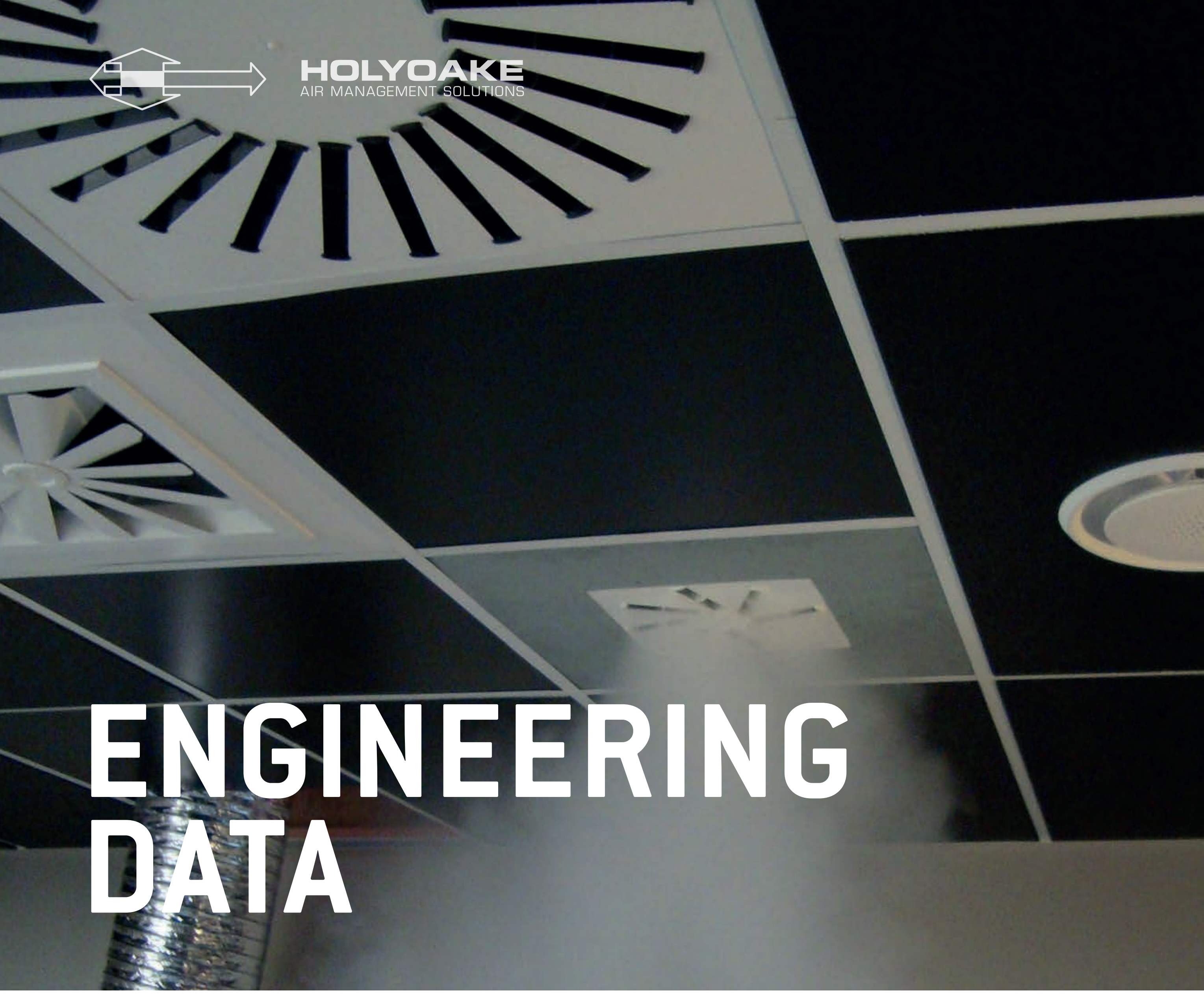


ENGINEERING DATA

A

ROOM AIR DISTRIBUTION

ENGINEERING DATA – Room Air Distribution

Room Air Distribution

Air Diffusion

Air distribution for heating, ventilating and air conditioning systems ranges from directional jets for spot heating, cooling, or air make-up, to totally diffused air streams of uniform velocity and direction for scientific and other specialised uses. Each type of application demands that its own needs be satisfied and there is no single method of distributing supply air which will meet the variety of requirements encountered in practice.

A design engineer must be familiar with the various means available to him and will make design decisions on major plant based upon the ability of the air stream to effectively offset the space load.

The most common requirement is to meet the needs of human comfort, and in this regard, the thermal balance which matches the load to heated or cooled supply air, must be achieved within parameters beyond the simple issue of temperature equilibrium and must include air motion, relative humidity and noise level.

The object of air diffusion in HVAC systems is to ensure an acceptable combination of these factors with the thermodynamic performance of the system. The key to the achievement of this is correct use of the principles of air diffusion.

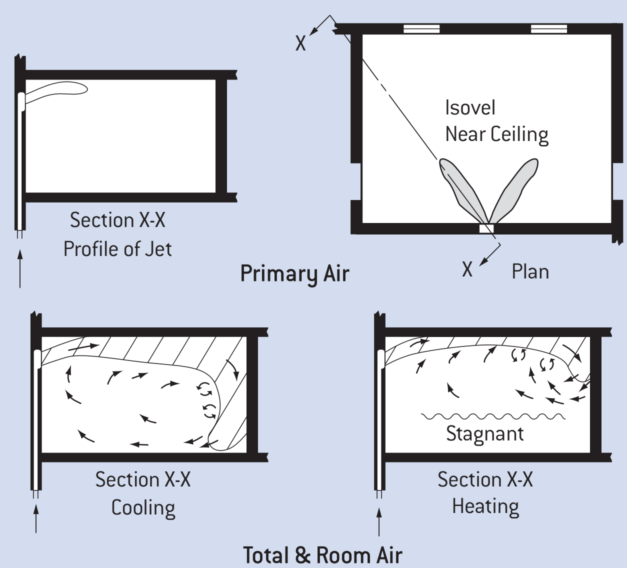
Air outlets have been classified into five groups:-

1. **Group A:** Outlets mounted in or near the ceiling and discharging the air horizontally. (Figures 1 & 2)
2. **Group B:** Outlets mounted in or near the floor and discharging the air vertically in a non-spreading jet. (Figure 3)
3. **Group C:** Outlets mounted in or near the floor and discharging the air in a vertical spreading jet. (Figure 4)
4. **Group D:** Outlets mounted in or near the floor and discharging the air horizontally. (Figure 5)
5. **Group E:** Outlets mounted in or near the ceiling and projecting the primary air vertically. (Figure 6)

While studying the following diagrams, which represent outlet performance in these categories, note that:-

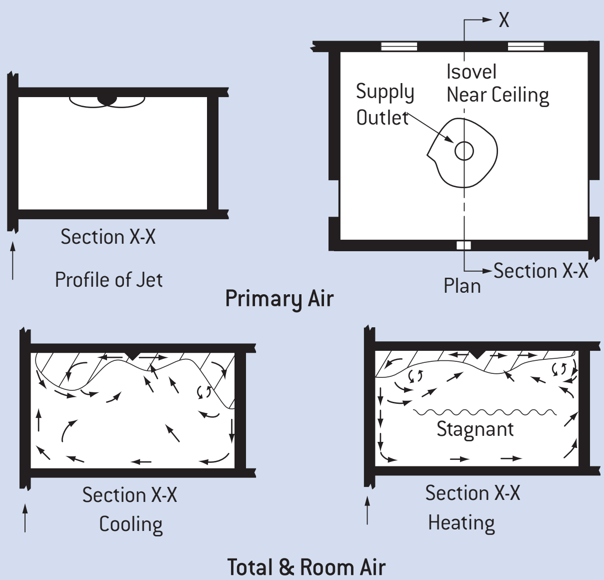
1. Primary air envelopes from outlets down to approximately 0.75 m/s are drawn as irregular lines enclosing plain colour. They are the same for heating or cooling.
2. Total air, shown by irregular lines enclosing colour with diagonal lines, is of relatively high velocity, but less than 0.75 m/s and generally within 0.5°C of room temperature. It is influenced by environment and tends to drop during cooling or rise during heating.
3. Natural convection currents form a stagnant zone from the ceiling down during cooling and from the floor up during heating. It should be noted that this zone is formed below the terminal point of the total air during heating and above the terminal point during cooling. Since this zone is a result of the natural convection currents, the air velocities within the zone are usually low (0.075 to 0.100 m/s), and the air stratifies in layers of increasing temperatures from a low to a high level. The concept of a stagnant zone is an important tool in the proper application and selection of outlets, since it permits due consideration of the natural convection currents from warm and cold surfaces and internal loads.
4. Location of return air collection points is of significance only within the immediate vicinity and should be chosen mainly from the viewpoint of localised drafts created by them. By selecting a location in the stagnant zone, summer and winter conditions are optimised. Induction and entrainment and therefore, load pick-up characteristics of the outlet, should be the main guide to acceptability of any given location when minimising short circuiting.
5. The general room air motion, shown by clear areas, is a gentle drift towards the total air. Room conditions are maintained by induction at the diffuser and entrainment of the room air into the total air stream.

Air Motion Characteristics of Group A Outlets



Sidewall Outlet

Figure 1. Horizontal projection from wall outlet. Total air can be made to follow wall down to floor if projection is sufficiently long. During heating this will result in a smaller stagnant zone.



Ceiling Outlet

Figure 2. Circular horizontal projection from ceiling outlet. Note influence of natural convection currents on total air pattern. Greater spread of this pattern results in more uniform space temperatures during cooling than those obtained from two-jet pattern of Sidewall outlet.

Air Motion Characteristics of Group B Outlets

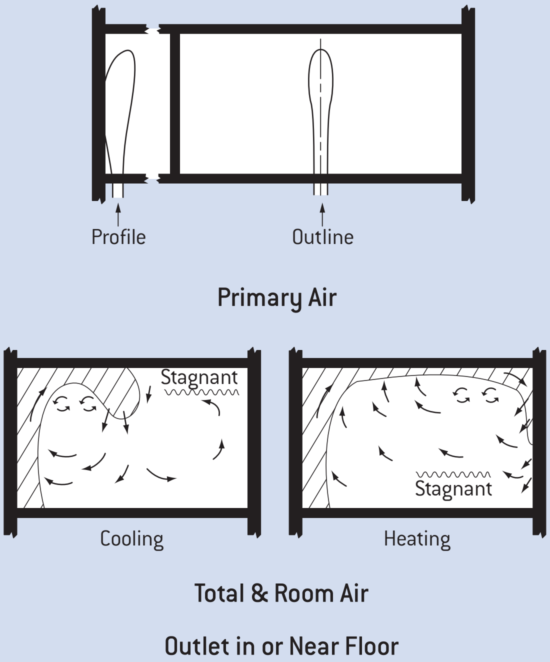


Figure 3. Non-Spreading Vertical Jet.

Air Motion Characteristics of Group D Outlets

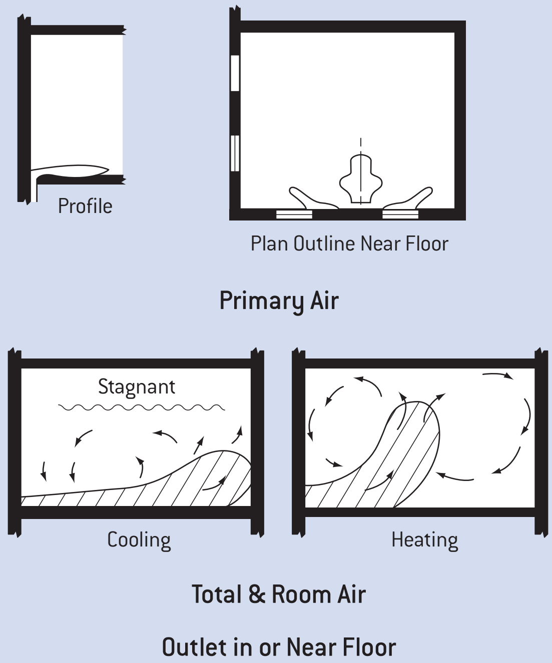


Figure 5. Horizontal Discharge.

Air Motion Characteristics of Group C Outlets

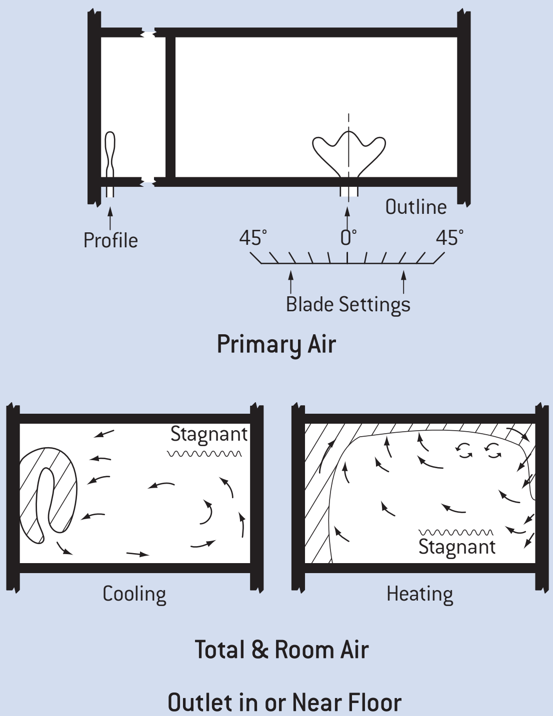


Figure 4. Non-Spreading Vertical Jet.

Air Motion Characteristics of Group E Outlets

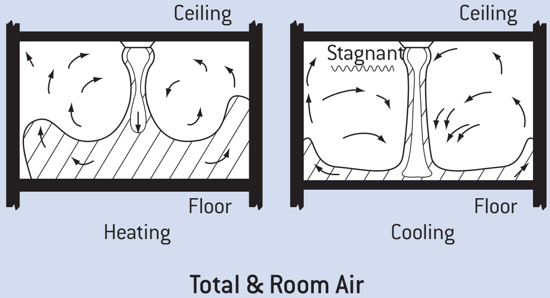


Figure 6. Vertical Projection from Ceiling Outlet.

Performance of Jets

All of the illustrated examples of room air distribution, figures 1 to 6 follow the rules that relate to the performance of jets, modified to the extent that their discharge shape and environment may influence flow.

Free isothermal jets perform to predictable effect. It is not the purpose of this publication to examine these effects in detail, but it is pertinent to identify the four zones of jet expansion[1] They are illustrated in the accompanying diagram (Figure 7).

Zone 1 extends only about four diameters or widths from the outlet face. Maximum velocity is practically constant.

Zone 2 over a further distance of about eight diameters or widths (depending on aspect ratio) the maximum velocity varies inversely as the square root of the distance from the outlet.

Zone 3 is the zone of fully established turbulent flow and may be a further 25 to 100 diameters long. Maximum velocity varies inversely with distance from the outlet.

Zone 4 is the terminal zone in which velocity diminishes rapidly to still air (below 0.25 m/s) in a few diameters.

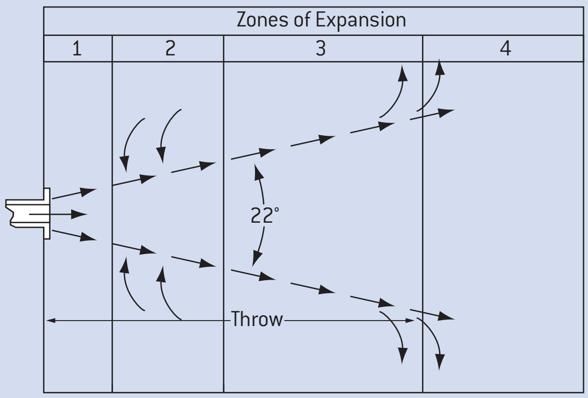


Figure 7. Isothermal Jet Near Ceiling.

Induction and Entrainment

The rate at which air from a jet can exchange heat with air from its environment is determined apart from Δt by induction and entrainment. For further discussion of this topic, refer to the section dealing with VAV air distribution.

For the vast majority of HVAC applications, the faster this heat exchange is performed, the better, and this usually calls for modification of a free jet.

Ceiling Surface (Coanda) Effect

If a jet is directed parallel and in close proximity to any surface, and in particular a ceiling, its own motion creates a low pressure area between it and the surface, due to its inability to induce or entrain air from the environment bounded by that surface (Figure 8).

This low pressure area causes a bodily shift of the jet so that it clings to that surface until its energy is either sufficiently dissipated, or some other influence such as high Δt or an obstruction separates the two.

The resulting reduction in induced or entrained air flow allows the energy of the jet to carry it for a greater distance and it is this feature which establishes the throw of ceiling, or near ceiling outlets designed for comfort cooling.

In the case of a sidewall outlet, this surface effect will be established if it is located such that its near boundary is within 300 mm of the surface in question - usually ceiling - and while it forms an angle of less than 40º with the surface.

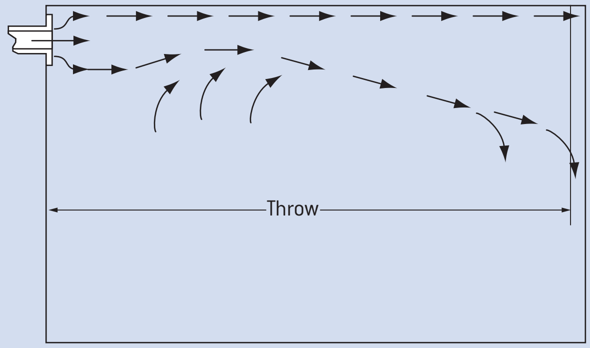


Figure 8. Isothermal Jet Near Ceiling.

Ceiling Diffuser Obstructions

As a rule of thumb, it can be taken that where the depth of an air stream from a ceiling diffuser exceeds 150 mm, its velocity and Δt are unlikely to create uncomfortable down drafts if it strikes an obstruction.

Surface mounting fluorescent lights are notorious for creating such down drafts and they should be avoided in well designed buildings. Where they can not be avoided, lowering them by 150 mm on rigid or chain supports can overcome their effect.

The throw of a diffuser, or grille, once the ceiling effect is established, can follow the line of the ceiling and down the wall, if over-blow is evident. Provided it is understood and designed for, it need not result in discomfort.

Non-isothermal Jets

With the exception of Series LD, CSRLA & EL (see Performance Notes) air diffuser ratings are tabulated as isothermal. The designer should allow for the effect of non-isothermal jets tending to rise during heating and drop during cooling. In both cases throw is reduced when compared with an isothermal supply, where velocities fall below 0.75 m/s. In the case of cooling, drop - or the collapse of ceiling effect will tend to occur sooner (Figure 9). For this reason diffuser design must ensure that at this point, mixing by induction and entrainment has minimised the room Δt.

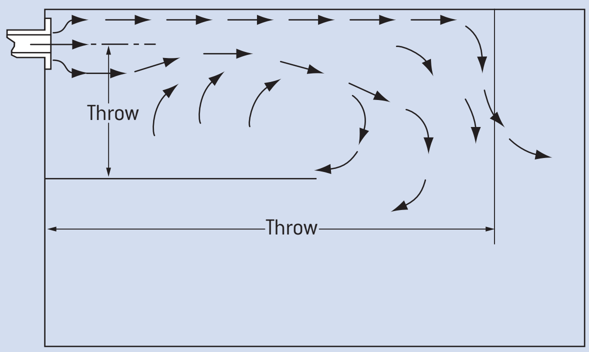


Figure 9. Cooling Jet Near Ceiling.

[1] See references on page 22A.

Effect of Vanes

More ceiling effect is achieved for sidewall grilles, by directing the air toward the ceiling. This causes the air stream to spread on the ceiling after impact and is effected by setting the horizontal vanes. A further improvement can be obtained by setting vertical vanes to give the air stream a spread to the 45° horizontal setting. Spreading the air stream by either or both means, reduces both throw and drop and for cooling or heating use, sidewall grilles are always best selected with double deflection vanes.

The greater the volume of air being projected by one outlet, the greater the drop. It is therefore wise to select a number of registers rather than a single grille handling a large flow, within the throw requirements of the space.

Establishing a good ceiling effect is critical to most comfort air conditioning applications. It is seldom good practice to direct the cooling air stream at a heat load in the space. Similarly, it is seldom good practice to direct a heating air stream at a heat loss area.

Room air motion, induction and entrainment should be used to offset the load without the need for confrontation of two air streams of widely different qualities. The best systems use these surfaces to best effect. Note in Figure 10, that where this is done, the high velocity sections of the air stream have less tendency to enter the occupied space.

It is not necessary to maintain a measurable level of air movement. Most of us still live in homes that are not air conditioned. Natural convection currents and other inherent air moving features of an untreated space are usually adequate to maintain comfort. In support of this, it has been shown that reducing the space temperature by 0.5°K is equivalent to increasing velocity by 0.08 m/s.

Jet Temperature, Velocity and Room ΔT

Comfort heating and cooling, unless there is a particular reason for turbulent air in occupied spaces, we invariably require that high velocities are kept in unoccupied zones. For practical purposes, in addition to ceiling areas, unoccupied zones are usually considered to include a space up to 300 mm away from walls.

Occupied zone velocities must be held below 0.25 m/s and of course there can be no temperature differential (ΔT) between the supply air stream and the surrounding air in this region. In other words, complete blending of supply and room air must be achieved in the unoccupied zones.

Choice of Terminal Velocity

The occupied zone air motion must naturally be less than the chosen terminal velocity (Vt), so it follows that first choice of throw should be to a Vt of 0.25 m/s, which is the longest of the three tabulated throw figures included in each of the performance tables of this manual. A knowledge of the type of occupancy, ceiling height, ceiling and wall obstructions, etc, is necessary for reasonable judgement of the true extent of acceptable throw, to a Vt of greater than 0.25 m/s.

Throw to a Vt of 0.5 m/s from a typical ceiling diffuser, if taken to the intersection of wall and ceiling, will usually result in the 0.25 m/s Vt being reached within the practical unoccupied zone, at a point down the wall before the floor is reached. However, sidewall registers typically have a much greater carrying distance between these two isovels and great care should be taken to check the 0.25 m/s Vt before any selection is finalised.

Mixing, and the resultant changes in room ΔT, are illustrated in figure 10, where it can be seen that a ΔT of 11°K where supply air enters the space at 5 m/s, reduces to 0.9°K by the time the air stream velocity reaches 0.5 m/s, and 0.4°K at 0.25 m/s.

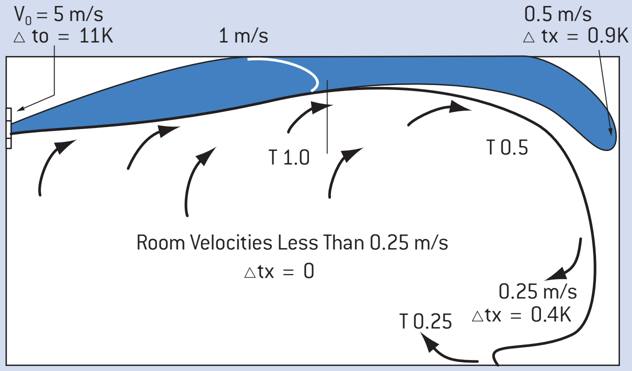


Figure 10. **Typical Sidewall Jet Showing Room Air Velocities and Temperatures.**

Use of the three terminal velocity throw figures in the performance tables will provide an image of the throw envelopes for the diffuser or register in question, so that the designer may then avoid discomfort in the occupied zone, while achieving his thermal balance.

Room ΔT Reduces with Distance

Where there may be doubt about temperature differences which may occur in the occupied zone due to such considerations as ceiling height limitations, drop, lack of ceiling effect, etc, then local temperature differentials may be approximately predicted by the use of the formula:-

(Equation 1)

$$\Delta tx\ ^{o}K = 0.8\ \Delta to\ Vx / Vo$$

Where Δ to = difference between room temperature and jet temperature at outlet (°K).

Δ tx = difference between room temperature and jet temperature at distance X from outlet (°K).

Vo = Centre line velocity of air stream at outlet (m/s).

Vx = Centre line velocity of air stream at distance X from outlet (m/s).

For a typical sidewall register, discharging supply air at 5 m/s and an initial ΔT of 11°K, the temperature difference derived by this formula, at various velocities would be as shown in the following table:-

Table 1

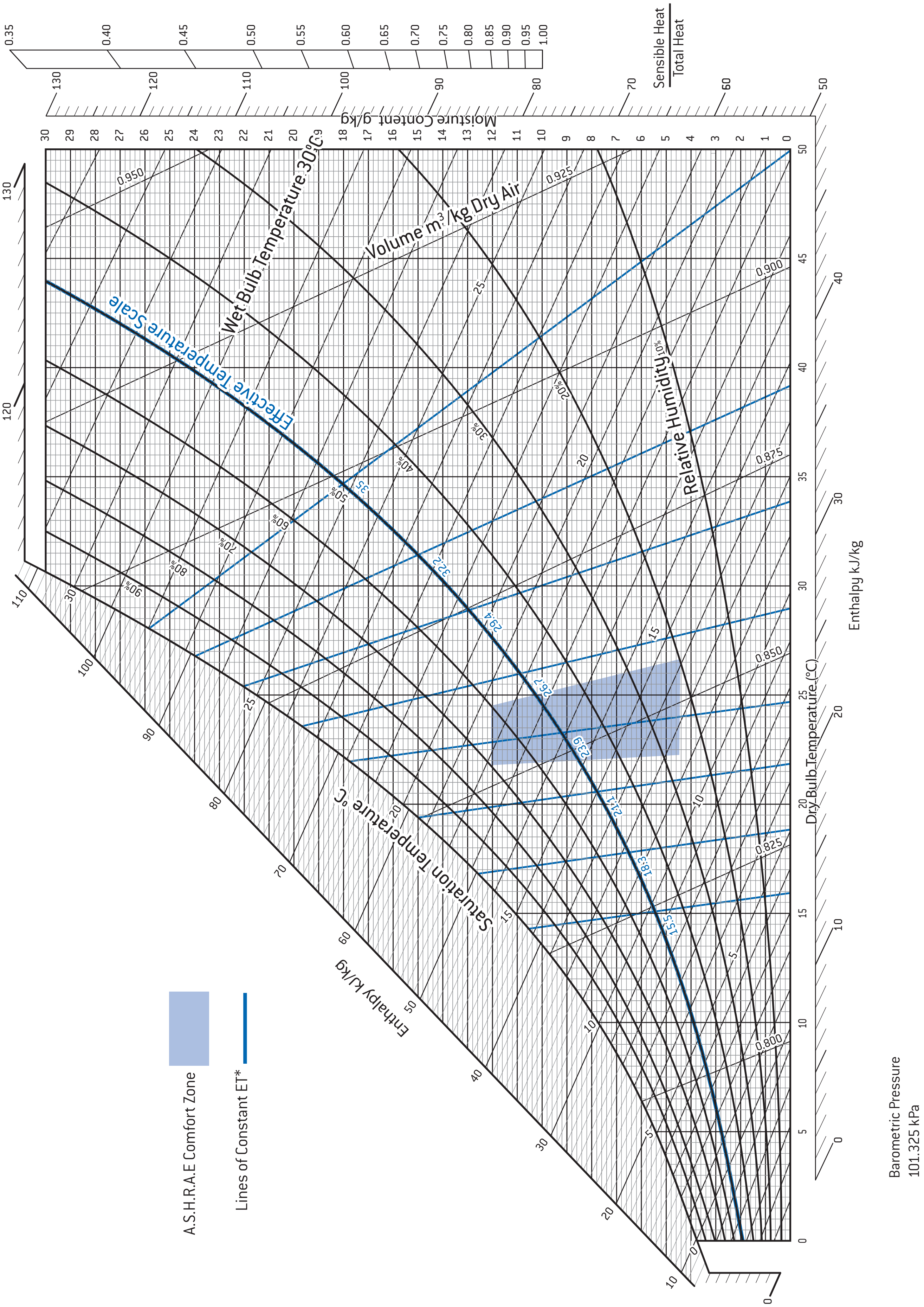
Vx (m/s)	3.00	2.50	2.00	1.50	1.00	0.75	0.50	0.25
Δ tx°K	5.28	4.40	3.52	2.64	1.76	1.32	0.88	0.44

Comfort Criteria

In 1974, A.S.H.R.A.E. developed its comfort standard 55-74, in which it defined a zone of comfort for the most common parameters (wide ranging environmental applications are more fully dealt with in standard 55-80). The parameters for standard 55-74 were:-

1. Altitudes from sea level to 2134 m.
2. Mean radiant temperature is nearly equal to dry bulb air temperature.
3. Air velocity is less than 0.2 m/s.
4. Occupants are in average clothing and at average levels of activity.

Psychrometric Chart with Effective Temperature and Comfort Zone



A new Effective Temperature (ET*) scale was developed in the 1960's and this superseded the old effective temperature scale that dates back to 1923.

Further studies were conducted in 1963 and onward, resulting in further definition of physiological stress related to environment, activity, period of exposure, sex and clothing. These studies relate comfort to lines of instant effective temperature and for young sedentary occupants, wearing light weight clothing, the following ET* v Stress relationships were observed:-

ET*	Physiological Condition
41.5	Intolerable
40	Very Uncomfortable
35	Uncomfortable
30	Slightly Uncomfortable
23.5	Neutral Comfort
below 20	Zone of Body Cooling

The Psychometric chart on the previous page shows in shaded colour, the comfort zone defined in 1974 by ASHRAE (average clothing & activity) and the new effective temperature scale, from which 23.5 ET*, the neutral comfort level for lightly clothed sedentary occupants, can be identified.

The two sets of criteria can be met by:-

1. Dry bulb temperature = mean radiant temperature
2. ET* = 23.5°C
3. Relative humidity range 22% to 65%
4. Air velocity BELOW 0.2 m/s

Selection and Location of Supply Air Devices

As shown in preceding pages, there are five groups of supply air outlets. This alone calls for five fundamental differences in design. Many variations within each of these groups are necessary to achieve particular performance features and beyond that, size, which often dictates construction methods, must ultimately be determined by flow, throw, acoustic and architectural considerations. For these reasons, it is not possible for any manufacturer to hold stocks that meet all possible needs. The development of manufacturing techniques, tooling and equipment in Holyoake plants has been and continues to be guided by the need for fast delivery of items built entirely from placement of order.

System design engineers therefore, have a virtually unlimited range to choose from and their choice must be made carefully and with the sympathetic understanding of architects and building designers. Errors or omissions are likely to be costly, if not impossible to correct once a building is occupied.

The three methods of selection that follow are complementary, and for best results, the other two methods should be kept in mind when concentrating on a selection by one of the three.

Methods of Selection

1. A.D.P.I. Method

Identification of the best type of diffuser, once a decision has been made on which of the five groups it belongs to (ceiling, wall etc.), can be further refined by reference to the ASHRAE Air Distribution Performance Index (A.D.P.I.) data. This subject is dealt with on page 17A and 18A (air diffuser types), in the section which discusses the design of air distribution systems for variable air volume. The A.D.P.I. chart is also reproduced on page 18A in order to apply a VAV desirability rating, based on throw tolerance for different diffuser types. This same data can be used to identify the range of throws that are acceptable for fixed air volume diffusers and Table 2 below summarizes the range of acceptable throw ratios for various types of outlet, to give optimum comfort conditions and an expected A.D.P.I. of 80, or better.

The A.D.P.I. is the percentage of (many) locations where measurements were taken which meet an effective draft temperature between -1.7 and +1.1° (and where air velocity is below 0.35 m/s). The nearer to an A.D.P.I. of 100, the better the air distribution and it is generally considered that an A.D.P.I. of 80 will meet the most critical appraisal.

Effective draft temperature can be calculated from:

(Equation 2)

$$\emptyset = (t_x - t_c) - 8(V_x - 0.15)$$

Where t_x = local temperature °C

t_c = room average temperature °C

V_x = local velocity m/s

2. Jet Performance Method

By using the three tabulated velocities from the particular performance data and with a knowledge of the way jets behave, (see Figures 1 - 6 on pages 2A and 3A and Performance of Jets, on page 4A), a mental picture can be formed by the envelope mapping for each isovel.

This system is usually quite adequate where there is a good understanding of the space shape and likely occupied zone, since the envelopes enclosed by the three isovels will provide information from which judgement can be used. For example a shopping mall may well be best served by "busy" air, with low ΔT, in which the 0.5 m/s isovel should be encouraged to enter the occupied zone, while a library could not tolerate even the intrusion of a light fixture in the 0.5 m/s air stream.

3. N.C. (Noise Criteria) Method

The subject of room noise criteria is covered more fully in following pages. However, a common procedure is to assume a 10 dB room absorption and a 1.5 m direct field from the single source and select on the basis of the tabulated outlet N.C. level as being equal to, or better than the desired space noise criterion.

The effect of number of outlets, distance, room absorption, etc, should be taken into account for critical areas, as described in the following material. Of course jet performance must be considered in making selections based on acoustic criteria.

Table 2

Description	High Sidewall Registers	Ceiling Diffusers					Sill Grilles
		Round	Perforated	Louvered	Slot	Light Troffer	
Series	DDL-20, 32	CRA, CSR	CPS, CPT	CMP, EL	CSD, CSDE	LTD	All Types
T.25 / L	1.3 - 2.0	0.5 - 1.3	1.0 - 2.7	1.0 - 2.7	1.5 - 3.3	to 4.5	0.8 - 1.5
T.5 / L					0.3 - 1.5		

T.25 and T.5 are throw distances to terminal velocities of 0.25 and 0.5 m/s respectively.

'L' is the characteristic length of the room, defined as the distance from the outlet to the nearest boundary wall in the principal horizontal direction of air flow, or half the distance between outlets plus downward travel of mixed air streams to reach the occupied zone.

ENGINEERING DATA – Outlet Location & Selection

Examples

For an office 4.5 x 3 x 2.7 m high, with a load of 125 W/m², 0.146 m³/s air flow and 11°K Δ T, with room absorption of 10 dB re 10^{-12} W, select a sidewall register located as shown in Figure 11, so that the space NC is less than 30.

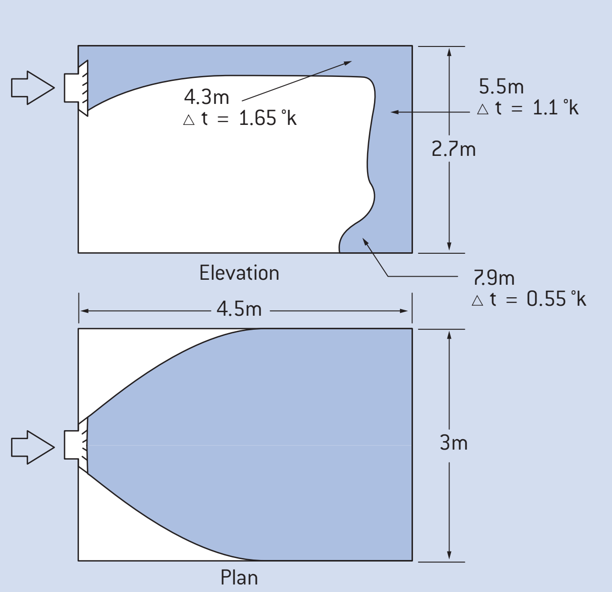


Figure 11.

1. Select on A.D.P.I. method

(a) Table 2 on page 7A indicates T.25/L should lie between 1.3 and 2.

(b) L = 4.5 m

(c) $V_{T\,0.25}$ = 4.5 x 1.3 min. = 5.85 or $V_{T\,0.25}$ = 4.5 x 2 max. = 9.00

From page 206E, a 350 x 125 DDL-20, with damper, will throw 9.8, 7.9 and 4.9m to $V_{T\,0.25}$ for straight, 22.5° and 45° vane settings. Use model DDL-20 size 350 x 125 with vanes set 22.5° divergent.

2. Jet Performance Method

Check jet performance of this selection from page 206E, VT.25 is 7.9 m, $V_{T\,0.5}$ is 5.5 m, and $V_{T\,0.75}$ is 4.3 m. Initial velocity Vo is 4 m/s.

From equation 1:-

at $V_{T\,0.25}$ tx = 0.8 x 11 x 0.25/4 = 0.55°K (at 7.9 m)

at $V_{T\,0.5}$ tx = 0.8 x 11 x 0.5/4 = 1.1°K (at 5.5 m)

at $V_{T\,0.75}$ tx = 0.8 x 11 x 0.75/4 = 1.65°K (at 4.3 m)

3. N.C. (Noise Criteria) Method

Is the noise level capable of meeting NC 30? Refer again to page 206E, where tabulated NC rating is 22. From page 202E note that NC values are based on a room absorption of 10 dB re 10^{-12} watts and that for 22.5° divergence listed noise levels should be increased by 1 NC.

Therefore, actual expected level of 23 NC is within the required noise criteria.

Note that in the above examples, several sizes of register would be acceptable in terms of these three selection criteria. In practice, the smaller the register, the higher its velocity and therefore its aspirating effect. Higher initial velocities would reduce the ΔT's listed above. For these reasons, as long as throw and noise levels are acceptable, a smaller register is likely to be a better selection than a larger one.

Allowing for "Drop"

The illustration in Figure 12 shows a free space cooling air pattern, i.e. there are no walls and may, or may not be a ceiling.

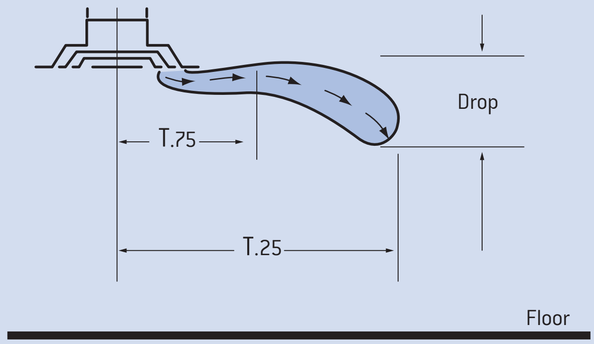


Figure 12.

The buoyancy effect of the lower temperature supply air causes it to drop as it approaches the end of its trajectory. The reverse is the case for warm air supply.

As discussed earlier, velocities in the occupied zone must normally be kept below 0.25 m/s and for comfort applications, the drop of the 0.25m/s isovel should be kept above the occupied zone.

Air mass and ΔT are the major influences on drop. The larger the quantity, the greater the drop. For this reason, larger numbers of smaller diffusers (see also page 5A "effect of vanes") are usually preferred.

Figures 13 and 14 on page 9A illustrate the extent of drop of a cool air supply, to a $V_{T\,0.25}$ limit, both with and without a ceiling. The diagrams are for a sidewall grille with zero deflection both horizontally and vertically.

No Ceiling Effect

Where there is no ceiling, or where the ceiling is more than 600 mm above the register, vertical deflection by 20° will reduce the drop by approximately 1m at the lowest point of the profile illustrated in Figure 14, on page 9A and at ⚠, with all other outlet velocity lines following a similarly raised locus while retaining a largely unchanged throw.

In the same case, i.e. where there is no ceiling, setting the vertical vanes for horizontal divergence of 45°, will reduce the throw in proportion to the tabulated values in the performance ratings, but reduce the drop by a full 1.5 m at the lowest point on the profile in Figure 14, and 1 m at ⚠, due to the wider distribution of the cold air stream, faster reduction of ΔT, and reduced buoyancy of the surrounding warm air.

Sidewall register performance ratings, (other than those for long throw e.g. series MDD and JD), are all based on the existence of a ceiling effect, so the tables may be used directly to establish the throw distance.

With Ceiling Effect

Where there is a ceiling, as is most commonly the case, vanes should be set to give sufficient upward deflection to establish the ceiling effect and the various outlet velocities will produce drop of the $V_{T\,0.25}$ isovel as shown in Figure 13.

Setting vertical vanes to give a 45º divergence, will reduce the throw as shown in the performance ratings and reduce the drop by approximately 1.8 m, at the lowest point on the profile illustrated in Figure 13 and 1 m at ⚠, again due to the reduced buoyancy of surrounding warm air.

Double deflection sidewall grilles are usually the best choice so that full advantage can be taken of these effects.

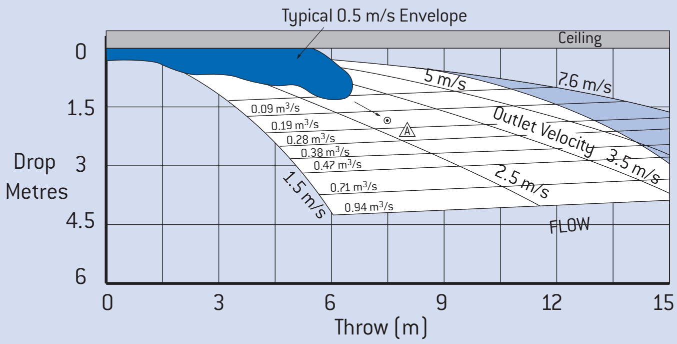


Figure 13. Mounted within 300mm of Ceiling. Vertical & Horizontal Deflection = 0°.

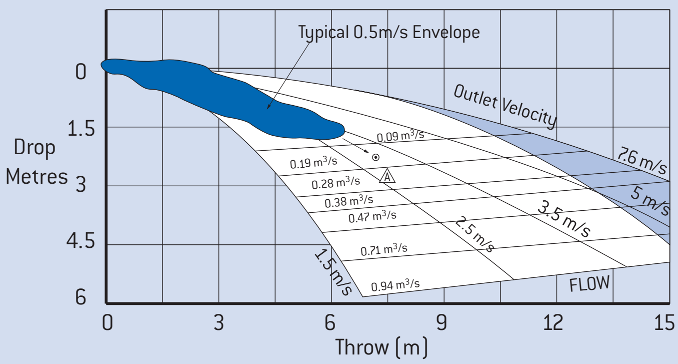


Figure 14. No Ceiling. Vertical & Horizontal Deflection = 0°.

Allowing for Drop

In Figures 13 and 14:

1. Throw and drop values are for a terminal velocity of 0.25 m/s.
2. Values are for free space (no walls) and 11ºK ΔT.
3. Data used is that for DDL or DDS -20, with OBD. Refer to page 202E for corrections for other models.
4. Small circle ⚠ in the white area of each diagram shows the comparative performance of one size grille handling 0.14 m^3/s at 3 m/s.
5. Refer to page 202E for deflection settings and resulting patterns.
6. The Light Blue shaded area to the upper right of each profile indicates noise levels above NC 30.

Drop with Linear Diffusers

As with the sidewall application, drop can be reduced with other types of diffuser by reducing the volume handled, increasing spread and establishing a ceiling effect. Linear diffusers or series CSD which have excellent ceiling effect characteristics, can achieve this by the inclusion of inactive sections alternating with active sections, by using more than one slot and directing the air two ways, or a combination of these. Minimum length of inactive sections to achieve best results should be as shown in Table 3.

Table 3			
Length of Active Sections	300	1500	3000
Length of Inactive Sections	300	600	900

Linear Diffuser Active v Inactive Sections to Minimise Drop

Design considerations should be based upon active diffuser lengths of 1.5 to 3 m and an inactive proportion of 30% of the total installed length.

Drop with Ceiling Diffusers

The selected air discharge pattern influences the drop with round and rectangular ceiling diffusers, as with sidewall registers, by virtue of volumes being handled from different segments, or sides. However, it is unlikely that drop will be of any concern in these cases, since the air quantity per diffuser is compatible with ceiling height.

Provided that air quantities are kept within the maximum values shown in Table 4, for the various ceiling heights, drop of the air stream will be maintained at a level that will satisfy the A.D.P.I. 80 criteria, or better.

Table 4						
Maximum Air Flow in m^3/s for Ceiling Height Listed						
Outlet Series	**Ceiling Height (mm)**					
	2400	2700	3000	3600	4200	4800
CRA	0.260	0.613	1.040	1.890	1.990	4.390
CSR	0.520	1.080	2.360			
CPS **CPT** **CMP***	0.095	0.165	0.260	0.425	0.660	0.755
EL* **LTD*** **CSD***	0.075	0.120	0.190	0.285	0.380	0.470

* m^3/s per side
Note that Series CSD should be installed with inactive sections to break up large flow levels. See also Table 3.

Notes

Throughout this manual references are made to the quantities Ac and Ak.
Ac = Diffuser or register core or neck area
Ak = Diffuser or register net jet area

Sound

Nature of Sound

Sound is the sensation perceived by the human ear resulting from rapid fluctuations in air pressure. These fluctuations are usually created by some vibrating object which sets up longitudinal wave motion in the air. In the case of air outlets it is caused by turbulence which creates rapid pressure pulses on the grille surfaces.

Most people have some intuitive idea of what constitutes a continuous wave, for example by observing the ripples created by a pebble striking the surface of water. Sound waves are a particular type of a general class of waves known as elastic waves, which can occur in media which possess the properties of mass and elasticity. If a particle in such a medium is displaced, then the elastic forces present will tend to pull the particle back to its original position.

The displaced particle possesses inertia and can therefore transfer momentum to a neighbouring particle, so that the initial disturbance can be propagated throughout the medium.

The major distinction between sound waves and the ripples on the surface of water, when using this as an analogy, is that water ripples are transversal, i.e. the particle velocity is at right angles to the direction of propagation, while sound in air is propagated by longitudinal waves, in which the particle velocity is in the direction of propagation.

A number of terms in common use describe the nature of propagation of a sound wave e.g. plane, diverging, spherical, progressive and standing waves.

Frequency of Sound

Repetitive, regular disturbances caused for example by a fan blade turning at constant speed, create recurring oscillations of the same cycle of particle movement and the number of cycles per second is referred to as frequency, 'f'. The unit of frequency is the Hertz (1 cycle/ second) and the time taken for the oscillation to repeat itself is the period, 'T'.

Wave Length

Wavelength, λ, is the distance between two successive pressure peaks in a plane wave. The speed of sound, 'C', is dependent on the mass and elasticity of the medium, which in turn is affected, in the case of air, by atmospheric pressure and density. At ordinary room temperature and pressure, the speed of sound is approximately 340 m/s.

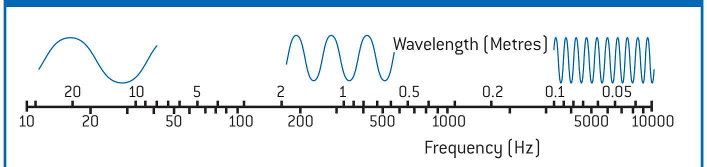
The relationship between wavelength λ, frequency f, and speed C, is:-

(Equation 3)

$$C = \lambda f$$

Table 5

Wavelength v Frequency



Amplitude

The maximum displacement experienced by a vibrating particle is known as the amplitude of vibration and for airborne sound this is the range from 10^{-7} mm, at which sound is just perceptible by the human ear, to a few mm, at which the ear would suffer damage.

Sound Pressure

The pressure variations produced when a sound wave propagates through the air are extremely small when compared with atmospheric pressure. The threshold of hearing for an average young adult corresponds to a sound pressure of 0.00002 Pa (1 Pa = 1 Nm^2). This sound pressure is superimposed on the ambient atmospheric pressure, which is in the order of 10^5 Pa.

The concept of sound pressure is extremely important, because it is the one universally chosen as the measurement basis for the evaluation of the strength of a sound wave.

Decibels and Reference Base

While sound pressure is correctly expressed in Pascals and sound power in Watts, both of these values become very cumbersome if used directly for practical measurement and equipment rating. In addition, the human ear does not evaluate sound by absolute values of pressure, or power fluctuation, but rather by subjectively judging the relative loudness of two sounds by the ratio of their intensities. This is a logarithmic behaviour and it is convenient to express sound levels, whether pressure or power, logarithmically. The most commonly used logarithmic scale for this purpose is the decibel scale.

One decibel is the energy or power ratio, r, defined by:

$\mathrm{Log}_{10}\ r = 0.1$

For sound pressure ratios, r, the definition is:

$\mathrm{Log}_{10}\ r = 0.05$

It is important to recognise that decibels give a relative measurement, each value in decibels being an expression of a ratio, relative to a reference pressure, or power, (or whatever other quantity is being considered).

By international agreement the reference base for each of the two values is:-

Wo, Sound Power: 10^{-12} Watts or 1 pW

Po, Sound Pressure: 2×10^{-5} Pa or 20µPa

Sound Power and Pressure Levels

Sound pressure level SPL (or Lp) of a sound of root mean square pressure Prms is defined by:-

(Equation 4)

$$SPL = 20 \log_{10} \frac{(Prms)}{Po}$$

Where Po is the sound pressure reference base.

SPL is the quantity actually measured when a microphone is placed in a sound field and Table 6 on page 11A gives some typical sound pressure levels.

Sound power level SWL (or Lw) is a measure of the energy output of a sound source and is defined by:-

(Equation 5)

$$SWL = 10 \log^{10} \frac{(W)}{Wo}$$

Sound (cont'd)

It is not sufficient to quote equipment noise levels by sound pressure level measured using the A-weighting network, since this is influenced by the environment. For this reason the power emitted by the device should be determined, as this is the fundamental indication of noise output and is virtually independent of the environment. Some typical sound power outputs of various sources are shown in Table 7 below.

Loudness and Weighting Networks

Due to the response mechanism and characteristics of the human ear, sound intensity expressed in W/m^2, (which incidentally is nearly identical numerically to Prms in μPa), loudness, as subjectively judged by the ear, cannot be measured as a simple function of sound pressure level. A sound pressure level of 20 dB would be clearly audible at 1000 Hz, but inaudible at 100 Hz.

The Phon is the unit of loudness level and the loudness level in phons of any sound, is defined as being numerically equal to the intensity level in decibels of a 1000 Hz tone, that is judged by the average observer to be equally loud.

Weighting networks were originally introduced to modify sound pressure levels in order to make them correspond as closely as possible to perceived loudness levels.

Four weighting networks, A, B, C & D were selected to cover the ranges in phons, respectively of less than 55; 55-85; greater than 85 and lastly to account for the increase in annoyance produced by certain aircraft noises.

Recent work has not supported these associations and frequency weightings, while now being a matter of convention, are of doubtful value in measuring, or rating noise levels of air distribution devices. Furthermore, the A scale is now frequently specified for rating sounds, irrespective of level.

Room Acoustics and N.C. Curves

The study of room acoustics is a science on its own and it is normally beyond the scope of an HVAC engineer's brief to be precise in his specification of acoustic performance, this generally being a matter for specialists in acoustics for special purpose rooms. A general expectation however, can be identified by the use of NC Curves. In most cases, an intuitive understanding of the particular room acoustics is all that is possible and of course the room plays a most important part in achieving desired results.

Table 6 – Some Typical Sound Pressure Levels.

Sound Pressure (Pa)	Sound Pressure Levels (dB re 2 x 10^{-5} Pa)	
	140	
100000000		
	130	Near a jet aircraft taking off
	120	
10000000		
	110	
	100	Near a pneumatic drill
1000000		
	90	
		Inside of motorcar
	80	
100000		
	70	
	60	General office
10000		
	50	
	40	
1000		Quiet livingroom
	30	
	20	
100		Quiet countryside
	10	
20	0	Threshold of hearing

Table 7 – Some Typical Sound Power Levels.

Power (Watts)	Power Level (dB re 10^{-12} W)
100,000,000	200
1,000,000	180
10,000	160
100	140
0	120
0,01	100
0,000,1	80
0,000,001	60
0,000,000,01	40
0,000,000,000,1	20
0,000,000,000,001	0

Source	Power
Saturn rocket	(50,000,000 W)
Jet airliner	(50,000 W)
Large orchestra	(10 W)
Chipping hammer	(1 W)
Shouted speech	(0,001 W)
Conversational speech	(20 x 10^{-6} W)
Whisper	(10^{-9} W)

Sound (cont'd)

Growth and Decay; Reverberation; Room Absorption

The extent to which an air outlet affects the room noise level is influenced by the sound absorbing characteristics of the room. Sound intensity, as measured at a particular point, will increase in a series of small increments, due to the arrival of reflection from walls, floors, ceilings, etc, until equilibrium is established and the energy absorbed by the room is equal to the energy radiated by the sound source. When the sound source is abruptly stopped, sound intensity will fade at a rate of decay determined by the amount and positioning of absorbing material in the room. This lingering of sound is known as reverberation and the rate of room absorption of sound energy is mainly proportional to the sound intensity (W/m^2).

An imperical relationship was first developed, between the volume of an auditorium, the amount of absorptive materials in it and reverberation time, at the beginning of the 20th century by W.C. Sabine, whose formula states that:-

(Equation 6)

$$RT = 0.161V / A$$

Where RT = reverberation time defined as the time taken for a sound to decay by 60 dB after the sound source is abruptly stopped.

V = volume of auditorium in m^3

A = total absorption of the auditorium in m^2-SABINS.

The absorption unit of 1 m^2-SABIN represents a surface capable of absorbing sound at the same rate as 1 m^2 of perfectly absorbing surface, e.g. an open window.
The absorption coefficient of a material is the ratio of sound absorbed by it, to that absorbed by the same area of an open window.
A perfect absorption coefficient, would be 1.

Sabine's formula gives a good indication of expected behaviour of fairly reverberant rooms with a uniform distribution of absorptive materials. Inaccuracies increase as the room becomes more dead and derivations of this formula have been developed for these and other cases.

Sound absorption is usually different for the same material at different frequencies, as illustrated by table 8. Room absorption can be calculated by measuring the reverberation time for each frequency of interest and predicted from shape, volume and the amount and types of absorptive materials used in its construction.

Every room imposes its own characteristics on those of any sound source present, so that the fluctuations in sound pressure which occur as a microphone is moved from one position to another, may completely obscure the true output characteristics of the source. Duct borne noise from fans, dampers, turning vanes, etc, may well be greater than the noise generated by air passing through a grille or diffuser and these may require special and separate treatment.

Table 8

Typical absorption coefficients of 1 m^2 of material

Material	Frequency, Hz					
	125	250	500	1000	2000	4000
Air, per cu. m.	nil	nil	nil	0.003	0.007	0.02
Acoustic Panelling	0.15	0.30	0.75	0.85	0.75	0.40
Plaster	0.03	0.03	0.02	0.03	0.04	0.05
Floor, Concrete	0.02	0.02	0.02	0.04	0.05	0.05
Floor, Wood	0.15	0.20	0.10	0.10	0.10	0.10
Floor, Carpeted	0.10	0.15	0.25	0.30	0.30	0.30
Brick Wall	0.05	0.04	0.02	0.04	0.05	0.05
Curtains	0.05	0.12	0.15	0.27	0.37	0.50
Total absorption of one seated person	**0.18**	**0.40**	**0.46**	**0.46**	**0.51**	**0.46**

Dampering

Where an accessory damper is attached to a grille or diffuser, its noise generation can not be separately attenuated and consideration should be given to its effect. Ratings listed will increase due to pressure drop increases across such an assembly, due to dampering, approximately as shown in table 9. Where possible, in noise critical applications, dampering should be done at the branch takeoff.

Table 9

Effect of Integral Dampering

Times Rated Total Pressure	1	2	3	4
dB Increase	0	7	12	18

Noise Criteria Curves and Ratings

NC curves, as developed by Beranek in the USA, are intended as a guide to average acceptability of noise as expressed by sound pressure levels and are the most widely used in Australasia. N.R. curves are similarly intended and are common in Europe. Although they are similar, one should not be confused with the other, refer to Figure 15 below.

Notes

Note that the NR-40 curve is shown for approximate comparison. (See also Section G for large scale NC chart and acoustic selection of Series HCV equipment.)

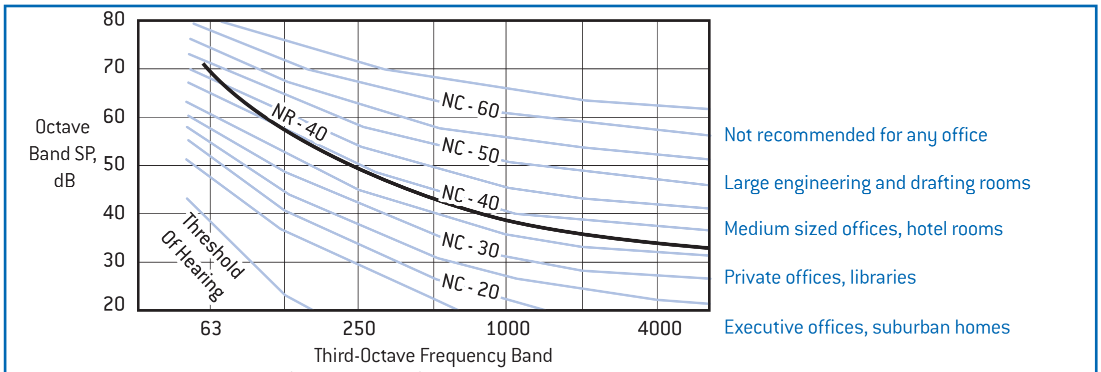


Figure 15. N.C. (Noise Criteria) Curves.

Sound (cont'd)

Care should also be taken to ensure even air distribution across a diffuser or grille. Misalignment of a flexible connection by 0.5 times the outlet diameter can raise the sound level by 12 to 15 dB, while misalignment by up to 0.125 D will have no appreciable effect.

Ranges of NC levels most commonly considered acceptable for different applications are shown in Table 10.

Table 10

Recommended NC values for various environments

Environment	Range of NC Levels likely to be acceptable
Factories (heavy engineering)	55 - 75
Factories (light engineering)	45 - 65
Kitchens	40 - 50
Swimming baths & sports areas	35 - 50
Department stores & shops	35 - 45
Restaurants, bars, cafeterias & canteens	35 - 45
Mechanised offices	40 - 50
General offices	35 - 45
Private offices, libraries,court rooms & school rooms	30 - 35
Homes, bedrooms	25 - 35
Hospital wards & operating theatres	25 - 35
Cinemas	30 - 35
Theatres, assembly halls & churches	25 - 30
Concert & opera halls	20 - 25
Broadcasting & recording studios	15 - 20

Noise ratings published in this manual are based upon a room absorption at all frequencies, of 10 dB re 10^{-12} W, in the direct field and about 1.5 m from a single source, i.e. the NC rating is established in SPL (or Lp) by subtracting 10 dB from the sound power level (SWL) of the outlet, to account for a likely and average room effect.

This is not to say that the measured sound pressure level or its calculated sound power output follow the NC curve. In fact, for a typical ceiling diffuser, it is likely that the sound pressure level, after room effect is deducted, would be 30 dB or so lower at 125 Hz and 10 dB lower at 4000 Hz, while meeting the NC curve only at 500 or 1000 Hz, or both. These are the frequencies that commonly determine the NC rating of a diffuser.

Corrections for Outlet NC Rating

A more thorough assessment of the effect that a room would have on noise criteria may be found in 2003 "HVAC Applications" Chapter 47. However the corrections shown in Table 11 below will provide an approximation, in which a room may be identified on the basis of volume and general type of space and corrections established for both direct and reverberant fields, RA and RA^1 respectively.

For the direct field, the volume and type of room and the distance from the outlet are needed. For the reverberant field, the number of outlets for the same room are required.

The specification will be satisfied when the outlet NC level does not exceed the specified NC, plus the smaller of the two corrections RA and RA^1.

Note that 'Q' is the directivity factor, for which a value of 2 is used where ceiling outlets are centered in a room and mounted flush and a value of 4 where side-wall, or baseboard outlets are located near the junction of two surfaces.

Refer also Section G for calculation example for VAV application of NC.

Table 11

Room Volume (m^3)			Corrections for Room Attenuation										
			Reverberant Field	Direct Field, One Outlet RA									
				Ceiling Height (m) Ceiling Outlets: Q = 2					Distance from Outlet to Listener (m) Sidewall or Baseboard: Q = 4				
Hard	Average	Soft	RA^1	2.5	3.0	4.5	6.0	9.0	1.5	3.0	4.5	6.0	7.5
400	75	-	-2	-4	-3	-2	-2	-2	-4	-2	-2	-2	-2
2000	280	45	2	-2	-1	2	2	2	-2	1	2	2	2
5000	1000	120	5	-1	1	4	5	5	-1	2	4	4	5
13000	2300	375	9	0	3	6	7	9	0	4	6	7	8
50000	8500	1275	12	0	4	8	9	11	0	5	8	9	10

Corrections for number of outlets in reverberant field

No. of Outlets in Space	2	3	4	5	6	8	10	More than 12
Correction, k	3	5	6	7	8	9	10	11

Noise in Direct Field: Outlet NC Level = NC + RA

Noise in Reverberant Field: Outlet NC Level = NC + (RA^1 - k)

ENGINEERING DATA – Overhead Heating

Overhead Heating

Engineers are generally aware of the problems that arise with heating systems dependant upon overhead air distribution. Buoyancy of warm air, already discussed under the heading of "allowing for drop", must be overcome by the strategic placement and selection of diffusers and such air distribution systems should be avoided in favour of other types of low level heating, e.g. under sill, or floor supply, skirting convectors, or panel heating.

The problem is exaggerated in colder climates where it may be necessary to employ double glazing to reduce heat loss and high velocity down drafts at windows.

Where it can not be avoided however, the compromise can be improved by using the following guide lines.

Diffuser Types and Location

Use diffusers that have a high induction ratio and preferably multi slot linear. Locate the diffuser at least a metre away from the exterior wall so that the air stream can spread across the ceiling and wall, entraining room air as it does so. Do not drive the air at the windows, or vertically project near them. See Figure 16 below.

If drapes or double glazing are used, the velocity of natural convection air may be low enough to be neutralised by room air circulation induced in the opposite direction, by setting linear diffusers to throw inwards, rather than outwards and the selection of CSD series with more than one slot allows the greatest flexibility for on site adjustment.

Room ΔT

The difference between room and supply air temperature should be limited to a maximum of 14°K and preferably less, otherwise buoyancy forces will overcome the momentum of the supply air and turn it back toward the ceiling.

Consider both Heating and Cooling

Select throw to reach the wall on heating, but prevent over-blow on cooling. A good selection will have the heating 0.75 m/s throw just reaching the wall, while the 0.25 m/s throw on cooling must not exceed the distance to the wall plus the height of the wall.

Care with Vertical Projection

Where vertical projection is being used to drive warm air downwards and this assumes a "heating only" duty, the notes on the performance data for each type of diffuser should be carefully considered.

Data for adjustable diffusers, normally used in heating systems, are provided with either specific data for heating differentials, or a statement as to the basis of the performance table.

Models CRA and Model CSRLA give vertical throw based on 11°K differential and $V_{T\,0.25}$ m/s.

Unless otherwise stated in the performance tables, all other vertical projection figures are for isothermal supply (zero ΔT) and allowances should be made accordingly.

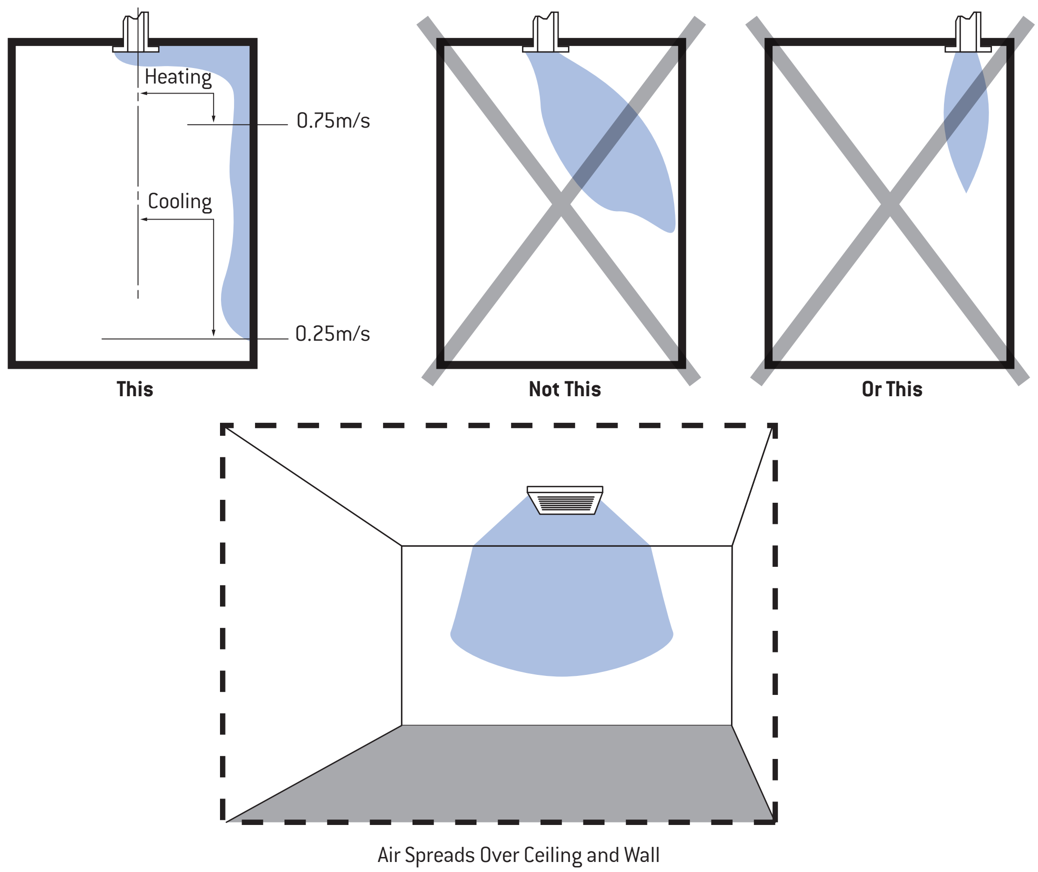


Figure 16. Outlet locations for overhead heating.

Pressure & Duct Design

Take-off to Outlet

Throw of registers, grilles and diffusers is assumed, for design purposes, to be uniformly centred about the outlet. The fact that it may not be so, may be of little or no importance, but without due attention being paid to the approach condition behind the outlet, the pattern is most likely to be asymmetrical, due to the approach angle of the supply air.

Where it is important that symmetrical diffusion patterns be established, then it is essential for branch take-offs to outlets to be fitted with curved splitters or guide vanes (see pages 374K - 375K for suitable types). If space or other considerations preclude this, then straight blade equalizing grids (refer to page 368K) may be used, but with somewhat less effect. A single bladed scoop type of deflector at the outlet is not desirable. Refer to figure 17 below.

Tabulated performance data gives the outlet total pressure (TPn) in pascals. This is the sum of the static and velocity pressures in a straight approach supply duct to the outlet and at a point 1.5 diameters upstream of it. Velocity pressure is the pressure corresponding to the duct velocity. The outlet static pressure then equals the total pressure minus the velocity pressure and it is actually measured at the above duct position, in accordance with industry standards.

The duct static pressure required upstream from a take-off, to obtain required flow with various accessories, may be established by using the take-off loss from figure 18. The duct static pressure will be outlet total pressure, plus take-off loss:-

(Equation 7)

SPu = Take-off loss + TPn

So called cushion heads, or chambers, have no beneficial effect.

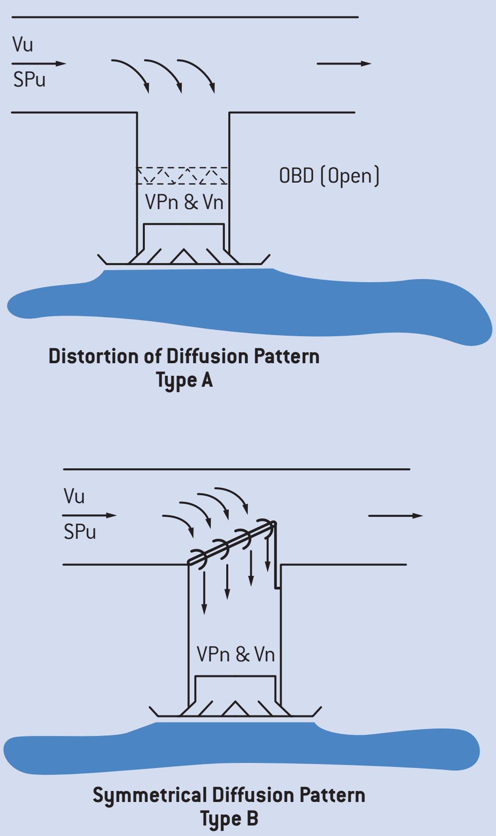


Figure 17. Effect of air approach.

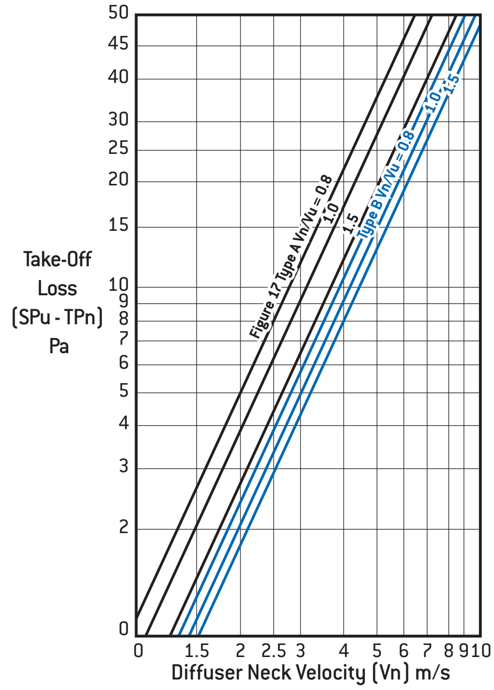


Figure 18. Take-Off Pressure Loss.

Velocity Distribution from Linear And Slot Diffusers

When a linear diffuser is fed from one end, air stream profiles are determined by the ratio of the area of the slot, to the area of the supply duct. The air discharge angle from a slot in a tapered duct will be uniform, while from a slot in a duct of constant area, the profile shape is determined by the percentage of slot area at any point along its length. See figure 19.

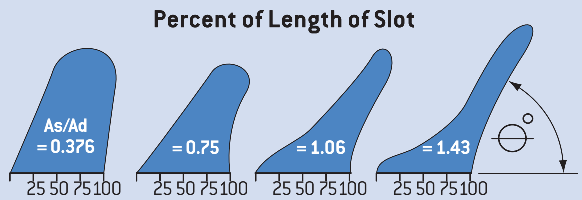


Figure 19. Shape of Airstream envelopes as slot area increases.

In both cases, the angle of discharge can be calculated by:-

(Equation 8)

$$\cot \theta = As \frac{Cd}{Ad}$$

Where θ = discharge angle in degrees
As = Area of slot m^2
Ad = Cross sectional area of duct at upstream end, m^2
Cd = Coefficient of discharge

Table 12 gives a guide to the air stream angles for different slot/duct area ratios.

Table 12				
Ducts for Linear Diffusers				
Ratio As/Ad	0.5	0.75	1.0	1.5
Angle of discharge θ	68°	59°	51°	33°

Linear outlets with inactive sections are desirable to minimise drops and they may be fed from one end. Duct design can take advantage of this by arranging for the take-off from the main duct to be attached to a diffuser plenum at an inactive section. The air will then be divided to equally, or proportionally serve the two active lengths.

Holyoake straightening vanes similar to Model TV on page 374K are available to eliminate this effect if required.

Expansion & Contraction of Aluminium Linear Grilles

Figure 20 below gives the amount of linear expansion and contraction caused by changes in the temperature of aluminium continuous diffusers, or grilles. In most cases the small amount of clearance that naturally occurs between the ends of diffuser lengths butted together, is sufficient to meet commonly encountered temperature changes. Installation is normally carried out in ambient conditions and it is only the rise from this temperature to the maximum operating temperature, that is significant.

Double duty (heating and cooling) grilles will contract on cooling, causing no problem and expand on heating. Unless specifically required otherwise, linear grilles and slot diffusers are furnished in maximum 2400 lengths, for convenient handling both in the factory and on-site. The gap per piece therefore, need only be 1 mm for a 17°K temperature rise. Alignment strips available with both types will slide easily to accommodate movement and make it easy to allow for this slight clearance.

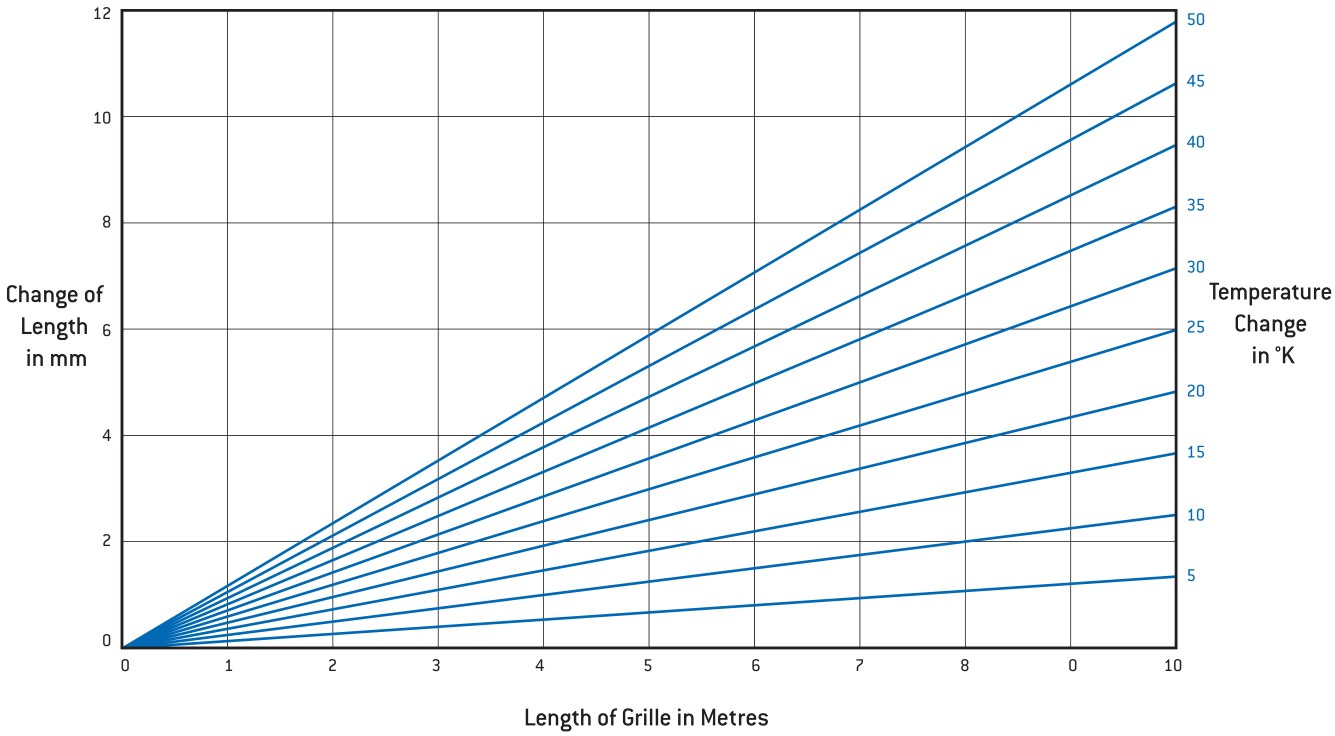


Figure 20. Expansion & Contraction of Aluminium Linear Grilles and Diffusers.

Variable Air Volume Systems

Introduction

Variable Air Volume has become a preferred system for most buildings where occupancy is of the type which demands unobtrusive, but closely controlled air conditioning. It has gained acceptance in office buildings of all types and has almost displaced some of the more complex and energy wasteful systems, such as constant volume dual duct and high pressure induction systems. It is important therefore, that the subject be given due attention in this handbook.

As with any system, the air distribution devices need to be selected with great care, for VAV applications more so. The range of compatible air diffusers is smaller and it is less likely that a field adjustment, to correct a draft complaint, will suit all conditions of air flow.

Two Basic System Concepts

The first decision the designer must make relates to which of the two broad categories of system type is to be used. With a reasonably flexible interpretation of the terms, all VAV air distribution systems can be categorised as either "fixed orifice-velocity re-set", or "variable orifice-pressure compensated".

By "fixed orifice-velocity re-set" we refer to systems in which the basic components are more or less conventional air diffusers, selected with an understanding of their intended flow variations and volume control assemblies fitted into the ductwork, one assembly per zone, as shown in the layout on page 260G for the offices designated conference, director, reception and general office.

The alternative approach, referred to as "variable orifice-pressure compensated", uses diffusers which are specifically designed for this type of system. They incorporate some type of actuator for opening and closing the gap or orifice, through which the air is released into the space. A regulating device in the duct feeding a number of diffusers, senses changes in demand and opens or closes, to suit these changes. Such a system could look like the one shown again on page 260G for the laboratory, senior analyst and accountant.

The basic device required to regulate air flow in the supply duct is the same in both cases, with different control systems. Both types of system are very flexible in terms of layout and tenancy changes and there is no reason why the system types should not be mixed, making a composite of the two typical layouts.

In either case, the above ceiling arrangement can be very straight forward and could easily consist of components available off the shelf, or at least of compatible construction, requiring little in the way of sheet metal trade skills, or fabrication labour, as shown in the illustration on page 260G. Note that for variable volume systems, it is preferable to use Spiroset (Semi Rigid) or Spiroloc (rigid) run-outs, rather than Spiroflex or similar "floppy" duct, to avoid any possibility of duct movement as pressures fluctuate. **Holyoake does not recommend the use of "floppy" flexible duct in any VAV system.**

Attenuation of noise created by dropping up to 500pa between the main supply system and the diffuser run-out and conditioned space, is achieved by either attenuators supplied as part of the regulating device package, or lined ductwork. It is usually very successful. In some types of equipment, the pressure reducing device and fixed orifice diffuser are contained in the one item, with acoustic treatment being concentrated in the diffuser and plenum assembly, but it is fair to say that these systems are more prone to noise level fluctuation, especially where duct pressures are capable of varying between 250 and 1250 Pa.

Air Diffuser Types

It has been known for years that complaints of drafts far outweigh complaints of lack of air movement. People may talk about stuffiness and there are certainly minimum air change rates and outside air ventilation ratios, that must be maintained to avoid this. Even too low a relative humidity, or too high a temperature can be responsible, but rarely insufficient breeze.

The A.D.P.I. or Air Diffusion Performance Index system of rating occupier comfort, as expounded in ASHRAE Fundamentals, identifies the percentage of a large number of locations where measurements are taken, which fall between +1.1° and -1.7°C effective draft temperature and below 0.35 m/s. At these conditions a high percentage of people are comfortable in office occupations. (Refer to Table 13 on page 18A)

Effective draft temperature is not directly related to dry bulb temperature, relative humidity, or mean radiant temperature, but is a function of the local temperature difference from the room average, as amended by local air velocity. (Refer to Equation 2 on page 7A).

It is important to recognise that a minimum velocity is not set and that high levels of satisfaction can only be achieved below 0.35 m/s. This emphasizes the need for draftless air distribution.

By examining the results of applying the A.D.P.I. method to the various types of diffuser, it can be seen that modern architectural trends are not the only reason for the virtual demise of sidewall registers, in favour of ceiling diffusers of various types in office applications.

Light troffers and linear ceiling diffusers are clearly more suitable to varying air flows (and therefore throws), than say sidewall registers.

0.005 m^3/s with a 11°K differential is a load of about 60 W/m^2, so in most office applications, the bottom two lines (60 & 130), would cover most centre zones and the 190 W/m^2 line would suit most perimeter areas.

Types have been rated in order of preference, based upon the breadth of the range of ratios $T_{0.25}$/L, consistent with high A.D.P.I. values, because the throw varies in a VAV system.

Note also that the 0.25 m/s isovel is given prominence, again emphasizing the lower velocities that maintain minimum draft levels.

All of the configurations listed need a ceiling for draft-less air distribution. In fact the most important component of an air distribution device for office type air conditioning is the ceiling itself.

ASHRAE gives formulae for calculating both entrainment and induction ratios, which illustrate the reason for superior performance of a slot over a square or circular jet.

Diffuser types can be categorised as an extension of one or the other. For example a light troffer and a linear diffuser are obviously slots. While these slots are actually in the plane of the ceiling, the design of their discharge opening, which involves a certain amount of intricate detail and streamlining is such that the air is immediately turned through 90° when it emerges, so that the effect is to drive a column of air along the ceiling surface.

A plaque type of diffuser with circular neck, such as the type used for variable orifice systems is, in effect, a slot wrapped around in a circle.

Variable Air Volume Systems (cont'd)

A circular ceiling diffuser, including those which have a square face and smooth transition from a round neck, is similar, with the exception that the air stream is divided by louver face pressings, giving the effect of a wider slot and higher induction ratio.

The traditional rectangular "multi-pattern" louver face diffuser is essentially a collection of square or circular jets and of course, side wall or sill grilles are more obviously square, or modified square jets.

There are two features of diffuser performance that should be highlighted. These are:-

1. Entrainment, which is also called "secondary air motion". It is the effect of the movement of a column of air as it draws room air along with it. The entrainment ratio is the total volumetric flow of air at a given distance from the discharge, divided by the volumetric flow of primary air at the same point. Because the entrainment effect is dominant in the turbulent zone, mixing is not predictable, or reliable.
2. Induction, which is also called "aspiration". This is the room air drawn into an outlet by the primary air stream. Mixing is complete and highly effective. The induction ratio is the volumetric flow of total air at the discharge, divided by the volumetric flow of primary air delivered to that point by the supply duct. The higher the induction ratio the shorter the distance before room ΔT is insignificant.

High induction ratios bring supply air temperatures up close to room temperature rapidly by mixing with room air. The throw of a diffuser as tabulated is mainly critical to the extent that it actually exists, where low turndowns are involved.

In other words, if the tabulated throw at a terminal velocity of 0.25 m/s does not extend further than the adjacent wall and say halfway down, then over-throw will not cause drafts at maximum flow.

At the other end of the scale, if the turned down air quantity is so low that no tabulated values can be found for it, then the diffuser is probably too large, or turn down too low.

However, as long as there is a ceiling effect and a high induction ratio, the load will be taken care of, either at minimum or maximum volume. Refer to page 260G for a selection of compatible diffusers for velocity reset control.

Variable Orifice Diffusers

Systems that use variable orifice diffusers with static pressure compensation, have the advantage of maintaining a constant centre line velocity at all flow levels and this feature maintains a more or less constant induction ratio, rather than a falling one, as volume flow is reduced.

The throw of these devices must reduce as mass is reduced, but to a lesser extent than with fixed orifice types, since constant pressure, which is the force per unit area behind the jet, means constant discharge velocity, so that $mV^2/2g$, the conservation of momentum statement, is not affected by the dominant variable. Refer to Section D for examples of variable orifice type diffusers. Model CSRVL, on page 178D combines the features of variable orifice and fixed orifice louver face types. These should also be used with upstream pressure regulators.

Minimum Turn-down

It is suggested that turn-down air quantities be held to a minimum of 0.003 $m^3/s/m^2$. A study of the incidence of complaints of "stale air" in Canadian buildings with VAV systems, refers to data collected by the National Research Council of Canada and offers a consensus of the opinions of engineering design consultants, that suggests a minimum air circulation rate of four air changes per hour, which assumes adequate dilution of the various contaminants present in a commercial space, e.g. vapours from synthetic materials used in wall linings, furniture, clothing, etc.

Table 13

Air diffusion performance index (based on A.S.H.R.A.E. fundamentals) and VAV diffuser rating

Terminal Device	Room Load W/m²	$T_{0.25}$/L for Max. ADPI	Maximum ADPI	For ADPI greater than	Range of T.25/L	VAV Rating*
High Sidewall Grilles	250	1.8	68	-	-	5
	190	1.8	72	70	1.5-2.2	
	130	1.6	78	70	1.2-2.3	
	60	1.5	85	80	1.0-1.9	
Circular Ceiling Diffusers/ Swirl Diffusers	250	0.8	76	70	0.7-1.3	3
	190	0.8	83	80	0.7-1.2	
	130	0.8	88	80	0.5-1.5	
	60	0.8	93	90	0.7-1.3	
Sill Grille Straight Vanes	250	1.7	61	60	1.5-1.7	5
	190	1.7	72	70	1.4-1.7	
	130	1.3	86	80	1.2-1.8	
	60	0.9	95	90	0.8-1.3	
Sill Grille Spread Vanes	250	0.7	94	90	0.8-1.5	5
	190	0.7	94	80	0.6-1.7	
	130	0.7	94	-	-	
	60	0.7	94	-	-	
Ceiling Slot Diffusers	250	0.3*	85	80	0.3-0.7	2
	190	0.3*	88	80	0.3-0.8	
	130	0.3*	91	80	0.3-1.1	
	60	0.3*	92	80	0.3-1.5	
Light Troffer Diffusers	190	2.5	86	80	3.8	1
	130	1.0	92	90	3	
	60	1.0	95	90	4.5	
Perforated & Louvered Diffusers	35 - 160	2.0	96	90	1.4-2.7	4
				80	1.0-3.4	

Tv = Throw distance at terminal velocity, V. *$T_{0.5}$ / L

L = Characteristic room length (diffuser to wall, etc).

This equates to 0.003 m^3/s per m^2 for a 2.6 m ceiling height and that in turn contrasts with the results of the N.R.C. survey, of a maximum of 0.0015 m^3/s per m^2, for any of the cases analysed.

Where a high proportion of natural fabrics and less finishes exuding heady mixtures of gaseous contaminants can be expected, this figure could be reduced, but there has to be a minimum for dilution somewhere. Ideally it should be identified with the intended contents of a building.

Variable Air Volume Systems (cont'd)

VAV Assembly and Control Devices

Volume control assemblies which provide either reset velocity control, or static pressure control, are to be found in Section G.

It is a very risky business to allow volume to be controlled directly by a thermostat. Upstream pressures must vary and differing flow levels can be expected for the same thermostat signal. Furthermore, the control feedback loop is too long and loose, if there is any such variation in the flow for a given setting, hunting is inevitable.

For these reasons units that are able to maintain downstream flow or static pressure, independently of upstream pressure fluctuations, are essential.The control devices for these units fall into two broad categories:

Pneumatic or electronic.

Few can maintain the often specified ±5% control tolerance, over zero (or very low minimum) volume flow, through to maximum and over inlet static pressures between minimum and 1000 Pa.

This tolerance requirement was introduced in the days of mechanical constant volume controllers, where the one flow, not reset by temperature, could with great care, be maintained over a wide inlet static pressure range. All air passed through the controller.

Specifiers should recognise the futility of such tolerances in the light of uncontrolled air inlet conditions on site and simply nominate the type of controls that are acceptable.

Briefly, Δ p means that the difference between total and static pressure, as a measure of velocity is translated into a control signal, which maintains a predetermined velocity by referring it to the desired value dialed into the controller.

The direct Δp needs amplification at the sensing probe to increase accuracy at low velocities.

Since $\Delta p = {}^{1}/_{2}\, \varrho\, v^2$ (where ϱ = density & v = velocity). It moves as the square of the velocity.

The controller is essentially a diaphragm and spring assembly, which uses a throttling range, or proportional band to position the damper between open and closed. A good quality commercial controller has a throttling range of 250 Pascals.

By calculating a series of velocities from high to low, a point will be found where a step down of 2.5 Pascals represents the required control tolerance as a percentage of velocity.

Each manufacturer makes his own decision on amplification of sensed Δ p, and in the case of the Holyoake series HCV, the critical velocity for ±5% is 2.5 m/s. However, control extends well below this and at ±7%, it is 2.0 m/s.

Electronic velocity reset control can be either analog, or digital, stand alone, or part of a building management system. For air velocity sensing, most depend upon the heat lost by a heated element, compared with an unheated element, to an air stream passing over both.

There are most often single point sensors, sampling air velocity at, or near the centre of the air stream.

More recent developments have introduced an approximation of average velocity sensing, which depends upon the continuous measurement of velocity pressure, or Δ p, at several points in the air stream.

Pneumatic controls readily accommodate this, but electronic systems require either a pressure transducer, or similar device, which will convert the air pressure differential to an electronic signal. It uses a similar principle to the single point sensor, in that the averaged velocity pressure produces a minute flow through a small device containing a chip, which measures heat lost to the air stream and relates this to mass flow.

Pick-ups used in conjunction with any such electronic Δ p measuring system vary widely in quality. Cheap plastic devices should be avoided, but with good quality, performance rated pick-ups, VAV device performance, with less than ideal inlet conditions, is greatly enhanced by the averaging Δ p method.

Electronic control manufactures rarely publish data on the accuracy, or overall tolerance of their systems. When such data is provided, it normally relates to individual elements only and the aggregation of these tolerances does not indicate highly accurate performance. Typically ±5% of full scale, which at low velocities may mean ±15%.

In practice, we have found that most commercial systems can be made to operate within ±5% at velocities above 3 m/s.

The other features available with electronic controls, such as immediate temperature and live flow readings from thermostat locations and the ability to incorporate them into a building supervisory system, are advantages which must be seen as being of far greater significance than absolute air velocity accuracy.

ENGINEERING DATA – S.I. Metric Conversions

Base & Simple Derived Units

length	m	metre
mass	kg	kilogram
time	s	second
electric current	A	ampere
therm. temperature	K	Kelvin
measured temperature	°C	Celsius
luminous intensity	cd	candela
luminous flux	lm	lumen (= cd sr)
illumination	lx	lux (= lm/m^2)
angle	rad	radian
solid angle	sr	steradian
force	N	Newton (= kgm/s^2)
pressure	Pa	Pascal (= N/m^2)
energy	J	Joule (= Nm)
power	W	Watt (= J/S)

Acceleration — m/s²

1 ft/sec²	= 0.3048 m/s²
32.2 ft/sec²	= 9.80665 m/s²

Air Flow Per Area — m/s m² = m³/s m²

1 cfm/sq ft	= 0.00508 m³/sm²

Angle — rad

1 degree	= 0.017453 3 rad
1 minute	= 0.290888 x 10^{-3} rad
1 second	= 4.84814 x 10^{-6} rad

Area — m²

1 sq in	= 645.16 mm²
	= 6.4516 x 10^{-4}m²
1 sq ft	= 92903.04 mm²
	0.09290304 m²
1 sq yd	= 0.836127 m²
1 sq mile	= 2.589999 km²
1 acre	= 4046.86 m²
	(0.404686 ha)
1 are	= 100 m²
1 centare	= 1 m²
1 hectare	= 10 000 m²

Bending Moment — Nm

1 lbf in	= 0.113 Nm
	= 113 Nmm
1 1bf ft	= 1.35582 Nm

Coefficient Of Heat Transfer (conductance) — W/m²K

1 BTU/hr ft² °F	= 5.67826 W/m²K

Conductivity (Thermal) (Conductance per unit length) — W/m K

1 BTU in/hr ft² °F	= 0.144228 W/m K
1 BTU/hr ft °F	= 1.73073 W/m K

Decimal Multipliers

Tera	T = 10^{12}
giga	G = 10^{9}
mega	M = 10^{6}
kilo	k = 10^{3}
hecto	h = 10^{2}
deca	da = 10
deci	d = 10^{-1}
centi	c = 10^{-2}
milli	m = 10^{-3}
micro	µ = 10^{-6}
nano	n = 10^{-9}
pico	p = 10^{-12}
femto	f = 10^{-15}
atto	a = 10^{-18}

Density — kg/m³

1 lb/in³	= 2.768 x 10^4 kg/m³
	= 27.68 kg/litre
1 lb/ft³	= 16.0185 kg /m³
density (dry air)	1.2 kg/m³ at 21°C
density (moist air)	1.184 kg/m³ at 21°C 50% RH
density (air)	$1.2 \times \dfrac{kPa}{101.325} \times \dfrac{294}{k}$
density (water)	1 kg/litre
	1 kg/dm³
	1000 kg/m³
1 tonne/m³	= 1 kg/litre = 1 g/millilitre

Energy — J=Nm=Ws

1 kWhr	= 3.6 MJ
1 BTU	= 1.05506 kJ
1 Therm	= 105.506 MJ
1 ft lbf	= 1.35582 J

Enthalpy — kJ/kg

(In Psychr. Charts Enthalpy = 0 at t = 0°C)

1 BTU/lb	= 2.326 kJ/kg

Force — N

1 lbf	= 4.44822 N
1 tonf	= 9.9640 kN
1 kip	= 4.44822 kN

Force Per Unit Length — N/m

1 lbf/ft	= 14.5939 N/m
1 kip/ft	= 14.5939 kN/m
1 tonf/ft	= 32.6903 kN/m

Heat Flow — W = (J/s)

1 BTU/hr	= 0.293071 W
1 Ton (Refr.)	= 3.51685 kW

Illumination — lx (= lm/m²)

1 foot-candle	= 10.7639 lx

Latent Heat Formula

2950 x m³/s x Δ g/kg	= Latent heat in Watts
2.95 x d m³/s x Δ g/kg	= Latent heat in Watts
2.95 x m³/s x Δ g/kg	= Latent heat in kW
2950	$= \dfrac{2500\ KJ/kg}{0.845\ m^3/kg}$

2500 = Heat required to evaporate or condense unit kJ/kg mass of water
0.845 = Specific volume m³/kg of moist air

Length — m

1 inch	= 25.4 mm
	= 0.0254 m
1 foot	= 304.8 mm
	= 0.3048 m
1 yard	= 0.9144 m
1 chain	= 20.1168 m
1 mile	= 1.609344 km

Lighting Per Unit Area — W/m^2
1 watt/sq ft = 10.7639 W/m^2

Liquid Flow — litre/s
1 imp gpm = 0.075 7682 litre/s
= 0.00007577 m^3/s
1 US gpm = 0.0631 litre/s
= 0.0000631 m^3/s
1 imp gph = 1.26280 x 10^{-3} litre/s

Load Distribution — N/m
1 lbf/ft = 14.5939 N/m

Loading Rate — W/m^2
1 BTU/hr ft^2 = 3.155 W/m^2

Mass — kg
1 oz = 28.3495 g
= 0.02835 kg
1 lb = 0.45359237 kg
1 ton = 1.01605 Mg or tonne
= 1016.05 kg
Mass of 1 litre of water = 1kg

Mass Flow Rate — kg/s
1 lb/hr = 0.00012599 kg/s
1 lb/m = 0.00756 kg/s
1 lb/s = 0.453592 kg/s

Mass Flow Rate Per Unit Area — kg/sm^2
1 lb/hr ft^2 = 0.0013562 kg/sm^2

Mass Per Unit Length — kg/m
1 lb/ft = 1.48816 kg/m

Modulus Of Elasticity — Pa
1 kip/in^2 = 6.89476 MPa
1 tonf/in^2 = 15.4443 MPa
29 000 000 psi = 0.2 x 10^6 MPa

Moisture Content — kg/kg
1 lb/lb = 1 kg/kg
1 grain/lb = 0.1428 g/kg

Moment Of Area — mm^4
1 in^4 = 416 231.4256 mm^4

Moment Of Inertia — kg m^2
1 lb ft^2 = 0.04214 kg m^2
= 42140 kg mm^2
1 lb in^2 = 0.000292 kg m^2
= 292.64 kg mm^2

Power — W = (J/s)
1 H P = 0.7457 k W

Pressure — Pa
1 psi = 6.89476 kPa
1 in Hg = 3.38639 kPa
1 in Water = 0.24908 kPa
1 ft Water = 2.98896 kPa
1 tonf/in^2 = 15.4443 MPa
14.696 psi = 101.325 kPa
1 atm = 101.325 kPa
Atmospheric pressure
14.696 psi = 101325 Pa
= 101.325 kPa
= 1013.25 mbar
= 1.01325 bar

Pump Power Input — kW
Power input in kW $= \dfrac{\text{litre/s x kPa x S.G}}{\text{1000 x Pump efficiency}}$

Resistance — m^2 K/W
1 sq ft °F hr/BTU = 0.1761 m^2 K/W
= 176.1 m^2 K/kW

Resistivity — m K/W
1 ft^2 hr °F/BTU in = 6.934 m K/W

Sensible Heat Formula
1213 x m^3/s x Δ K = Sens. heat in watts
1.213 x dm^3/s x Δ K = Sens. heat in watts
1.213 x m^3/s x Δ K = Sens. heat in kW
1.213 $= \dfrac{1.025\ \text{kJ/kgK}}{0.845\ m^3/\text{kg}}$
1.025 KJ/Kg K = specific heat of moist air
0.845 m^3/kg = specific volume of moist air

Specific Heat Capacity — kJ/kgK
1 BTU/lb °F = 4.1868 kJ/kgK
For Dry Air = 1.005 kJ/kgK
For Water Vapour = 1.89 kJ/kgK
For Moist Air = 1.025 kJ/kgK
For Water = 4.1868 kJ/kgK

Specific Volume — m^3/kg
1 cu ft/lb = 0.062428 m^3/kg
= 62.428 litre/kg
Specific volume for
dry air = 0.833 m^3/kg at 21°C
moist air = 0.845 m^3/kg at 21°C 50% RH
water = 0.001 m^3/kg
= 1 litre/kg

Stress — Pa
1 psi = 6.89476 kPa
= 0.00689476 MPa
1000 psi = 6.89476 MPa

Thermodynamic Temperature — K
1 R $= \dfrac{273.15}{491.67}\ K \approx .555\ K$
°C = K – 273.15

Temperature Difference — K
10 F = 5.55 K
9 F = 5 K

Temperature — °C
°F = 1.8°C + 32
°C $= \dfrac{(°F - 32)}{1.8}$

Torque — Nm
1 lbf ft = 1.35582 Nm
1 kip ft = 1.35582 kNm
1 ton ft = 3.03703 kNm
TORQUE FORMULA
Torque in Nm $= \dfrac{\text{kW (output) x 9560}}{\text{rpm}}$
9560 $= \dfrac{60 \times 1000}{2\pi}$

Total Heat Formula
1184 x m^3/s x Δ kJ/kg = total heat in Watts
1.184 x dm^3/s x Δ kJ/kg = total heat in Watts
1.184 = m^3/s x Δ kJ/kg = total heat in kW
1.184 = density of moist air, kg/m^3

ENGINEERING DATA – Sheet & Wire Gauges

Engineering Data

Universal Gas Constant J/kmol K

8315 J/kmol K

gas constant, air = 287 J/kg K

gas constant, water vapour = 462 J/kg K

Velocity m/s

1 fps	= 0.3048 m/s
1 fpm	= 0.00508 m/s
1 mph	= 0.44704 m/s
	= 1.609344 km/h

Velocity Pressure Pa

Velocity pressure in Pascals = 1/2 pv^2

p = density, kg/m^3

v = velocity m/s

For moist air (50% RH) at 21°C

250 Pa ≈ 20.55 m/s ≈ 4045 fpm

Volume m^3, litre

1 cu in	= 16387 mm^3
	= 1.6387 x 10^{-5} m^3
	= 16.3871 ml
1 cu ft	= 28.3 x 10^6 mm^3
	= 0.0283168 m^3
	= 28.3168 litres
1 cu yd	= 0.764555 m^3
1 imp. gal	= 0.00454609 m^3
	= 4.54609 litres
1 US gal	= 0.003785 m^3
	= 3.785 litres
1 pint	= 0.568261 litres
1 fl oz	= 0.028413 litres

Volumetric Flow

1 cfm	= 0.000471947 m^3/s
	= 0.471947 dm^3/s (litre/s)
1ft^3/s	= 0.0283168 m^3/s

VOLUME FLOW FOR B.S. 1042 : 1943

Q = 1.111 cZeED^2 $\sqrt{P/\varrho}$

where

Q	= m^3/s
1.111	= $^{\pi}/_{2}\sqrt{2}$
D	= internal diameter in m
P	= pressure in Pa
ϱ	= density in kg/m^3

Water Cooling

COOLING in kW	= 4.187 x litre/s x Δ K
COOLING in WATTS	= 4187 x litre/s Δ K
4.187	= specific heat capacity of water
kJ/kg K	
FOR 10F rise	
COOLING in kW	= 23.2 x litre/s

Performance Data Air Flow Rates

For the user's ease of calculation of total air quantities, velocities, duct areas, etc, performance ratings in this manual generally show volumetric flow in m^3/s.

To convert to l/s (litres or cubic decimetres) per second, shift the decimal point three places to the right.

In most all cases, values are shown to three decimal places, so that to convert the performance rating from **Cubic Metres Per Second, Ignore The Decimal Point And Read As Litres Per Second.**

Sheet Steel Weight

Metric Equivalent mm	Kg/m^2	Sheets per Tonne
2400 x 900 x 2.0	16.112	31
2400 x 1200 x 2.0		22
2400 x 900 x 1.6	12.889	37
2400 x 1200 x 1.6		27
2400 x 900 x 1.2	10.253	49
2400 x 1200 x 1.2		37
2400 x 1200 x 1.0	8.251	59
2400 x 1200 x 1.0		44
2400 x 900 x 0.8	6.640	74
2400 x 1200 x 0.8		55
2400 x 900 x 0.6	4.980	98
2400 x 1200 x 0.6		74
2324 x 1137 x 0.6		72
2400 x 900 x 0.5	3.906	118
2400 x 1200 x 0.5		88

Sheet & Wire Gauges

Gauge No.	Imperial Standard Wire Gauge (S.W.G.)	
	mm	inches
14	2.03	0.080
15	1.83	0.072
16	1.63	0.064
17	1.42	0.056
18	1.22	0.048
19	1.02	0.040
20	0.91	0.036
21	0.81	0.032
22	0.71	0.028
23	0.61	0.024
24	0.56	0.022
25	0.51	0.020
26	0.46	0,018
27	0.42	0.016
28	0.38	0.015
29	0.35	0.014
30	0.32	0.012

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Variable	Constants	Law	Equation
Rotational Speed	Fan Size Air Density Duct System	1. Flow is directly proportional to speed	$\frac{q_1}{q_2} = \frac{N_1}{N_2}$
		2. Pressure is directly proportional to speed²	$\frac{P_1}{P_2} = \left[\frac{N_1}{N_2}\right]^2$
		3. Air power is directly proportional to speed³	$\frac{A_1}{A_2} = \left[\frac{N_1}{N_2}\right]^3$
Fan Size and Rotational Speed	Tip Speed Air Density	4. Flow and air power are directly proportional to diameter²	$\frac{q_1}{q_2} = \frac{A_1}{A_2} = \left[\frac{D_1}{D_2}\right]^2$
		5. Speed is inversely proportional to diameter	$\frac{N_1}{N_2} = \frac{D_1}{D_2}$
		6. Pressure remains constant	$P_1 = P_2$
Fan Size	Rotational Speed Air Density	7. Flow is directly proportional to diameter³	$\frac{q_1}{q_2} = \left[\frac{D_1}{D_2}\right]^3$
		8. Pressure is directly proportional to diameter²	$\frac{P_1}{P_2} = \left[\frac{D_1}{D_2}\right]^2$
		9. Air power is directly proportional to diameter⁵	$\frac{A_1}{A_2} = \left[\frac{D_1}{D_2}\right]^5$
Rotational Speed and Air Density	Fan Size Pressure	10. Speed, flow and air power are inversely proportional to the square root of $\sqrt{\text{density}}$	$\frac{N_1}{N_2} = \frac{q_1}{q_2} = \frac{A_1}{A_2} = \left[\frac{\rho_2}{\rho_1}\right]^{\frac{1}{2}}$
Air Density	Rotational Speed Fan Size Duct System	11. Pressure and air power are directly proportional to density	$\frac{P_1}{P_2} = \frac{A_1}{A_2} = \frac{\rho_1}{\rho_2}$
		12. Flow remains constant	$q_1 = q_2$

Symbols:

A = Air power in watts

D = Impeller diameter in metres

N = Number of revolutions per second

P = Total pressure in pascals

q = Volume flow in m^3/s

ρ = Density in kg/m^3

Notes

1. Total Pressure = Static Pressure + Velocity Pressure
2. A = P x q
3. Shaft Power = A / Efficiency
   (Fan efficiencies usually range between 0.45 and 0.8)
4. System resistance usually varies as the square of velocity.
5. Fan laws apply accurately only to geometrically similar fans operating at the same point on the characteristic curve.

Geometric Formulae and Trigonometric Relationships

Volume:	Ellipsoid of revolution about "a" axis	$4/3\,\pi\, b^2 a$
	Ellipsoid of revolution about "b" axis	$4/3\,\pi\, a^2 b$
	Sphere	$4/3\,\pi\, r^3$
	Cylinder	$\pi\, r^2\, l$
	Cone	$\pi\, b^2 a / 12$
Circumference:	Circle	$2\,\pi\, r$
Area:	Circle	$\pi\, r^2$
	Ellipse	$\pi\, ab$
	Sphere	$4\,\pi\, r^2$
	Cylinder	$2\,\pi\, r\,[r+l]$
	Triangle	$1/2\, ab$

Trigonometric Ratios:		$\sin\theta = y/r$ $\cos\theta = x/r$ $\tan\theta = y/x$ $\sin^2\theta + \cos^2\theta = 1$