s (static pressure) by introducing P_t as a vector component. ($P_t = P_v + P_s$)

Although dampers or pressure reducing devices are not always shown, their effects should always be taken into consideration when establishing the P_t/P_v ratio.

The following are used in the subsequent figures.

TPu -Total pressure in duct before take-off

VPu -Velocity pressure in duct before take-off

TPb -Total pressure in plenum box

VPb -Velocity pressure in plenum box

Vu -Average duct velocity before take-off

Vc -Average velocity in connector duct

Vb -Average velocity in plenum box

Øn -Neck diameter for circular diffusers or equivalent dia for rectangular diffusers

Vn -Average velocity in neck

Exhaust air devices have not been specifically indicated, there being insufficient data currently available for the losses of fittings used for return air. The friction and dynamic losses for straight duct and duct elements are the same on the return side as the supply, but at junctions of branch and main ducts the dynamic losses result from the jointing rather than the dividing of air streams. The present practice is to employ the same methods as for supply units but with reverse airflow.

5.2 DIFFUSER CONNECTIONS

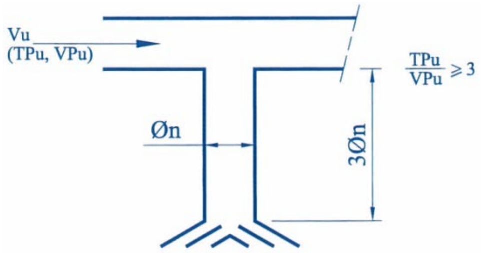


Figure 5.1 Branch connection (long)

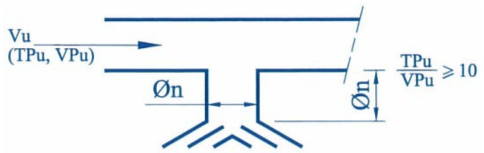


Figure 5.2 Branch connection (short)

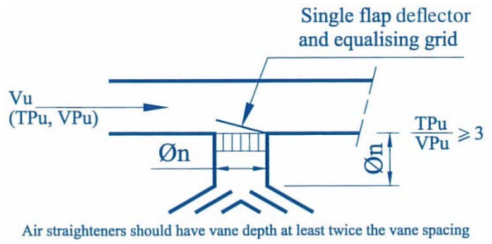


Figure 5.3 Branch connection (flap deflector)

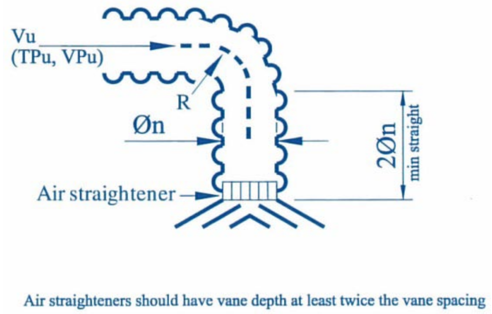


Figure 5.4 Flexible connection

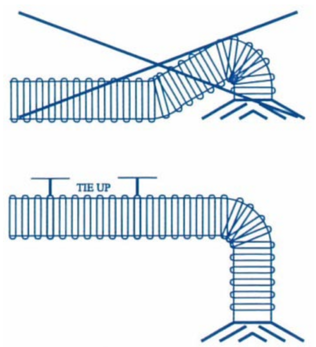


Figure 5.5 Flexible connection - good practice to ensure correct results

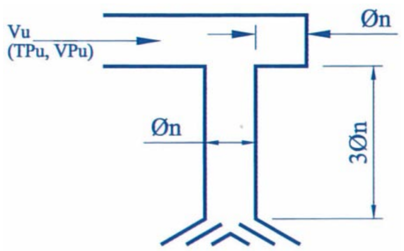


Figure 5.6 End connection

End connections are as branch connections with identical pressure ratio restrictions.

N.B. The length of straight duct between a 90° 1.5d radiused bend (radius to centre line) and the diffuser depends on the pressure ratio for branch connections.

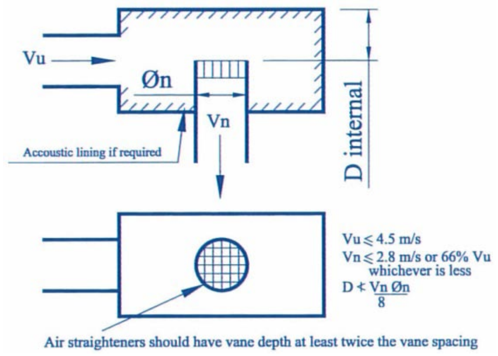


Figure 5.7 Connection for use where height is Restricted

5.3 GRILLE CONNECTIONS

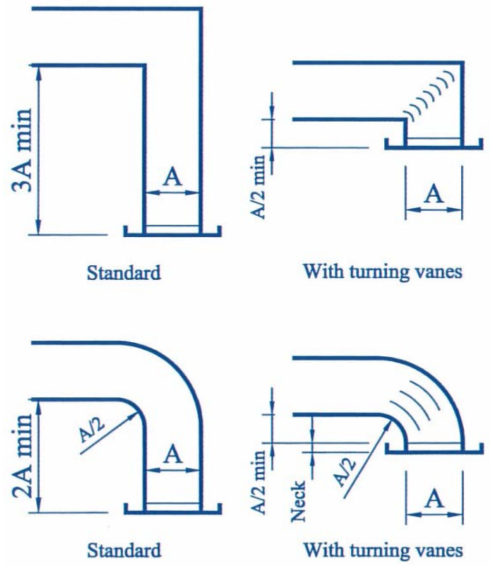


Figure 5.8 After rectangular or radiused bends

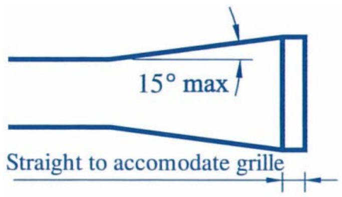


Figure 5.9 With change section connector

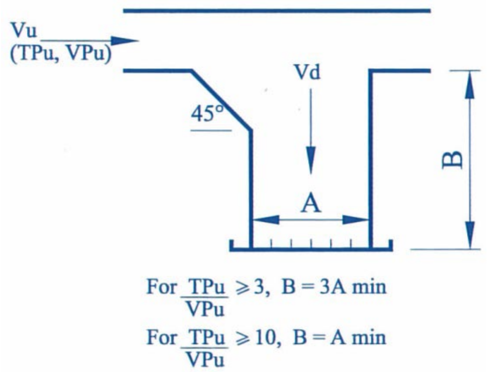


Figure 5.10 Longitudinal section with plenum header box

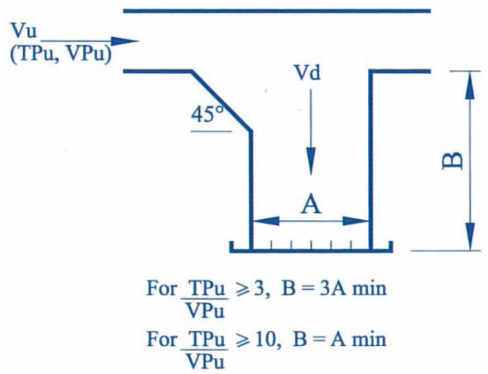


Figure 5.11 From branch duct

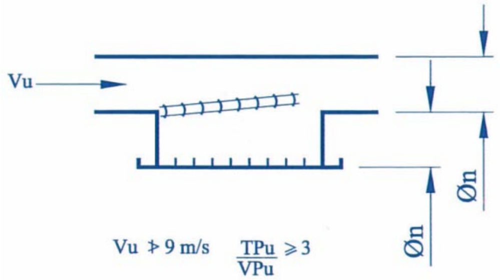


Figure 5.12 From branch with deflector

5.4 LINEAR GRILLE/DIFFUSER CONNECTIONS

5.4.1 Straight boxes

With few exceptions, these should be designed as plenum header boxes. The cross sectional aspect ratio of the header boxes should not exceed 4:1.

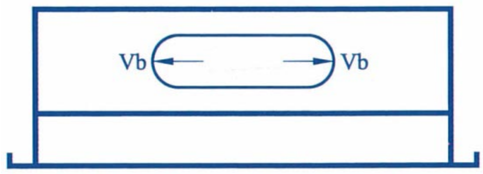


Figure 5.13 Straight box side entry (entry view)

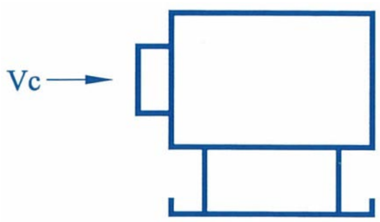


Figure 5.14 Straight box side entry (side view)

The average velocity in the connection (Vc) should not exceed 4.5 m/s.

The cross sectional area of the box should be such that the TPb/VPb ≥ 3 at each side of the connection.

TPb should be the TP of the air terminal device plus 15%.

If the box exceeds 2m in length, multiple connections can be used, but at not greater than 2m distances.

If the air may impinge directly on the air terminal device a perforated plate of approximately 50% free area should be used.

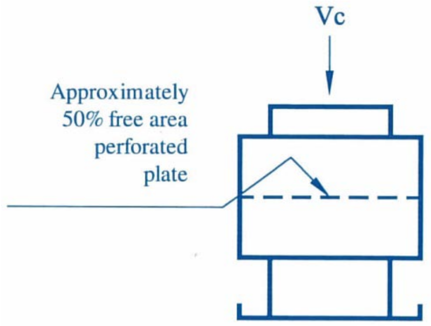


Figure 5.15 Straight box top entry (side view)

5.4.2 Tapered boxes

Tapered boxes maintain the velocity pressure and keep the TPb/VPb ratio constant.

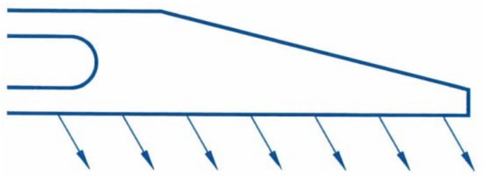


Figure 5.16 Tapered box (side view)

The air stream is not perpendicular to the air terminal device and a spreading pattern results.

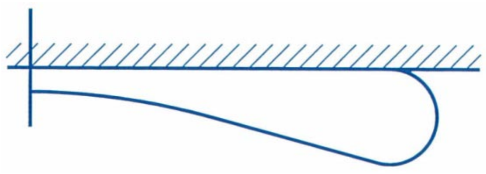


Figure 5.17 Tapered box air pattern

5.5 DISPLACEMENT FLOW DIFFUSERS

The duct entry conditions for a displacement flow diffuser depend upon the internal design of the specific model. This varies considerably from manufacturer to manufacturer and a generic condition is not applicable. Please refer to manufacturers' data for the relevant information for the model selected.

Chapter Six

6 THE FIXING AND INSTALLATION OF AIR TERMINAL DEVICES

There is a wide range of air terminal devices and applications for them and a wide variety of fixing methods have therefore evolved. The principal current fixing methods are illustrated in the following sub sections. It is advisable to use fixing details issued by the manufacturer where these are available.

In addition to the following groups, there are ATDs such as nozzles, disc valves, etc., which have their own proprietary fixing methods and therefore have not been included in this section.

In general, ATDs are shown in their normal orientation of use, i.e. ceiling, wall or floor mounted. Where details are specifically not suited for other orientations this is made clear in an accompanying note.

6.1 SQUARE, RECTANGULAR AND LINEAR GRILLES

6.1.1 Visible fixing

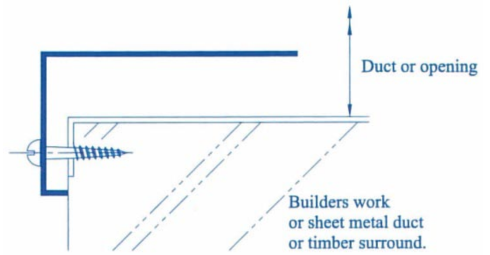


Figure 6.1 Flange screw fixing

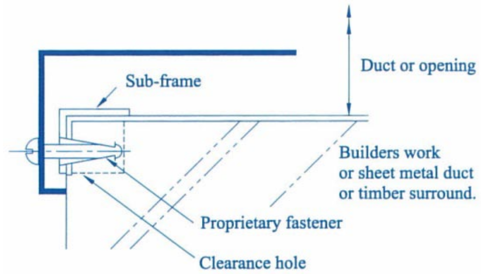


Figure 6.2 Sub-frame/quick release fastener
Normally not recommended for ceiling use

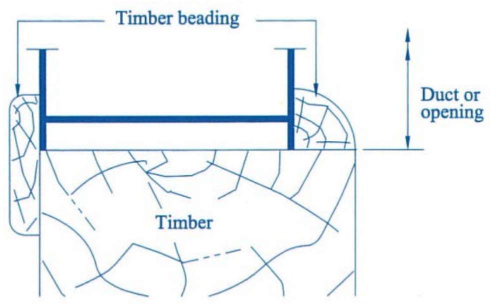


Figure 6.3 Beading (transfer grilles)

6.1.2 Concealed fixing

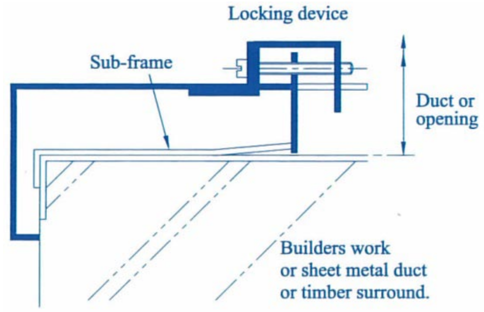


Figure 6.4 Sub-frame/locking device

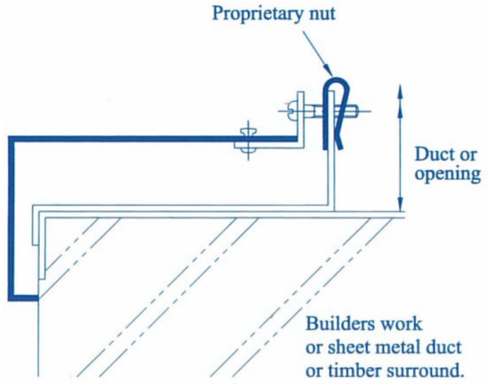


Figure 6.5 Sub-frame/Rear bracket

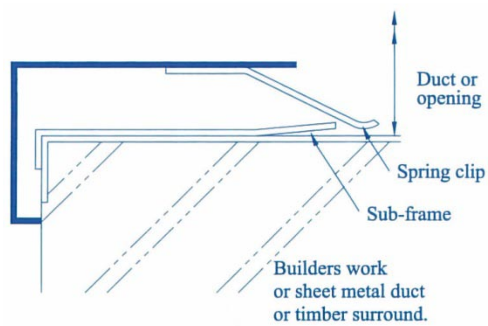


Figure 6.6 Sub-frame/Spring clip
Not recommended for use in ceilings or doors

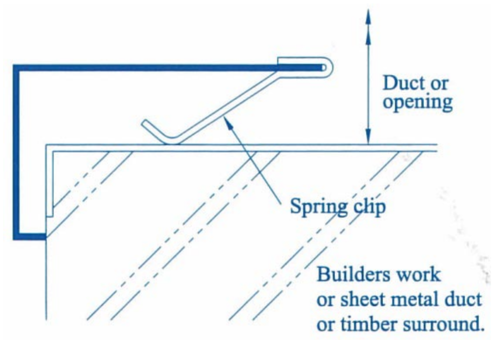


Figure 6.7 Spring clip
Not recommended for use in ceilings or doors,or where distortion of surround can occur

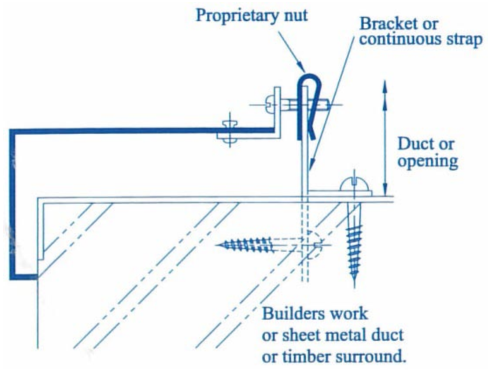


Figure 6.8 Rear bracket or strap

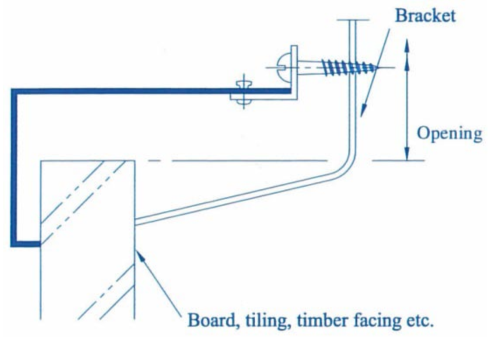


Figure 6.9 Rear bracket and saddle bracket

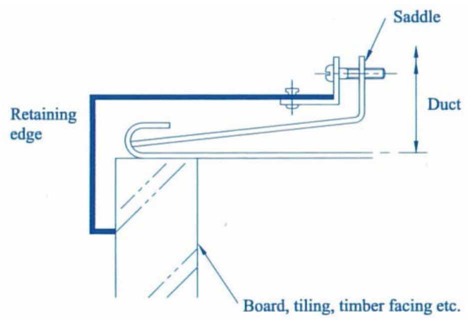


Figure 6.10 Saddle bracket into duct

6.2 LINEAR AND SLOT DIFFUSERS

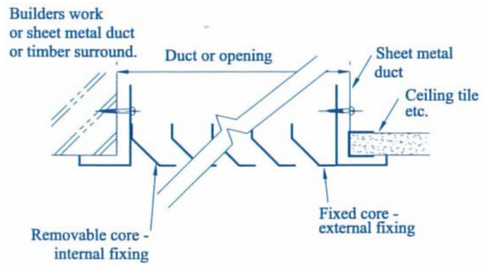


Figure 6.11 Hidden screw fixing

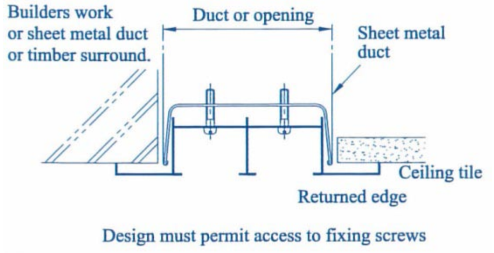


Figure 6.12 Bridge or saddle bracket

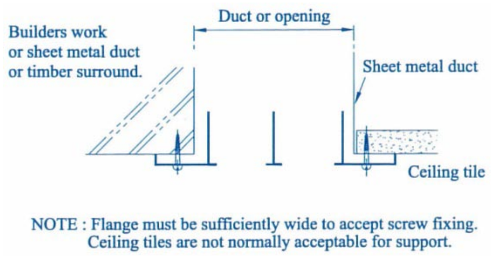


Figure 6.13 Flange screw fixing

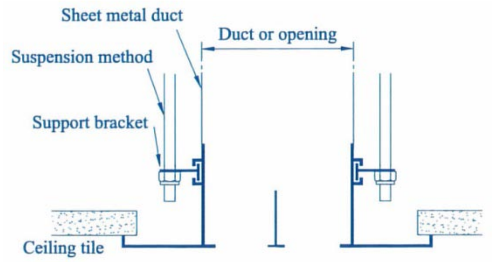


Figure 6.14 Rear support bracket

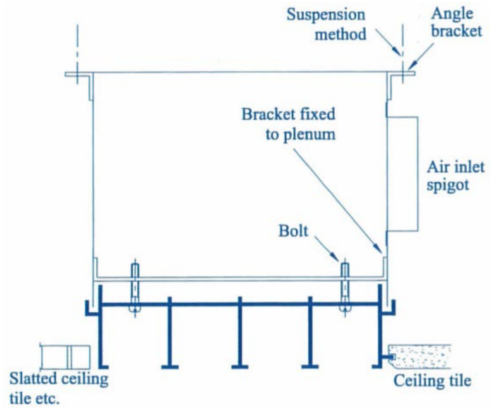


Figure 6.15 Internal suspension bracket

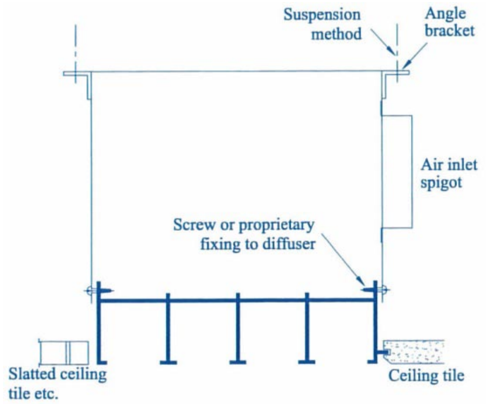


Figure 6.16 Angle brackets from plenum box

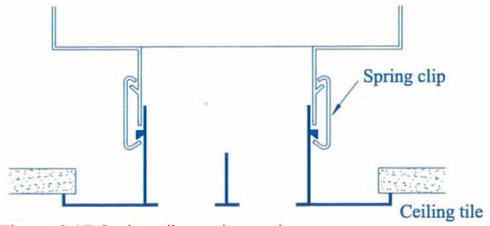


Figure 6.17 Spring clip to plenum box

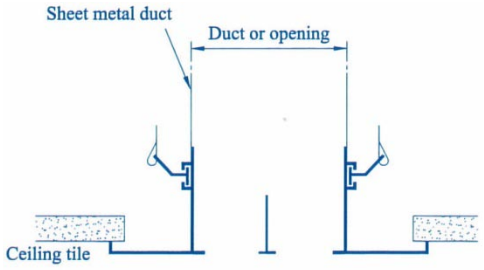


Figure 6.18 Wire support
Not for use in walls

6.3 CIRCULAR AND RECTANGULAR DIFFUSERS

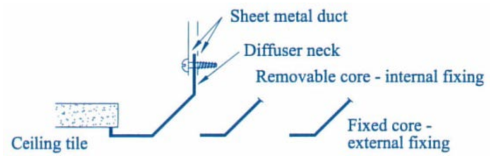


Figure 6.19 Hidden screw fixing

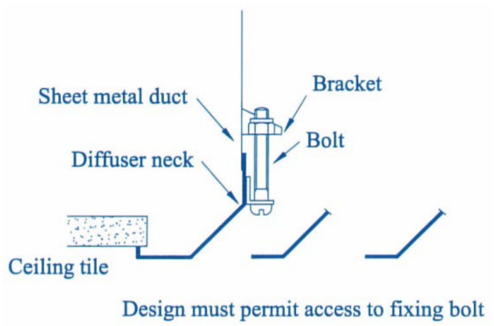


Figure 6.20 Suspension bolts and brackets

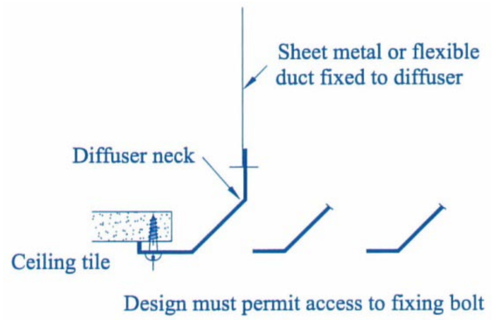


Figure 6.21 Flange screw fixing
Ceiling must be of a suitable material

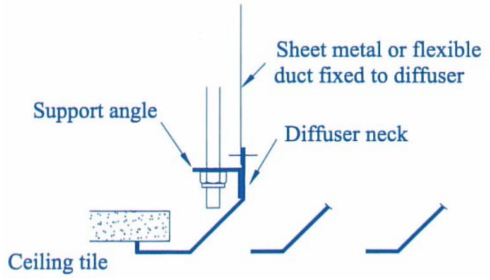


Figure 6.22 Rear support angles
a) Suspension method

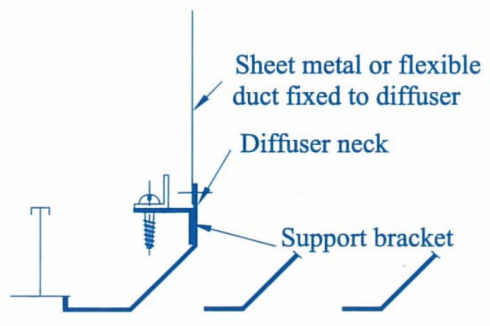


Figure 6.22 Rear support angles
b) Angle support from Tee bar

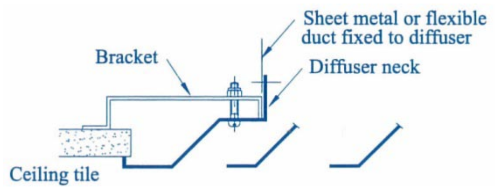


Figure 6.23 Rear suspension brackets
Ceiling must be of a suitable material

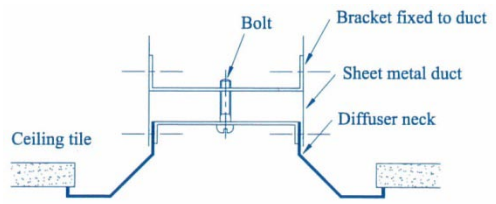


Figure 6.24 a) Internal suspension bracket

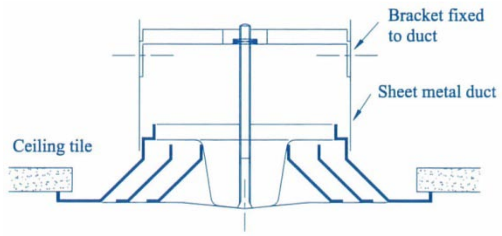


Figure 6.24 b) Screw access either through core or with core removed

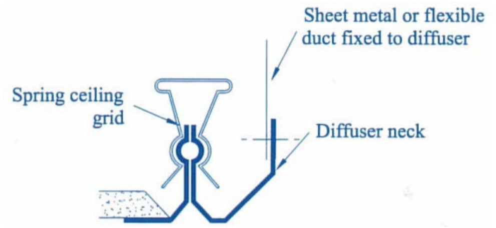


Figure 6.25 Spring edge clip

6.4 FLOOR MOUNTED GRILLES AND DIFFUSERS

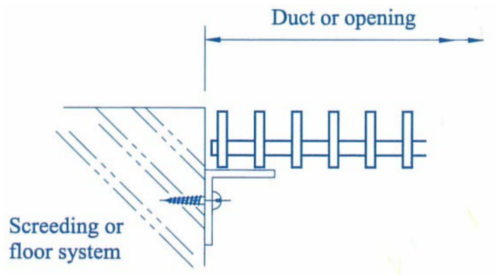


Figure 6.26 Drop in angles support
(Linear grilles only)

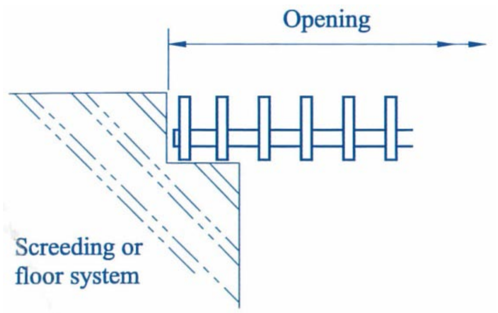


Figure 6.27 Drop in recessed support

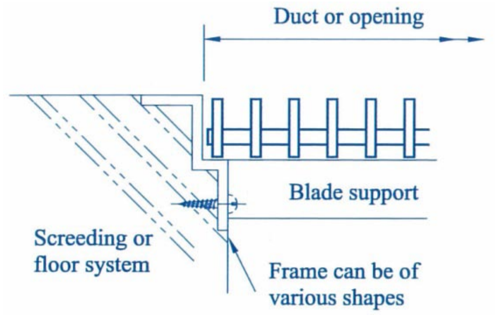


Figure 6.28 Drop in frame support

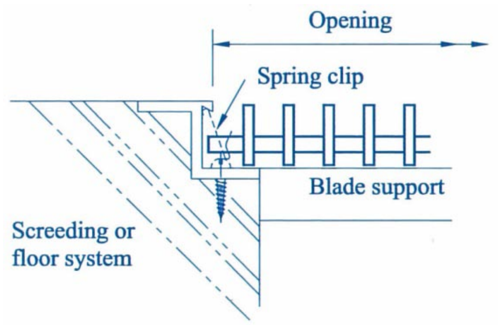


Figure 6.29 Drop in spring fixing

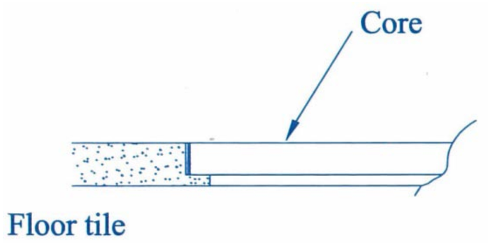


Figure 6.30 Swirl diffuser (circular)
a) Lay in core

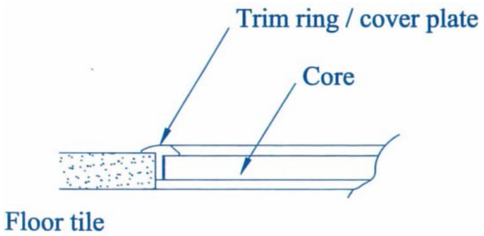


Figure 6.30 Swirl diffuser (circular)
b) Core and trim ring/cover plate

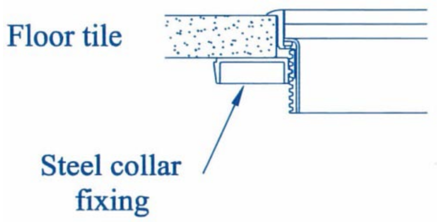


Figure 6.30 Swirl diffuser (circular)
c) Back collar fixing

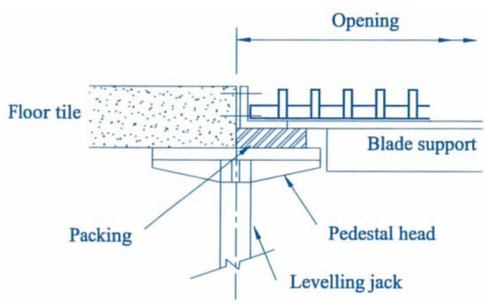


Figure 6.31 Floor grille with jacks

6.5 INSTALLATION

It is important that all ducts or fixing openings are regular in shape and correctly dimensioned.

When carrying out installation ensure that:

- Specific manufacturer's instructions are followed.
- The ATD fits snugly into the opening without distortion.
- There is sufficient material around the fixing points to give a secure fixing into the duct, ceiling or building fabric.
- Suitable fixings are used for the material to which the ATD is to be fixed.
- All joints are correctly sealed using self adhesive foam strip or similar to avoid air leakage.
- Mastic or similar is not used except on sub-frames which are not intended to be removable.

- Visible edge joints have a neat finish.
- Floor mounted ATDs should be of an appropriate design to reduce risk of tripping and suitable for any loads it may be exposed to.
- Sharp edges and protruding screws are not left where they could pose a safety risk in normal use or building maintenance.
- Once fitted, all moving parts operate correctly and removable cores can be taken out and replaced.
- Safety wires with quick release ends must be used on removable cores where ATDs are installed in high level or ceiling applications.

Chapter Seven

7 MEASUREMENT ON SITE

Although there now exists a large range of sophisticated instrumentation that makes site measurement easier, quicker and more comprehensive, there is a lot of older, more basic, equipment still in everyday use. In some instances, for example resolution of dispute, it is common to revert back to fundamental devices to obtain the definitive measurement. For these reasons this section attempts to cover both old and newer technology instruments.

7.1 MAINTENANCE AND USE OF INSTRUMENTATION

All test equipment shall be in accordance with the specifications laid down in BS EN 13182, BS EN 12238 and BS EN 12239.

To ensure a high level of confidence in test data, instruments must have been checked and calibrated within the recommended time period.

It is the responsibility of the test engineer to make sure that, where a choice is available, the best instrument is selected to suit the measurement to be taken.

For consistency it is advisable that the series of readings taken during a test from one or a number of instruments should be completed by the same individual.

Vibration will upset the accuracy of most instruments. Care must be taken to ensure that all instruments are placed on a vibration free surface.

Many instruments are temperature and/or ambient pressure sensitive. Adequate time must be allowed, before readings are taken, for the conditions to stabilise. Recognition of the appropriate correction factors should be made.

Ancillary equipment, such as lights, tables, platforms, should be arranged in such a manner that the best possible resolution of reading can be obtained from the instrument in use.

7.2 PRESSURE MEASUREMENT

There are two basic types of pressure measuring instruments used in the H & V industry:

(i) The liquid filled manometer in which the application of pressure is counterbalanced by the weight of a column of liquid.

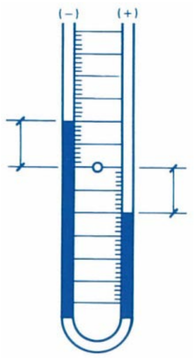


Figure 7.1 U tube manometer

(ii) The dry gauge in which the application of pressure causes the mechanical displacement of a diaphragm or bellows which is sensed electronically or counterbalanced by a spring system with an attached pointer.

Figure 7.2 Electronic instrument

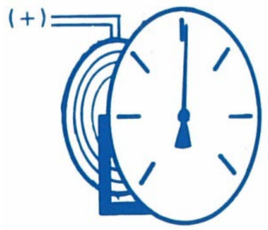


Figure 7.3 Dial gauge,mechanical

7.2.1 Liquid filled instruments

The fundamental instrument for pressure measurement is the vertical or inclined liquid in glass manometer. The sight glass must be of precision bore in either glass or plastic material. The liquid must be of known relative density, low viscosity and low surface tension.

Vertical manometers are made with either single or double limbs. In the double limb arrangements both sides of the ‘U’ are visible. The pressure is read by summing the change of liquid height in each limb either by reading against a fixed scale or by micrometer pointers adjusted to just touch the liquid surfaces (Hook Gauge).

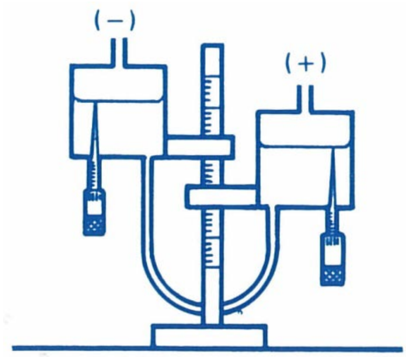


Figure 7.4 Hook gauge

In the single limb arrangement only one limb of the ‘U’ tube is visible. The “hidden” limb is usually of very much larger cross-section (a tank) and the measuring scale divisions are calculated to take into account the level change in the tank as well as the relative density of the liquid.

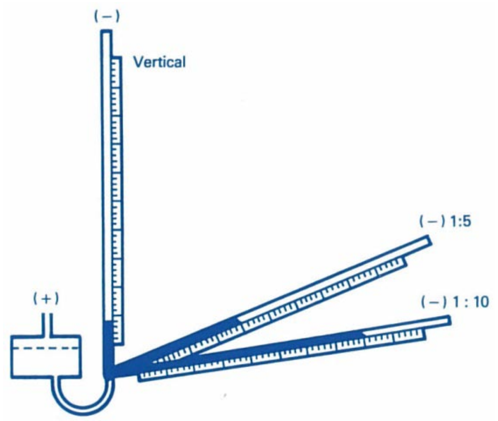


Figure 7.5 Single limb manometer showing various inclinations

Inclined manometers are usually of the single limb arrangement and offer the option of various angles of inclination.

Pressures below 500Pa should be read on an inclined manometer. The instrument should be frequently checked for level and zero where these facilities are available.

When a choice of “incline” is provided, a trial run will ascertain the lowest incline (and hence the reading of highest resolution) that may be safely used without over ranging the instrument with consequent loss of manometer fluid.

Check for polarity and overload, since these may lead to a discharge of liquid into the manometer pipelines, which if not cleared will cause false readings. Great care should be taken to ensure that the pipelines are always clear, dry and as short as practical.

When reading a concave meniscus (e.g. paraffin, alcohol, etc.) always measure to the lowest part of the liquid surface. Poor illumination can cause dark line shadows within the meniscus which, if misread, can lead to inaccurate results.

A variation of the single limb vertical manometer incorporates a transparent scale suspended from a float in the smaller section limb. The change in height of the scale resulting from the application of pressure is viewed through a high powered telescope. (Betz)

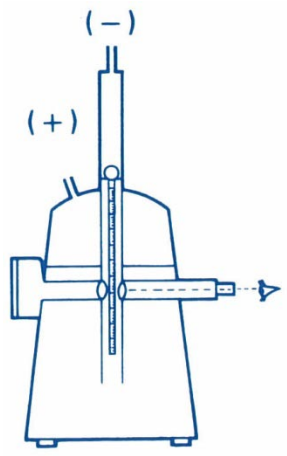


Figure 7.6 Floating scale manometer

A further variation of the ‘U’ tube allows for the change in liquid level in two interconnected reservoirs, caused by the application of pressure, to be restored to a datum mark by raising or lowering one of the reservoirs by a calibrated screw thread.

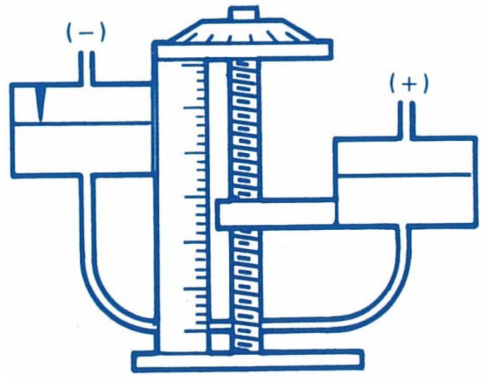


Figure 7.7 Null gauge

7.2.2 Dry Manometers

The basic form of the dry gauge converts the deflection of a diaphragm or bellows by mechanical linkage to the movement of a pointer over a calibrated scale.

An alternative linkage with less resistance to movement is obtained by magnetic means.

More sophisticated versions of the dry gauge convert the diaphragm movement into an electrical signal (transducer) which may be remotely displayed in either analogue or digital form. More commonly instruments of this type contain an integral electronic pressure sensing element. In addition to display of pressure reading many instruments are also capable of giving direct indication of velocity when used with a pitot static tube, or similar device. Volumetric flow rates can also be displayed by some instruments with the additional input of a characteristic area.


Figure 7.8 Volume flow indicating instrument

Particular care must be taken to ensure that gauges of this type are shielded from any form of shock loading or vibration during use.

The calibration can be upset by the incorrect orientation of this type of gauge. It is advisable to ascertain the plane in which calibration was carried out.

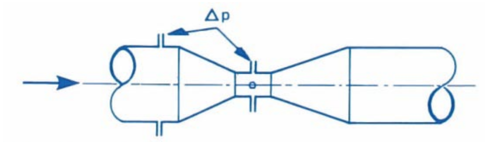
As with liquid filled instruments it is advisable to keep pressure connecting pipe lines as short as practical.

7.2.3 Other methods of pressure measurement

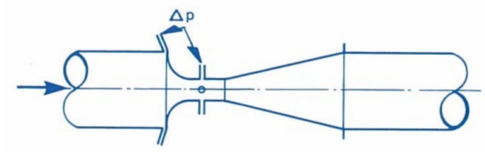
A variation of the transducer type of dry gauge depends upon the measurement of leakage flow rate through a calibrated orifice by means of either the rate of heat loss from a heated element or the mechanical deflection of a pivoted spring loaded vane.

7.3 FLOW RATE MEASUREMENT

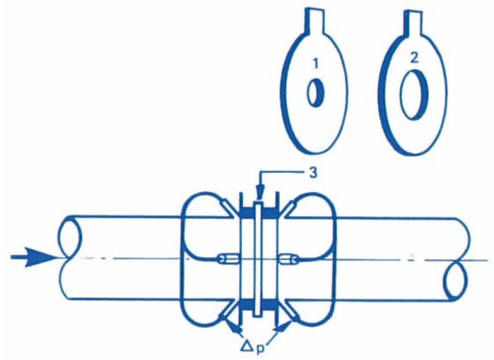
The preferred methods of flow rate measurement are specified in BS EN ISO 5167-1 and employ either the orifice plate, venturi tube, nozzle or nozzle venturi to create a pressure differential signal proportional to flow.



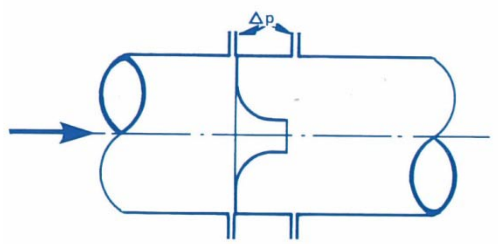
Classic venturi



Nozzle venturi



Orifice plate assembly



Nozzle

Figure 7.9 Differential pressure volume flow measuring devices

It is advised that the calculations and tolerances laid down in BS EN ISO 5167-1 will only apply provided all the dimensions and conditions up and down stream as specified are observed. If these recommendations cannot be followed or greater accuracy is required then in situ calibration will be necessary.

The range of flow rates accurately measurable by these methods is limited by the published Reynolds Number. In the case of the orifice plate method this can be overcome by preparing a series of easily exchangeable orifice plates whose sizes can offer overlapping flow rate ranges. Care must be taken to ensure accurate centralising and that airtight sealing is obtained when plates are changed.

Flow Grids

Figure 7.10 Wilson flow grid

A less costly device for in duct measurement is the flow grid. These devices range from two simple cross tubes, to multiple tube and manifold devices, which sample the flow with upstream and downstream facing sensing holes. Either differential pressure, or bypass flow-meter, techniques are used to measure volume flow.

Resistance to flow in ducts is less with these devices than orifice or venturi meters.

Since the flow is sampled at the sensing holes, which are distributed across the duct cross section, good mean flow values are measured. For this reason such devices are often more tolerant of less than optimum meter sitting within ducts following disturbances such as bends, branch ducts etc.

7.3.2 Capture Hoods

Figure 7.11 Capture hood

For flow rate measurement with air terminal devices capture hoods provide a convenient method. These devices are pushed against the surface surrounding the air terminal device thus making a seal and capturing the entire flow. The flow is channelled through the measurement section of the device where it is measured via a grid section and either a differential pressure method or bypass flow meter.

The capture hood is capable of making measurements in either supply or exhaust mode.

Some capture hoods, particularly those which use mechanical sensing methods, may be orientation sensitive. Care should therefore be exercised to ensure calibration data is valid in use whether this be for measuring vertical flows, from ceilings or floors, or horizontal flows from wall grills or diffusers.

The flow patterns associated with different grilles and diffusers can also significantly alter the calibration factor of some capture hoods. Swirl diffusers can give rise to particular problems.

If in any doubt about calibration validity ensure that the measuring device is calibrated under conditions as near as possible to those encountered in use.

In order to reduce the effect the measurement device has upon the ventilation system care should be exercised to ensure the capture hood chosen does not exert excessive back pressure on the system at the flow rates to be measured.

7.4 FLOW RATE MEASURED BY VELOCITY AREA METHODS

(see BS EN 1042-2.2: 1983, ISO 7145: 1982 (*Measurement of fluid flow in closed conduit - velocity area methods - method of measurement of velocity at one point of a conduit of circular cross section*) and Section 2.4.2 of this guide).

The air flow rate in a duct may be assessed by averaging the velocity readings obtained by traversing across the duct in the log Tchebychef distribution of points. (see method 10 of BS EN 1042-2.2 section 2.1)

Care must be taken to locate the velocity measuring head precisely at the points specified.

7.4.1 Flow rate measurement by vane anemometer

The vane anemometer may be used successfully to assess the air flow rate in a duct and at the entry/exit of a duct by the point traverse method.

It may also be used with a calibrated hood, provided due allowance is made for the additional resistance imposed by the hood on the system.

Due to the blockage effects and the difficulties of taking readings close to the duct wall the anemometer is rarely suitable for accurate measurement in small ducts, but may be used to detect proportional changes in flow rate following accurate measurement by other means.

Where the anemometer head forms part of the duct run and swirl conditions are eliminated it may be used as a permanent flow monitor when suitably calibrated.

If the anemometer projected face area is greater than 1/100 of the duct area the manufacturer's blockage correction factor should be applied.

Note should be taken of the manufacturer's calibration data, particularly at low speeds.

7.5 VELOCITY MEASUREMENT

There are four types of velocity measurement instruments readily suitable for use in the HEVAC market:

(i) Pitot static tube.
(ii) Heated element anemometer.
(iii) Vane anemometer
(iv) Ultrasonic anemometer.

7.5.1 Pitot static tubes

The primary standard instrument for measuring velocities above 2.5 m/s is the pitot static tube. The use of the instrument and the calculations involved are laid out in BS EN ISO 5167-4: 2003 and ISO 3996.

Inspect the tube carefully before use for damage or blockage of the small holes in the head.

Always ensure that the head is correctly aligned in the airstream.

The differential pressure output signal from the pitot static tube may be connected to either a suitable manometer for direct reading or a transducer, etc., for processing. Both manometery and electronic pressure reading instruments are available with direct read out in velocity.

If excessive pressure fluctuations exist and make reading of the manometer difficult additional damping may be obtained by using a resistance, which is linearly proportional to velocity in the pressure lines. Alternatively when using an electronic manometer damping factors are sometimes available.

7.5.2 Heated element anemometer

The most suitable instrument for measuring velocities below 2.5 m/s is the heated element instrument. The element is usually of the fine wire type or thermistor type. Both rely for their operation on the measurement of rate of energy loss caused by the flow of air past the element.

This type of instrument is very sensitive to low velocities but natural convection effects can become significant at flow velocities below 0.25m/s.

The instrument is also sensitive to changes in ambient temperature and pressure. The temperature variations can either be allowed for internally or by use of a correction chart. Pressure variations usually have to be corrected from manufacturers data.

Heated element anemometers are also direction sensitive and care should be taken to correctly align the device with the direction of flow to be measured to ensure reproduction of the instruments calibrated performance. This can be particularly relevant where trying to measure a specific velocity vector with an omni-directional probe where significant polar calibration variations often exist.

The elements are usually very fragile and subject to drift in calibration. Frequent calibration checks are advised for accurate results.

Care should be taken to ensure that the element is clean since trapped particles of dust and dirt can affect the heat transfer characteristic and give false readings.

Versions of thermal anemometers are available where a characteristic area may be input and volume flow displayed.

7.5.3 Vane anemometers

The vane type anemometer (already considered when discussing flow rate measurement) overlaps both the above velocity ranges.

The direct reading (electronic) instrument may be used to measure velocities both in large airways and at duct inlets/outlets. Care must be taken when observing low velocities to allow the vane assembly to attain full speed before taking a reading. Similarly where the instrument is designed to time average, care must be taken to hold the vane assembly in the airstream for the whole of the time averaging period.

The mechanical instrument may be used to measure a velocity by holding it at the required station for a suitable time interval. With instruments where the stop/start lever is close to the path of the airstream being measured, disturbance may be caused by its operation. It is then advisable to note the time between a predetermined number of counts whilst the instrument is left running.

The vane anemometer should be regarded as an obstruction in the air path. The usual calibration assumes that the instrument will be used in free space. If the instrument is used with a hood assembly or in a confined space such as a duct whose cross-sectional area is less than 100 times the projected face area of the anemometer, then knowledge of special calibration and pressure drop will be necessary for accurate results.

Versions of vane anemometers are available where a characteristic area may be input and volume flow displayed.

7.5.4 The Pivoting Vane Anemometer

The principle of a pivoting vane anemometer is dependent on the velocity pressure detected by the instrument probe forcing a small flow of air along a large bore flexible tubing in order to impinge on a lightly pivoted vane within the instrument case. The deflection of the vane is related to the bleed flow rate and the velocity is indicated by a pointer moving across a calibrated scale. The range of the instrument is varied by restricting the bleed flow rate.

Several different shapes of probe are usually available with this instrument. Selection is dependent upon the site situation and is adequately described in the manufacturer’s literature.

Care must be taken to hold the instrument case in the plane in which it has been calibrated.

Care must be taken to ensure that the flexible tubing is not so bent or “kinked” as to restrict the bleed flow rate.

The instrument must be handled with care since the vane assembly is very delicate and the calibration may be impaired by accidental impact.

7.5.5 Ultrasonic Anemometer

Figure 7.12 Ultrasonic Anemometer

Ultrasonic anemometers are capable of making velocity measurements over a wide range of velocities since they have an inherently linear response to velocity. They do not suffer from thermal device low velocity natural convection errors, vane anemometer low velocity bearing friction or differential pressure device high velocity compressibility effects.

Ultrasonic measurements, if configured to make corresponding measurements in both the upstream and downstream directions, are also insensitive to changes in the flow media such as; temperature, pressure, humidity or gas composition.

Due to the very fast sensing activity these devices are also able to measure turbulence values.

7.6 TURBULENCE INTENSITY MEASUREMENT (BS EN 13182)

Personal comfort within an environment depends not only upon limited velocity of air, especially in the occupied zone, but also the level of turbulence intensity associated with the air flow.

Turbulence intensity is defined as the ratio of the standard deviation of the air velocity to the mean air velocity and is expressed as a percentage. Hence in areas of disturbed air, with no significant mean value velocity vector, turbulence intensity values can reach values of several hundred percent. Personal comfort depends upon an acceptable combination of air temperature, mean velocity and turbulence intensity. See PD CR 1752:1999 and BS EN ISO 7730: 2005.

7.7 TEMPERATURE MEASUREMENT

There are three standard sensors for temperature measurement:

(i) Mercury-in-glass thermometer.
(ii) Resistance thermometer.
(iii) Thermo-couple.

All readings taken may be classified as either surface temperature or air temperature.

When measuring air temperature care should be taken that the temperature sensor is fully immersed in the airstream. When the sensor is used inside a protecting sleeve the temperature difference due to heat loss down the sleeve should be allowed for by calibration.

When measuring surface temperatures care should be taken that the sensor is in pressure contact with the surface to be measured and covered on the exposed side to reduce heat losses. It is not advisable to use the mercury-in-glass thermometer without preparing a pocket or recess which will completely contain the mercury bulb.

When measuring metallic surface temperature the contact (open circuit) thermo-couple may be used. Care must be taken that the probe and the surface are thoroughly clean to ensure good electrical contact.

It is always necessary to allow sufficient time for the sensor to attain a steady value of the surface or fluid (air) temperature being measured before taking a reading.

TYPE	OPERATION	MEASURES OR RELAYS	RANGES (TYPICAL)	CALIBRATION	SETTING	ACCURACY IN FIELD
Liquid filled manometer	Adjustable inclined or vertical liquid column is displaced by applied pressure	Pressure	0-125Pa 0-250 0-500 0-2500	By manufacturer	Level instrument and set zero	± 1.0% of reading or ± 1 Pa whichever is the greater
Diaphragm pressure gauge	Various types using dry diaphragms, spring and magnet assembles to deflect a face pointer	Pressure	0 – 100 to 0-5000 (Pa)	By approved test agent (annually)	Set zero	± 2% of full scale deflection
Pitot static tube	Specially designed coaxial probe	Static and total velocity pressure	4-82 m/s	None	None	± 2% of reading (with suitable manometer)
Rotating vane anemometer electronic & mechanical 75-125mm	Rotating vanes relay impulses to electronic signal processor (remote head)	Velocity	0.12 – 2.5 to 0.12 – 25 (m/s)	By approved test agent (annually)	Set required range	± 2% of reading or ± 0.1 m/s whichever is the greater.
Miniature vane Anemometer electronic 10-25mm	Rotating vanes relay impulses to electronic signal processor (remote head)	Velocity	0.3 – 5 to 0.3 – 20 (m/s)	By approved test agent (annually)	Set required range	± 5% of reading or ±0.2 m/s whichever is the greater
Deflecting vane anemometer	Remote jets relay air pressure to case mounted deflecting vane calibrated for velocity	Velocity	0 – 1 0 – 10 0 – 20 (m/s)	By approved test agent (annually)	Set zero Set range	± 10% full scale deflection
Thermo-electric anemometers	Hot-wire or thermistor probe exhibits resistance/temperature characteristic which is electronically calibrated for air velocity/cooling effect	Velocity	0 - 0.5 to 0 – 20 (m/s)	By Manufacturer	Set zero Set range	± 3% of reading or ± 0.1 m/sec whichever is the greater
Ultrasonic anemometer	Ultrasonic anemometer Ultrasonic pressure wave times of flight used to compute flow velocity	Velocity	0 – 50 (m/s)	By Manufacturer	None	± 1% of reading

Care must be taken, when making measurements approaching zero velocity, that the instrument calibration is valid.

Table 7.1. Instrumentation for velocity and performance measurement

Chapter Eight

8 REGULATION OF AIR TERMINA L DEVICES

8.1 APPLICATION

This section gives guidance on the regulation of air flow rates through air terminal devices. The more comprehensive task of commissioning is covered in the CIBSE Commissioning Code A for air distribution systems, and is defined as “the advancement of an installation from the stage of static completion to full working order within specified requirements”.

These procedures may be applied to those parts of supply and exhaust systems which handle air at low velocity.

The instruments and techniques considered are those in common use at the time of publication and represent good or recommended practice.

8.2 MEASUREMENT, INSTRUMENTATION AND EQUIPMENT

Regulation of air flow rate passing through an air terminal device entails the measurement of a characteristic air velocity or pressure, which may be measured by various techniques.

The instruments detailed in Table 7.1 are those most commonly used for field measurements and have in the past proved reasonably reliable, accurate, robust and portable.

MEASUREMENT STATION	TECHNIQUES					
	1. Velocity traverse	2. Point velocity reading	3. Face velocity	4. Calibration hood	5. Characteristic velocity calibration	6. Calibrated terminal loss coefficient
Main/branch ducts	✓					
Connecting stub ducts to terminals	✓	✓				
Supply Grilles			✓	✓	✓	
Linear slot supply diffusers				✓	✓	✓
Perforated face diffusers	✓			✓	✓	✓
Exhaust grilles			✓	✓	✓	
Exhaust slots	✓			✓	✓	
Swirl diffusers	✓			✓		✓

Table 8.1 Air flow measuring techniques

It is important that all instruments are regularly maintained and recalibrated by the supplier or an approved agent and that suitable test records are kept. If instruments are well maintained, they will provide the commissioning staff with accurate test results and thus ensure an acceptable system balance (see Section 7).

8.3 MEASURING TECHNIQUES

All the following flow rate measurement techniques require the measurement of air velocity or pressure at a characteristic location either in a main, branch or stub duct or at the air terminal device itself. See table 8.1.

A velocity reading, or more usually the average of a set of velocity readings, may be used to calculate air flow rate, thus:

$$q = \bar{v} \times A$$

where q = air flow rate (m^3/s)

$\bar{v}$ = average air velocity (m/s)
A = area of air flow in measurement Plane (m^2)

The area “A” may be duct cross-sectional area, hood outlet area, nominal grille outlet area or effective terminal outlet area, depending on the technique used. It is possible to measure the absolute air flow rate or a proportional or indicated air flow rate.

8.3.1 Absolute Air Flow Rate

Air velocities are measured at a plane of known cross-sectional or effective area and their product is equal to the actual air flow rate, which may be corrected, if necessary, for standard air density.

8.3.2 Proportional Air Flow Rate

Where a number of similar terminals are served by the same branch duct it is unnecessary to measure the absolute air flow rate; it is adequate to regulate air flow rate on one terminal as a proportion of the air flow rate at a reference terminal. In this way, each of the terminals on the branch duct are in balance and it only remains to regulate the branch flow rate with respect to the main duct flow rate (The proportional balancing method is detailed further in Section 8.6.) The “indicated air flow rate” is the product of the measured velocity (v) and the terminal area (A).

$$q = v \times A$$

8.4 IN-DUCT VELOCITY TRAVERSE

By this method, point velocity readings are taken at defined locations within an air duct usually across a fixed plane, as shown in Table 8.2. This method is a time consuming but reasonably accurate technique which is generally reserved for air flow measurements in main or branch ducts. It is however sometimes necessary with certain air terminal devices to measure the velocity at connecting or stub ducts where face or hood velocity readings are suspect.

8.4.1 Pitot Tube Traverse

The pitot tube is connected to an inclined manometer, or other suitable differential pressure gauge, and the velocity pressure is recorded at each of the designated measuring stations shown in Table 8.2.

Average duct velocity which is derived from velocity pressure measurements is equal to the arithmetic average of all the velocity readings (it is not correct to average the velocity pressure readings and convert the result to a single velocity reading). For field use, it is generally adequate to use the velocity scale supplied with the manometer as this considerably reduces the measurement and calculation time.

The pitot tube/manometer measuring technique will produce field accuracy of ±5% with a reasonably uniform duct velocity profile. Accuracy of the method will depend on the velocity profile in the duct and readings should not be taken unless at least five equivalent diameters are available both upstream and downstream of the measuring station.

As accuracy will be drastically affected by non-uniform duct flow, it is therefore recommended that a preliminary duct traverse is made with the pitot tube, prior to taking any readings; if air flow is particularly uneven, an alternative measurement station or technique should be used. Air velocities less than 4 m/s should not be measured with the common pitot tube and manometer. Further reference may be made to BS EN ISO 5167-1:2003 *Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full. General principles and requirements*.

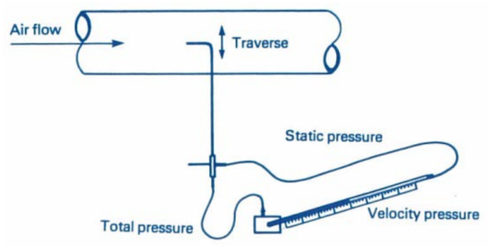
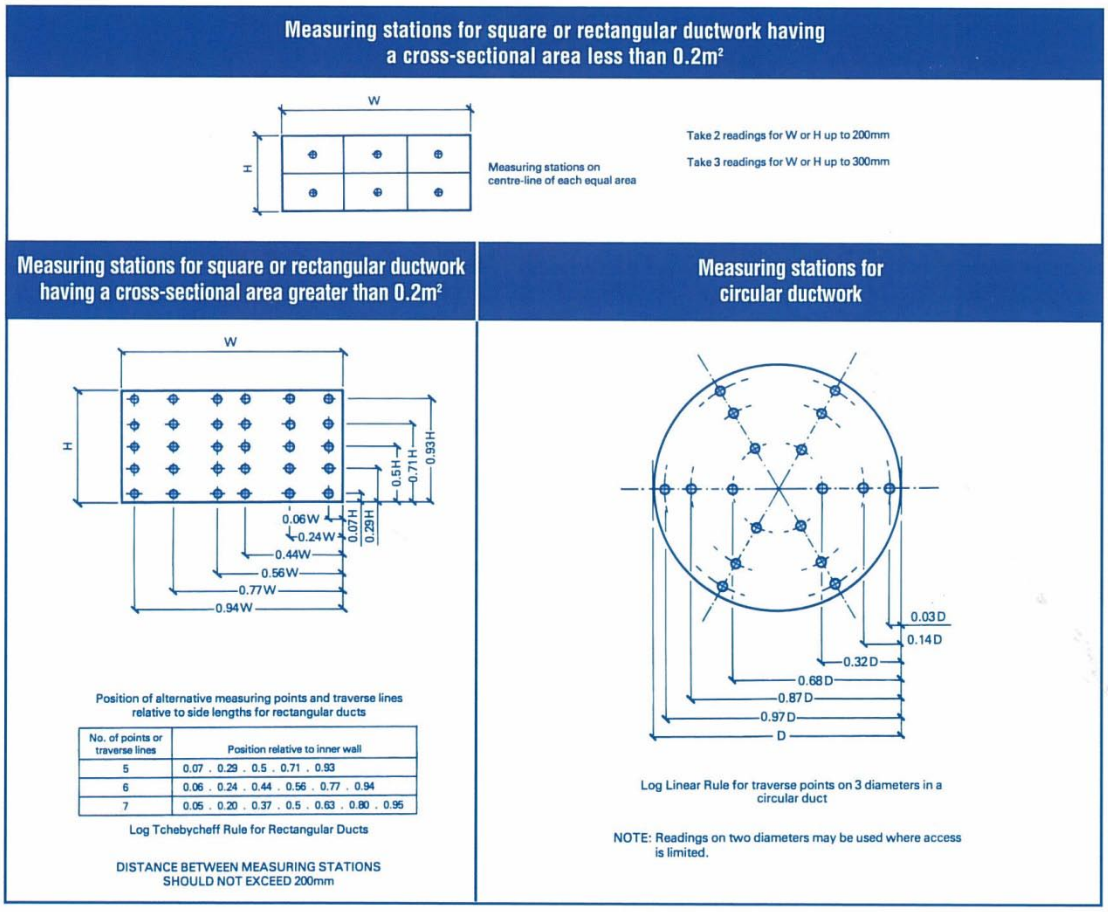


Figure 8.1 Pitot static tube traverse



No. of points or traverse lines	Position relative to inner wall
5	0.07 . 0.29 . 0.5 . 0.71 . 0.93
6	0.06 . 0.24 . 0.44 . 0.56 . 0.77 . 0.94
7	0.05 . 0.20 . 0.37 . 0.5 . 0.63 . 0.80 . 0.95

Table 8.2 Velocity measurement positions

8.4.1.1 Corrections for Standard Air Density

Air Velocity may be calculated from the velocity pressure readings according to the following formula (for standard air density = 1.2 kg/m$_3$)

$$V = 1.29 \sqrt{(P_v)}$$

For non standard air conditions, the formula becomes

$$V = 1.29 \sqrt{\left(\frac{1013}{P_o} \times \frac{T}{293} \times P_v\right)}$$

where

V = air velocity (m/s)
P_v = velocity pressure (Pa)
P_o = absolute static pressure at point of measurement (millibars)
T = absolute temperature (K) (0^oC = 273K)

Alternatively, multiply the measurement velocity by the correction factor from Table 8.3 based on the atmospheric pressure and temperature.

PRESSURE mb	AIR TEMPERATURE °C						
	0	10	20	30	40	50	60
800	1.088	1.106	1.125	1.144	1.163	1.182	1.200
820	1.073	1.093	1.112	1.130	1.149	1.167	1.185
840	1.060	1.079	1.098	1.117	1.135	1.153	1.171
860	1.048	1.067	1.085	1.104	1.122	1.140	1.157
880	1.028	1.055	1.073	1.091	1.109	1.127	1.144
900	1.024	1.043	1.061	1.079	1.097	1.114	1.131
920	1.013	1.031	1.049	1.057	1.085	1.102	1.119
940	1.002	1.020	1.038	1.058	1.073	1.090	1.107
960	0.992	1.010	1.027	1.045	1.062	1.079	1.095
980	0.982	0.999	1.017	1.034	1.051	1.068	1.085
1000	0.972	0.989	1.007	1.024	1.040	1.057	1.073
1013	0.965	0.983	1.000	1.017	1.034	1.050	1.068
1020	0.962	0.980	0.997	1.014	1.030	1.047	1.063
1040	0.953	0.970	0.987	1.004	1.020	1.038	1.052
1060	0.944	0.951	0.978	0.994	1.011	1.027	1.042
1080	0.935	0.952	0.969	0.985	1.001	1.017	1.033
1100	0.926	0.943	0.960	0.978	0.992	1.008	1.023

Table 8.3 Measured velocity multiplication correction factor

8.4.2 Point Velocity Readings

Where flow conditions in a connecting or stub duct to an ATD are particularly even it may be adequate to record a single centreline point measurement. For the proportional balance method, no correction is necessary; however, an approximation of air flow may be obtained from:

$$q = v \times A \times 0.8$$

where

v = centre-line velocity reading (m/s)
A = duct area (m^2)

Measurements may be taken with a point reading probe which presents a minimum obstruction to air flow rate, e.g., pitot static tube.

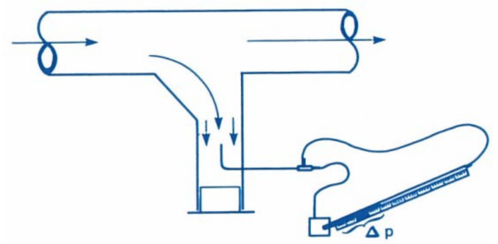


Figure 8.2 Point reading

8.5 FACE VELOCITY READINGS

Face velocity readings are used for proportional balancing of a series of ATD's of the same type, size and blade setting and involves recording the air velocity at the face of a supply or exhaust terminal.

The proportional rate may be calculated from

$$q = \bar{v} \times A$$

where A = effective core area (m^2)

$\bar{v}$ = mean velocity reading (m/s)

With a large rotating vane anemometer, it is sufficient to take one reading for each 150 x 150mm face area up to a maximum of 16 readings equally spaced. The anemometer head should be set to face the supply or exhaust flow as required and may be situated directly on the face or as much as 25mm from the face of the ATD.

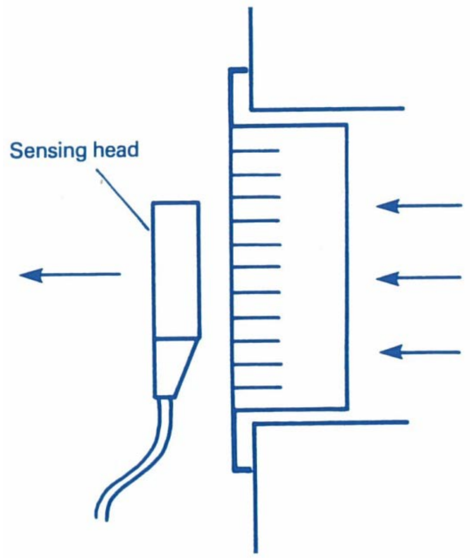


Figure 8.3 Face velocity measurement with a vane anemometer

This method is only valid if all measured velocities are in the same direction.

A throttled damper behind a supply grille causes a series of higher velocity jets which may result in erroneous readings and in this situation, unless correction factors are given, it may be necessary to use a balancing hood. Alternatively a short length, approximately two characteristic diameters, of stub duct can be constructed, using cardboard or similar material, outside of the grille to facilitate measurement.

8.5.1 Balancing Hood

This method is most practically applied to air flow measurement with most ATD's, particularly where duct entry and face velocity readings are unreliable.

Wherever a balancing hood is used, the measured air flow rate may be less than the actual air flow rate, due to the additional pressure loss imposed by the hood itself.

The hood may be of two types:

(i) Calibrated Type: usually a fairly sophisticated venturi shape having a known pressure loss co-efficient and provision for fitting an anemometer or grid and differential pressure measurement, or by-pass flow meter, device. The hood should be precalibrated such that its pressure loss characteristic is known and designed so that entry flow irregularities into the hood are reduced at the point of measurement.

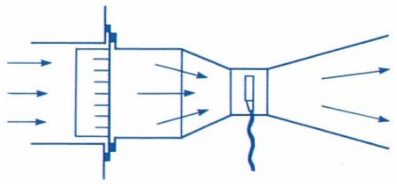


Figure 8.4 Calibrated device

(ii) Uncalibrated Type: can be made on site from available materials and is used to produce a reasonably uniform outlet velocity profile, with a minimum pressure loss. This hood may produce incorrect results if used with fixed vane diffusers which tend to divert the air to the sides of the hood.

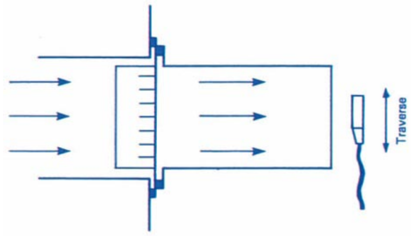


Figure 8.5 Uncalibrated application

Using an anemometer to measure hood outlet velocity by point or multipoint readings, the flow rate is obtained from:

$$q = \bar{v} \times AH$$

where $\bar{v}$ = average outlet velocity (m/s)

AH = outlet hood area (m^2)

8.5.2 Characteristic Terminal Area Factors

Although not recommended, this technique entails measurement of velocities at or within the ATD which combined with a quoted area factor allows an assessment of air flow rate to be made:

$$q = \bar{v}_K \times A_K$$

where $\bar{v}_K$ = average outlet velocity (m/s)

A_K = area factor (m^2)

The type of instrument, sensing head location and area factor must be supplied by the ATD manufacturer. Accuracy is generally of a low order and this method should only be used where no other technique is possible.

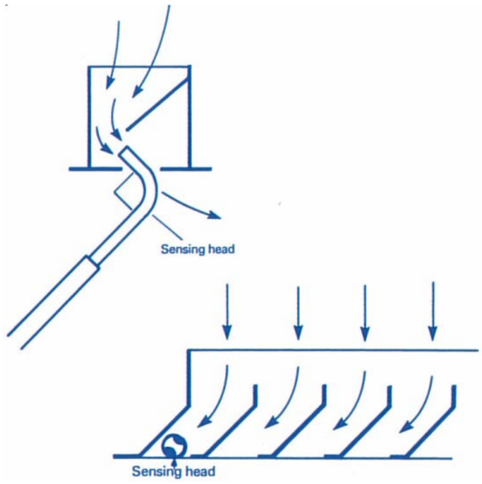
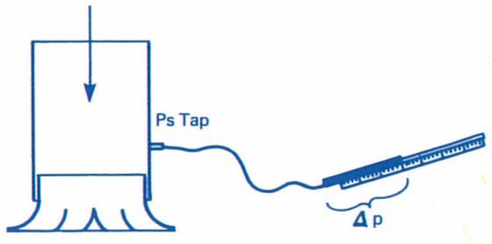


Figure 8.6 Measurement within ATD

8.5.3 Calibrated Terminal Pressure Loss Factor

With certain ATD's, particularly slot, square and circular diffusers, the terminal pressure loss may be used as a calibration of flow rate. With smooth (not turbulent) air flow profiles, a duct wall tapping, pitot/static tube or hypodermic probe reading in the neck of the ATD will provide a stable pressure loss reading of which the square root multiplied by the manufacturer's loss co-efficient "K" equals air flow rate. This technique relies on a reasonable pressure reading combined with stable and uniform duct entry conditions to ensure a usable accuracy.



$$q = K\sqrt{\Delta p}$$

Figure 8.7 Uncalibrated application

Where a large number of one type and size of terminals are served by a single duct system, the loss co-efficient system provides a fast method of proportional balancing.

It is essential that the terminal loss coefficient is constant, and this entails the setting of air pattern controls to the design requirements and the location of the volume control dampers before the pressure measurement station.

8.6 REGULATION AND BALANCING PROCEDURE

Regulation is defined as "the process of adjusting the rates of air flow in a duct system within specified tolerances".

Balancing the system consists of "setting the correct proportional air flow rate at each terminal or junction without regard for absolute air flow rate measurements".

A balanced system is then finally regulated at the main duct damper so that the design system air flow rate is achieved.

8.6.1 Preliminaries

(i) Ensure system cleanliness.

(ii) Check correct installation of duct turning vanes, dampers and air terminal devices.

(iii) Check that all dampers are set fully open.

(iv) Set all adjustable ATD's for the specified operating arrangement.

(v) Check that test access is provided in suitable locations as necessary.

(vi) Check for obvious air leakage, particularly at such locations as builders' ducts, access panels, etc.

(vii) Check that the duct system to be regulated has sufficient air flow for accurate measurement and is isolated from interaction with other systems, e.g., exhaust.

SYSTEM	TERMINAL BALANCE (proportion of percentage flow at index terminal)	BRANCH BALANCE (proportion of percentage flow at index sub-branch)	MAIN DUCT BALANCE Design flow
Systems where all terminals on any sub-branch serve one area	+ 20% - 0%	+ 10% - 0%	+ 10% - 0%
Systems where the terminals on any one sub-branch serve more than one area	+ 15% - 0%	+ 10% - 0%	+ 10% - 0%

Table 8.4 Typical balancing tolerances

8.6.2 Method

Use consistently, any of the measurement methods indicated in Sections 8.3 to 8.5 and adopt the following procedure:

(i) Within the branch duct system to be regulated, locate the terminal handling the lowest “percentage air flow rate” (i.e., the lowest ratio of the measured to the design air flow rate). This is the index terminal which will be used as a reference to balance all terminals on the branch duct (see Section 8.6.4.1.). (It is necessary for the last terminal to be, or to be equal to, the index terminal; if it is not, adjust it until it is handling the same percentage air flow rate as the index.)

(ii) Measure the indicated air flow rate at the next terminal and calculate the percentage air flow rate. If it is within the tolerances specified in Table 8.4 then no balancing is necessary. If not, adjust the damper and determine the new percentage flow rate.

(iii) Return to the index terminal and determine its new percentage air flow rate.

(iv) The two terminals will be in balance when the compared percentage air flow rates are within the tolerance indicated in Section 8.6.3.

(v) Repeat the procedure set out in (ii), (iii) and at the subsequent terminals.

(vi) Adjustments to terminal dampers remote from the index have little effect on the index flow rate. In practice, therefore, it is seldom necessary to return at each stage to re-measure the index terminal flow rate.

8.6.3 System Regulation and Balancing Tolerances

Table 8.4 provides a guideline for realistic typical accumulated tolerances which can be applied to low velocity supply and exhaust air systems.

8.6.4 Typical Examples

8.6.4.1 Calculation of Percentage Air Flow Rate

Index Terminal

– design flow rate = 720 l/s
– indicated flow rate = 680 l/s
– therefore percentage flow rate (680/720)
= 94%

Terminal being balanced

For a system where all terminals on a subbranch serve one area, balancing tolerance from Table 8.4 is:

Upper limit = + 20% (94% x 120% = 113%)
Lower limit = - 0% (94% x 100% = 94%)

Design flow rate of terminal to be balanced = 550 l/s

Indicated flow rate should be regulated to within:

Upper limit = 550 x 1.13 = 622 l/s
Lower limit = 550 x 0.94 = 517 l/s

8.6.4.2 Balancing Procedure

Complete preliminaries outlined in Section 8.6.1. Adopt procedure as shown in 8.6.2.

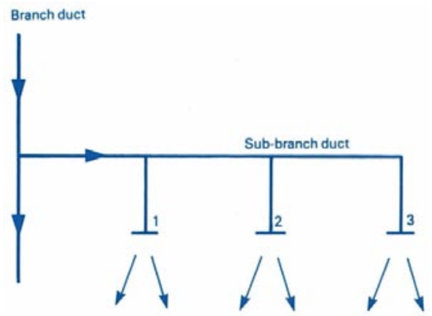


Figure 8.8 Typical branch/sub branch

8.7 DESIGNING FOR COMMISSIONING

8.7.1 Documentation

Provide the commissioning engineer with complete, clear installation drawings and if possible, schematic line drawings. Provide details of flow rates through terminal, each branch and the main duct; terminal pattern, etc., duct and terminal sizes; damper locations and measurement stations. All documentation should have clear identification coding for zones and terminal locations. This is particularly important in the case of large projects.

Results should be presented in a clear and concise manner. Standard forms such as those presented in the BSRIA "Manual for Regulating Air Conditioning Installations" (AG 3/89) are recommended.

8.7.2 Design Considerations

Good duct design and installation saves energy and minimises noise generation; remember design faults may be difficult or impossible to rectify at the commissioning stage.

Well designed stub ducts (see Section 5) will minimise the need for excessive terminal damper throttling; faulty design in this context can modify the ATD performance drastically with respect to noise generation, pressure loss and throw.

Rationalisation of the ATD range selected for installation throughout the project will provide the opportunity for the application of simple effective balancing techniques.

Provide adequate flow rate control dampers and test access or alternative flow measuring stations.

Ensure that good access to dampers, test holes, inspection panels and to other controls is included in the initial design.

TERMINAL	DESIGN DUTY (l/s)	MEASURED (INDICATED) DUTY (l/s)	% DESIGN	REGULATION TOLERANCE	UPPER LIMIT	LOWER LIMIT	COMMENT
3	200	210	105%				Index terminal
2	300	390	130%	+ 15% - 0%	121%	105%	Close damper 2 slightly Remeasure 3 & 2 flow rate
3	200	220	110%				Index terminal
2	300	345	115%	+15% - 0%	126%	110%	Within tolerance
1	200	260	130%	+ 15% - 0%	126%	110%	Close damper 1 slightly remeasure 3 & 1 flow rate
3	200	230	115%				Index terminal
1	200	240	120%	+ 15% - 0%	132%	115%	Within tolerance

Table 8.5 TYPICAL BRANCH/SUB BRANCH MEASUREMENT & BALANCING PROCEDURE Terminals being in balance, the sub-branch duct flow rate can be regulated with respect to the branch duct.

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