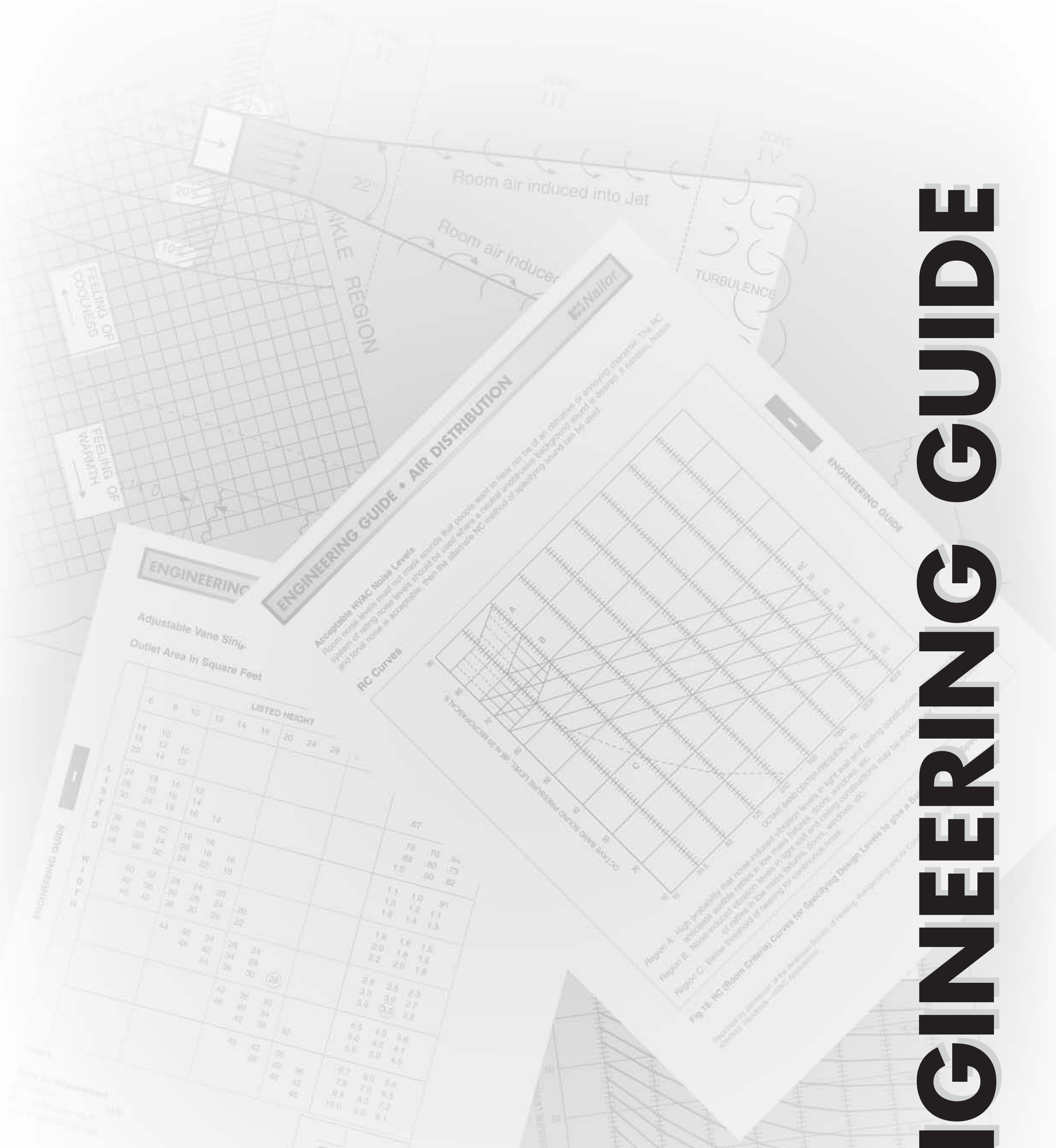


ENGINEERING GUIDE

Nailor Industries Inc.

Contents

Engineering Performance Data

The performance data shown in this catalog has been derived from tests conducted in accordance with ANSI/ASHRAE Standard 70 – 2006.

This standard incorporates ISO 5219 and ADC 1062 Test Standards by inclusion and also includes provisions for acoustical tests for ratings of diffusers using reverberant room test procedures, referencing ANSI Acoustical Test Procedures. Where possible, applicable ASHRAE standards are referenced for measurement specifications and procedures.

Comfortable in Today's Environment

The human body feels comfortable when four conditions in the immediate environment are correctly controlled. These are radiant temperature, air dry bulb temperature, relative humidity, and air motion.

Ideal comfort conditions are reached when the heat generated within the body is in balance with the heat lost to the surrounding air. Body heat is lost through radiation, evaporation of sweat by convection to air currents and conduction through surfaces with which it is in contact.

Not everyone is comfortable under identical conditions. Some prefer more air movement or higher or lower temperatures or higher or lower humidity. Generally speaking most people are comfortable when the relative humidity is between 30% and 65% and the dry bulb temperature is between 22° and 24° centigrade (72°–76°F). However, in the summertime when people wear lighter clothing, they can become accustomed to higher temperatures.

Radiant panels can affect body comfort. One degree F change in the temperature of a radiant panel will affect body comfort in the same way that 0.5°F change in air temperature will.

Humidity is very important to the feeling of comfort. Under normal conditions a relative humidity of 40% to 60% is ideal. As the ambient temperature rises, the body attempts to achieve a lower temperature by sweating. The sweat will evaporate and thus lower the skin surface temperature. The higher the relative humidity, the less sweat will evaporate. Therefore, as temperature rises, it is desirable to maintain lower humidities.

Air motion is also important to the feeling of comfort. The desirable velocity in normal situations for individuals who are moderately active and normally clothed is between 30 and 50 feet per minute. When velocities are too high, people complain of drafts. On the other hand, reduced air motion creates a feeling of oppressive discomfort because lower thermal conductivity rates between body and air result in higher skin temperature and less body heat loss. Even when the body is sweating profusely, low air motion inhibits evaporation. A rapidly moving air stream lowers skin temperature by convection and will evaporate sweat even under conditions of high humidity. The resulting lower skin temperature makes the body feel more comfortable. Drafts at ankle level can be 4°F cooler than at neck level and still be equally tolerable.

The basic criteria for room air movement may be obtained from the curves shown in Figure 1.

The Art of Comfort Ventilation

To maintain ideal comfort conditions in a space, ambient temperature, air movement, and humidity must meet design criteria. Heat losses in winter and heat gains in summer must be controlled. Sufficient "conditioned" air must be introduced unobtrusively into the space to mix with room air so that the resulting diluted conditions meet comfort requirements. The usual method is to supply air through circular or linear ceiling diffusers. Wall or floor grilles are often used in residences. Heating or cooling devices located in the space, supply air through specially designed outlet grilles. The art is to supply the air so that there will be no objectionable drafts in the occupied zone. High velocity streams of conditioned air should be supplied outside this zone, in the space one foot from the walls and above six feet from the floor.

The desired room temperature may be maintained by varying the volume of air supplied in response to a thermostat. This will be chilled air to cool or hot air for heating.

Alternatively the volume of air supplied can remain constant and only its temperature varied in response to load conditions.

Effect of Drafts on the Feeling of Comfort

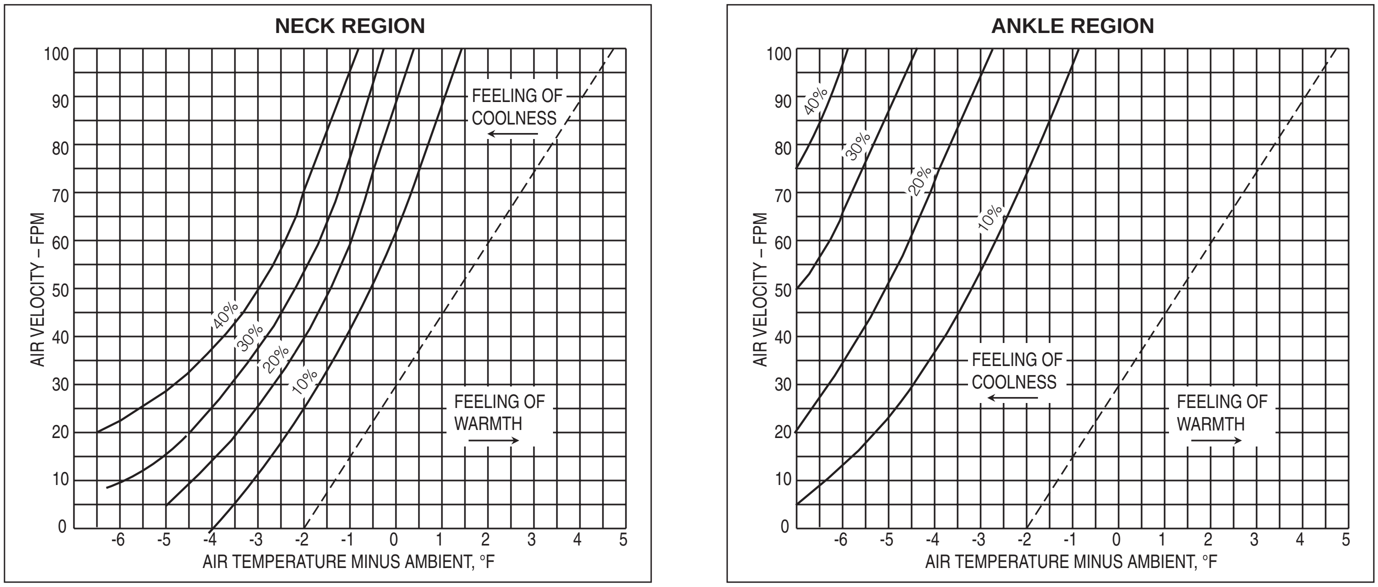


Fig. 1. Percentage of Occupants Objecting to Drafts in Air Conditioned Rooms.

Isothermal Free Air Jets

In order to perform this task of draft free ventilation we must first learn how high velocity air jets behave. An isothermal jet is a jet of air at the same temperature as the ambient air into which it is being introduced.

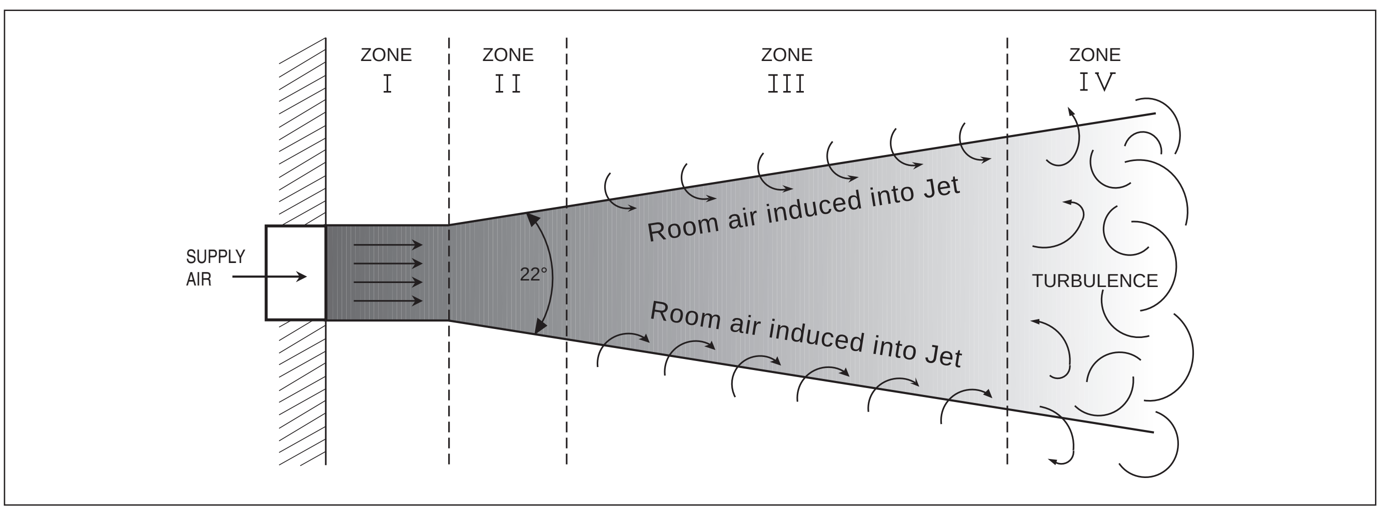


Fig. 2. Zones of Expansion of an Isothermal Jet.

For isothermal air jets, the relationship between average outlet face velocity and distance from the outlet has been shown to change as the jet progresses through four characteristic zones.

Zone I: A short zone, extending to about 4 diameters from the outlet, in which the centerline velocity remains practically unchanged.

$$V_x / V_o = \text{constant} \quad (1)$$

Zone II: A transition zone, extending to about eight diameters for round outlets, or for slots of large aspect ratio, to a distance of approximately the width multiplied by four times the aspect ratio. In this zone, velocities vary inversely to the square root of the distance from the outlet.

$$Vx \propto = \frac{1}{\sqrt{X}} \qquad (2)$$

Zone III: A long zone of turbulent flow where the jet expands as it draws room air into it. It may be 25 to 100 diameters long. In this zone, velocity varies inversely with the distance from the outlet.

$$Vx = \frac{K\ Vo\ \sqrt{Ao}}{X} \qquad (3)$$

or

$$Vx = \frac{K\ Qo}{X\sqrt{Ao}} \qquad (4)$$

Where: K = Proportionality constant.
Vx = Centerline velocity at distance X from outlet.
Vo = Outlet velocity.
Ao = Area of primary air stream (outlet) at the Vena Contracta.
X = Distance from outlet.
Q = Supply airflow rate (cfm).

Zone IV: A terminal zone in which the maximum velocity decreases rapidly to a velocity below 50 fpm and becomes a rolling mass of air at a temperature differential of about 1° to 2°F above or below room temperature, depending on whether the air jet is heating or cooling.

Air velocity at distance X from the outlet is approximately:

$$Vx = \frac{0.8\ \ K\ Qo}{X\ \sqrt{Ao}} \qquad (5)$$

For Non-isothermal Jets at Terminal Velocities Below 150 fpm

K =	Free openings, round or square	5.8
	Rectangular slots aspect ratio (< 40)	5.0
	Radial slots (use X / H instead of $X\sqrt{Ao}$)	3.9
	Grilles & Grids (> 40% Free Area)	4.8
	Perforated panels	4.0

Performance of Non-isothermal Jets

Formulas for isothermal jet behavior have been determined for jets projected into a space where the space air temperature is the same temperature as that of the jet. In normal heating, ventilating and air conditioning installations the temperature conditions are non-isothermal; that is, the temperature of the supply air jet is higher or lower than the temperature of the air in the space into which it is projected. The K factors in non-isothermal formulas have been adjusted to suit actual conditions.

Coanda Effect

The techniques of modern comfortable draft free air conditioning would not be possible were it not for the "Coanda Effect". This refers to the behavior of a jet of air on a flat surface. If the jet hits the surface at an angle less than 40 degrees, it will hug the surface, whether it be a soffit, a wall, or ceiling and progress along it. This is why air flowing from a side wall supply outlet 12" below the ceiling, will jump up to the ceiling and cling to it as it progresses across the room. Supply air from ceiling diffusers works in the same way.

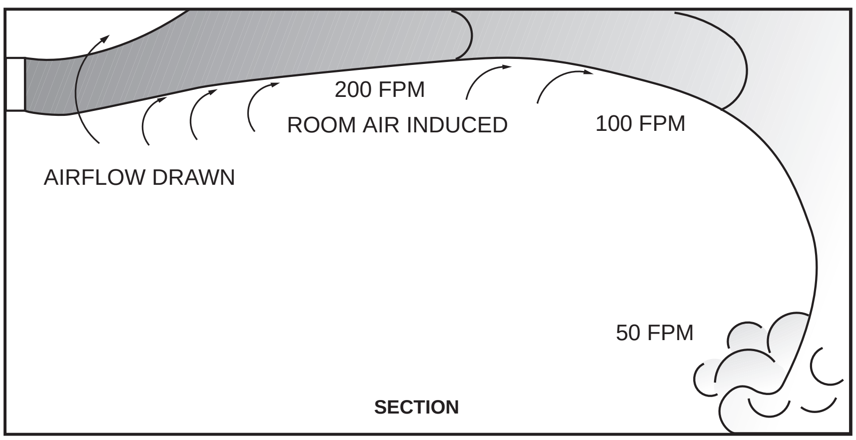
The phenomenon is due to the creation of a low pressure area between the jet and the surface and the reduced induction on the surface side of the jet. The "Coanda Effect" increases the throw for all types of outlets and reduces the drop for air projected horizontally as, for example, from a ceiling diffuser.

Jet Interference

A jet of air projected across a ceiling needs a clear smooth path to distribute its temporizing effect in an efficient manner. Beams, light fixtures, or architectural features, which interfere with the flow of the jet stream, will deflect it into the occupied zone. This not only causes uncomfortable drafts, but it also prevents the air conditioning system from doing a satisfactory job.

Air Through Grilles

When a jet of ventilation air is projected into a space, it begins to expand. The edges of the jet induce room air into the jet stream. This dilutes the primary jet reducing its temperature and slowing its forward motion. When the forward motion slows to 75 to 50 fpm, called the terminal velocity, there is no longer enough energy left to push room air out of the way and the jet breaks down into a rolling mass of turbulent air.



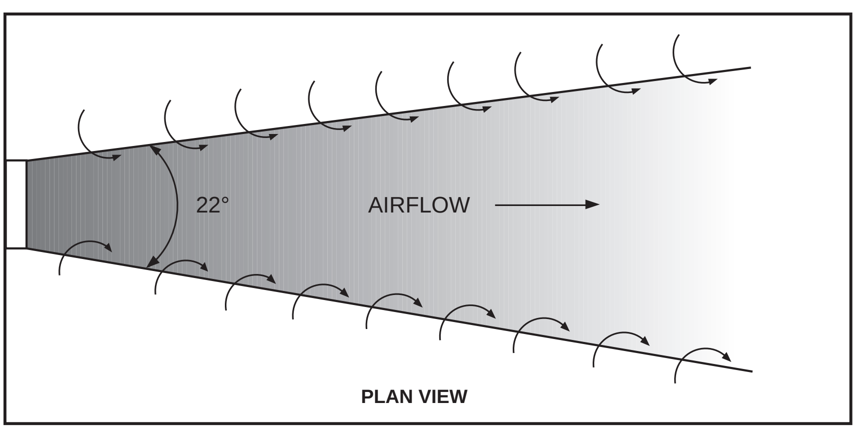


Fig. 3. A Jet of Ventilating Air from a Wall Outlet a Few Inches Below the Ceiling.

High Ceilings

Catalog data for diffuser throw is based on a 9 foot ceiling height. Diffuser supply conditions would be chosen so that throw terminal velocity of 100 fpm would occur midway between adjacent diffusers or at a wall. This will result in a room residual velocity of about 35 fpm. Where ceiling heights are above 10 feet, higher terminal velocities are required. Increase terminal velocities one foot for each foot the ceiling height is above 10 feet.

Return Air Inlets

Air return inlets affect air patterns only in their immediate vicinity. For heating, the return should be at the floor to drain off the cold air. For cooling locate the return in the stagnant zone.

Air Volume Requirements

The primary air jets supplying air to a space must contain enough energy to keep the room air mass in the occupied zone moving at a velocity of 20 to 50 fpm. Excessive velocities create uncomfortable drafts and should be avoided. Cold air is heavier than warm air and tends to fall to the floor. Warm air, being lighter, tends to rise. Because of this, pockets of stagnant air will form where room air motion is minimal, often resulting in large temperature gradients. Temperature differential near the floor, between average room air and stagnant air zones greater than 4°F, are undesirable. Higher primary air velocities will get room air moving and reduce stagnant air pockets.

Air volume of 0.75 to 1 cfm per square foot may satisfy cooling or heating loads at the low end of variable volume systems, however this may not provide enough air to prevent stuffy and stagnant room conditions. Changing the primary air temperature, higher for cooling and lower for heating, will increase supply air volumes and greatly improve comfort conditions.

At the high end of a variable volume system, interior zones may require as much as 3 cfm per square foot. Exterior zones may require more, often as high as 4 cfm per square foot. Air volumes for perimeter diffusers can vary from 20 to 200 cfm per linear foot. These volumes are governed by external wall heat transfer coefficients, wall height and infiltration rates.

Jet Temperature Change

As the jet progresses into the room, air next to the jet is drawn into it. The moving mass of air becomes more and more diluted by mixing with room air. The more it is diluted, the closer the jet temperature approaches room temperature. Temperature may be approximated by this equation:

$$\Delta tx = 0.8 \ \Delta to \left(\frac{Vx}{Vo}\right) \qquad (6)$$

Where: Δtx = Temperature differential degrees F at distance X ft. from outlet.

Δto = Temperature differential degrees F at outlet.

Vx = Velocity at centerline of jet distance X ft. from outlet.

Vo = Velocity of jet at outlet.

Jet Temperature						
Δto = 20°F	Vx fpm					
Vo fpm	500	400	300	200	100	50
2000	4.0	3.2	2.4	1.6	0.8	0.4
1500	5.3	4.3	3.2	2.1	1.1	0.5
1000	8.0	6.4	4.8	3.2	1.6	0.8
500	16.0	12.8	9.6	6.4	3.2	1.6

Table 1. Temperature Differential - Supply Air Stream Minus Ambient Room Temperature.

Throw

Throw is defined as the distance from the outlet to a location where the jet velocity, or terminal velocity, is some pre-determined value; usually 150, 100, 75 or 50 fpm. At 100 fpm the jet pattern is beginning to break down. At 50 fpm the jet has entered the fourth zone and has become a rolling mass of air. Here, air temperatures are within 1.0 degrees F of room temperature.

For zone III: $$Tv = \frac{K \; Qo}{Vt\sqrt{Ao}} \qquad (7)$$

For zone IV: $$Tv = K\left(\frac{Qo}{Vt}\right) \qquad (8)$$

Where: Tv = Throw to a distance where the centerline jet velocity is V fpm.
Qo = Supply Airflow Rate (cfm).
Vt = Terminal Velocity.

Spread

The divergence of an air stream vertically and horizontally after it leaves the outlet. In a free jet the spread proceeds at 22 degrees. Vertical grille vanes are adjustable to any angle.When set to deflect the outer 1/3 of the outlet in various degree patterns, the throw is reduced.

e.g. 22 1/2° pattern reduces throw by one quarter.
45° pattern reduces throw by one half.

Drop

Drop results from the vertical spread of an air stream in combination with the tendency for warm air to rise or cold air to fall.

Total Air Drop From Sidewall Outlet in Feet

Vk	Sidewall Throw In Feet													
Outlet Velocity	10		15		20		25		30		40		50	
fpm	-18F	-25F	-18F	-25F	-18F	-25F	-18F	-25F	-18F	-25F	-18F	-25F	-18F	-25F
500	3.5	4.0	5.5	6.0	7.5	8.5	9.0	10.0	10.5	13.5	15.5	18.0	18.5	23.0
750	2.5	3.5	4.0	5.5	6.0	6.5	7.0	8.0	8.5	10.5	11.5	14.5	15.0	18.5
1000	2.0	3.0	3.5	4.0	5.0	5.5	6.0	6.5	7.0	8.5	10.0	12.0	12.5	16.0
1250	2.0	2.5	3.0	3.5	4.5	5.0	5.5	6.0	6.5	7.5	9.0	11.0	11.5	13.5
1500	1.5	2.0	3.0	3.5	4.0	4.5	5.0	5.5	6.0	7.0	8.5	9.5	10.5	12.5
1750	1.0	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.5	8.0	9.0	10.0	11.5
2000	1.0	1.5	2.5	3.0	3.5	4.0	4.0	4.5	5.0	6.0	7.5	8.5	9.5	10.5

Table 2. Drop Due to Spread Plus Cold Air Fall.

Drop can be significantly lessened by projecting the air path upward. This is done by setting horizontal bars at 15° to 20° up.

	Throw in Feet						
	10	15	20	25	30	40	50
Drop Reduction in Feet	2.5	3.5	4.5	6.0	7.0	9.0	11.5

Table 3. Drop Correction Using 15° to 20° Upward Deflection Bars.

Effective Area

The effective area (Ak) of an outlet is not necessarily the area of the duct from which the air is projected. For grilles, the frame and vertical and horizontal bars restrict the opening. The spaces between bars are not completely filled with air. The sum of all these openings multiplied by the coefficient of contraction gives the effective area. The effective area, Ak, for each product is obtained by laboratory tests and will be found in the manufacturer's catalog.

$$Ak = \frac{Qo}{Vk} \qquad (9)$$

Where: Ak = The outlet effective area in sq. ft.
Qo = Quantity of air flowing, cfm.
Vk = Outlet velocity measured in a specified manner.

Entrainment Ratio

Also called the induction ratio, is the increased volumetric flow at a distance X from the face of the outlet, divided by the discharge volume.

$$\frac{Qx}{Qo} = C \frac{Vo}{Vx} \qquad (10)$$

$$C = \left(\frac{X}{K\sqrt{Ao}} \right) \qquad (11)$$

Where: Qx = Flow rate at section X feet from outlet.
Qo = Flow rate at outlet.
C = Entrainment coefficient
- 1.4 for slots.
- 2.0 for free round jets.
Vo = Outlet velocity, fpm.
Vx = Centerline velocity at X, fpm.
K = See equation 4.

Jet Behavior

- Free jets tend to become circular regardless of the original shape of the outlet, whether round, rectangular or a long slot. Jets hugging the ceiling or wall tend to become semi-circular.
- Wall air outlets within one foot of the ceiling project an air pattern which hugs the ceiling.
- Static room air drifts toward the fast flowing primary air stream where it is drawn into the expanding jet.
- The 'Coanda' surface effect propels the air further than if it were a free jet in space.
- Warm air has a buoyant effect and tends to rise.
- Cold air is heavier than warm room air and tends to fall.
- To reduce throw and drop, increase the spread.
- More outlets with less air each, will reduce throw and drop.
- Adjust vertical grille vanes to create a wide pattern and reduce throw. This reduces room temperature gradients, floor to ceiling.
- To achieve a wide pattern with shorter throw from a side wall diffuser, direct the air up to the ceiling at 45°.
- To reduce room air temperature gradients from floor to ceiling, increase primary air volumes or supply velocities.
- A spreading pattern produces shorter throws and thinner air stream with greater induction of room air.

Effect of Supply Air Patterns on Room Air Motion

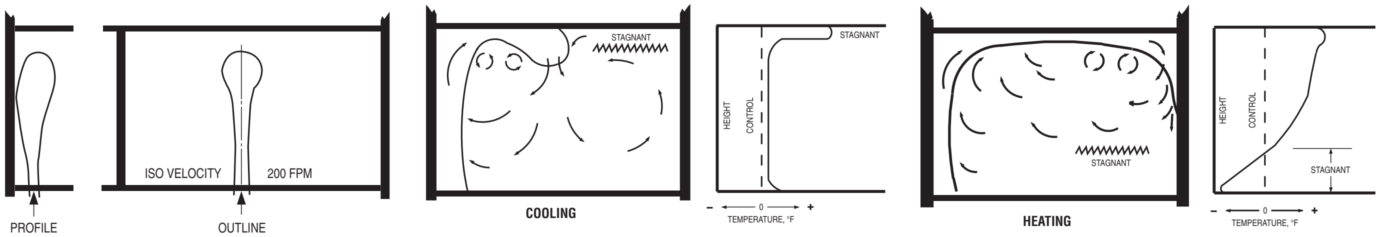


Fig. 4. Non-spreading Vertical Upward Jet. Perimeter Floor Grille.

- A long throw reaching far into the room. Room air is drawn to the jet in the center of the room.

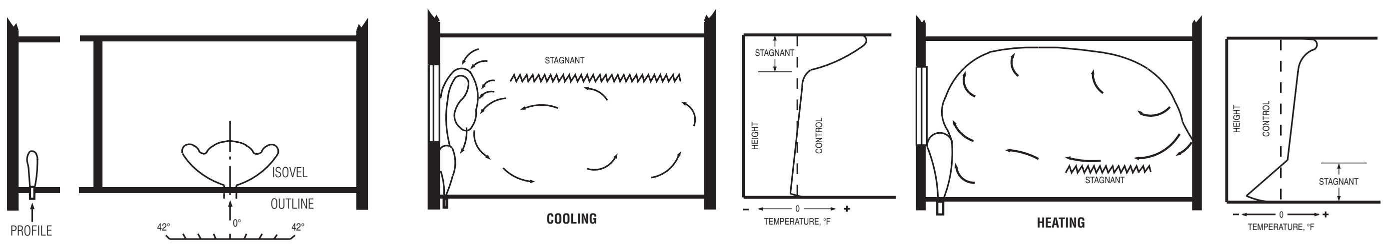


Fig. 5. Spreading Vertical Upward Jet. Floor Diffuser.

- When heating, large temperature variations occur near the floor in the stagnant zone.
- A spreading pattern provides a thinner air stream with greater room air induction, but does not project as far into the room. The tendency to drop is also reduced.

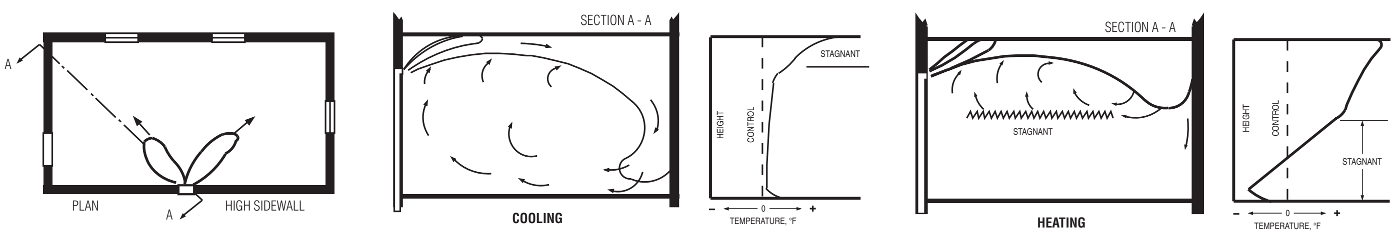


Fig. 6. Spreading Horizontal Projection at Ceiling Level. High Sidewall Grille.

- The air pattern will be across the ceiling and down the wall to the floor. When heating, this results in a smaller stagnant zone with more even temperatures in the occupied zone.

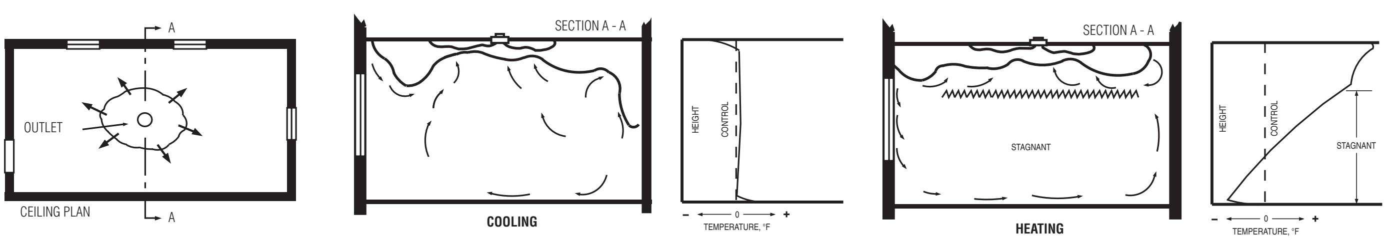


Fig. 7. Radial Horizontal Projection. Ceiling Diffuser.

- Ideal for year round cooling. Thinner air pattern gives greater spread and more uniform space temperatures.
- Wide temperature variations when heating.

Heating From Above

Heating a space by the downward projection of hot air from the ceiling is not recommended. However, if it becomes necessary, the designer should be guided by the following notes:

• Where possible, heat outside walls and windows by radiation, floor outlets, or under window units. Heat should be supplied at the location where it is lost to the outside.

• Temperature of downward projected hot air should not be more than 20° to 25°F higher than room temperature.

• When projecting air horizontally from slots, locate the slot several feet from a cold wall so that the jet 150 fpm throw distance occurs at the junction of the ceiling and wall. The air envelope will spread over the wall as it progresses toward the floor. The terminal 50 fpm throw point cannot be easily determined. It may occur close to the floor, but because of the buoyancy of the warm air it will be much less than the catalog throw data for cold air.

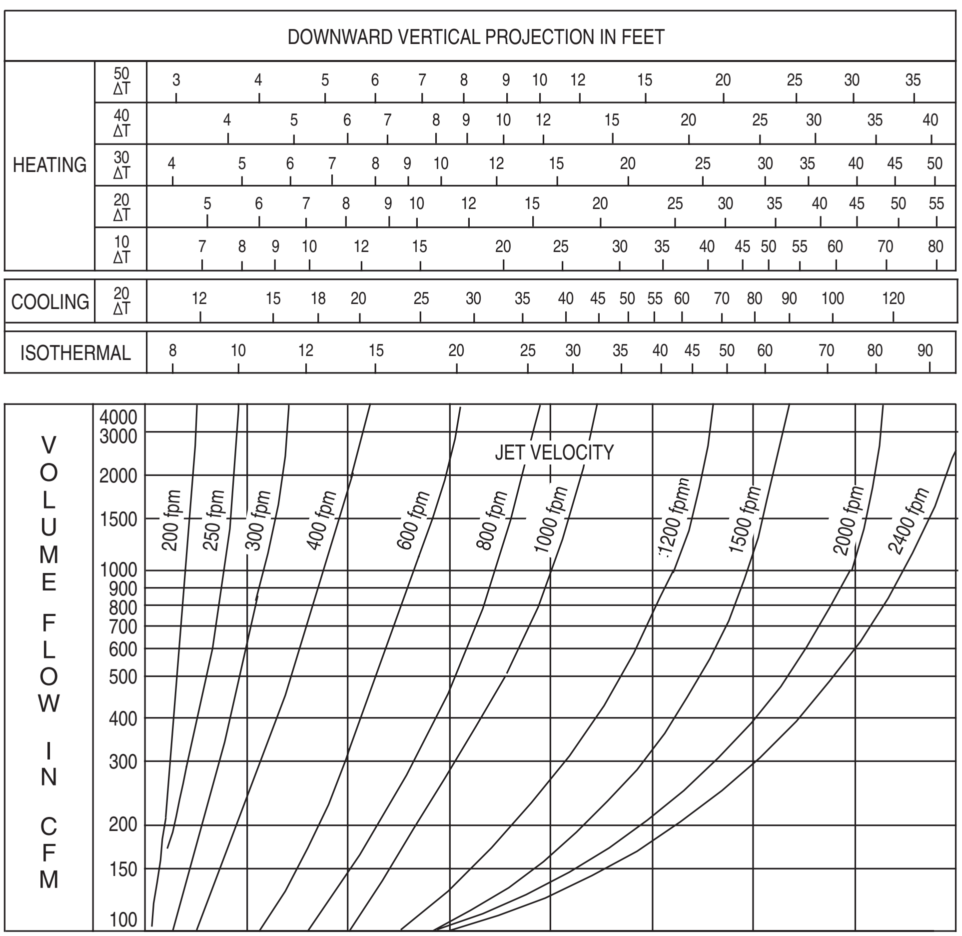
• Consideration must be made for outlet performance during the summer cooling season. Outlet throw to 150 fpm is the same as for heating, i.e., equal to the distance to the wall. The distance from 150 fpm to 50 fpm should be the same as approximately the distance from ceiling to the floor.

• A continuous slot directed toward the wall should be broken into active and inactive sections to allow the warm air to spread over the wall surface.

Table 4. Location of Inactive Sections of Slot Diffuser.

Length of Active Sections in Feet	1	5	10
Length of Inactive Sections in Feet	1	2	3

Fig. 8. Downward Projection of Hot Air from Grilles.



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Air Temperature and Direction

A temperature of 80°F is recommended when air is blowing directly on the subject. Air should be directed to the front upper torso.

People's tolerance to air temperature, air motion, and humidity varies a great deal. Provisions should be made for the individual to control air direction and velocity.

Outlet Location

For local area ventilation, outlets should be at about the 10 foot level. For spot cooling the outlet should be at about the 7 foot level and as close to the subject as is practical. The closer to the subject the less room air is induced into the jet stream.

High Velocity Jets

High outlet velocities are usually chosen in the 1000 to 2000 fpm range, although in extremely uncomfortable conditions velocities of up to 4000 fpm may be directed at a worker, and only for very short periods of time.

Higher velocities require higher pressures. This will create higher noise levels and induce more hot local air into the jet.

Discharge Volume

For spot cooling, volumes between 1500 and 5000 cfm per outlet are frequently used. For large areas volume may exceed 10,000 cfm per outlet. When the supply outlet is far away from the work station large amounts of room air are induced into the primary air, making it less effective. The longer the throw the greater the air volume required.

Directional Control

There should be some means of controlling the direction of the air for the following reasons:

- Seasonal: up in winter, down in summer.
- Change from one work station to another.
- Direct the air away from the product or process.
- Personal preference of the worker.

Individual volume control units are desirable for each outlet. Their use simplifies balancing of the system and ensures that the balance is not changed when one outlet is adjusted.

Special Features Are Available

- Insect screens or bird screens.
- Filter grilles.
- Special fastenings for quick and easy grille removal.
- Concealed fasteners and tamper-proof screws for security.
- Heavy duty security grilles.
- Stainless steel construction for corrosive atmospheres.
- Fire dampers.

Industrial Ventilation

Maintaining satisfactory working conditions in industrial situations often requires special treatment. A common problem is how to improve the environment for workers who are subject to high temperatures or noxious fumes. There are generally three problem categories:

(a) The hot and dry environment where the worker is subject to sensible and radiant heat exposure. An exposed worker will sweat profusely but heat loss by evaporation cannot match the high radiant heat gain the worker is receiving. Examples are found in the steel industry, rolling mills, open hearth furnaces, foundries and forging operations.

(b) High ambient temperatures due to weather, aggravated by the low air motion in the building. Examples are warehouses, and the light manufacturing industries where the working area is so large that it is not practical to air condition it. The worker cannot get cool enough by sweating because of high humidity and lack of air motion.

(c) Warm moist environments where the worker is subject to latent heat from wet processes. Examples are dye houses, textile mills, laundries and tanneries. The high humidity inhibits the evaporation of sweat and prevents the worker from keeping cool by natural processes.

Heat Shields

For hot, dry exposure use heat shields to reduce radiant heat loads. Where possible lower the temperature of hot objects by insulation, water cooling, or shielding. Heat reflective clothing is used with success. Install exhaust hoods to draw off process heat. Cooler air will flow over the worker as it progresses toward an exhaust hood.

Local Ventilation

When it is impractical to supply treated air in volumes large enough to control temperature, humidity, or air motion, the best method is to blow air directly on an individual worker or into a work station. The worker is now cooled by convection and the evaporation of sweat.

Supply air to the desired location by ducts or use "man cooler" portable fans.

Treated air is supplied at the 8 to 12 foot level to displace air heated by machines, processes, lights and people. No attempt is made to treat air above or outside the chosen work area.

In some cases the best method is to supply conditioned air to an enclosure which surrounds the worker.

Spot cool by blowing high velocity air directly on the worker.

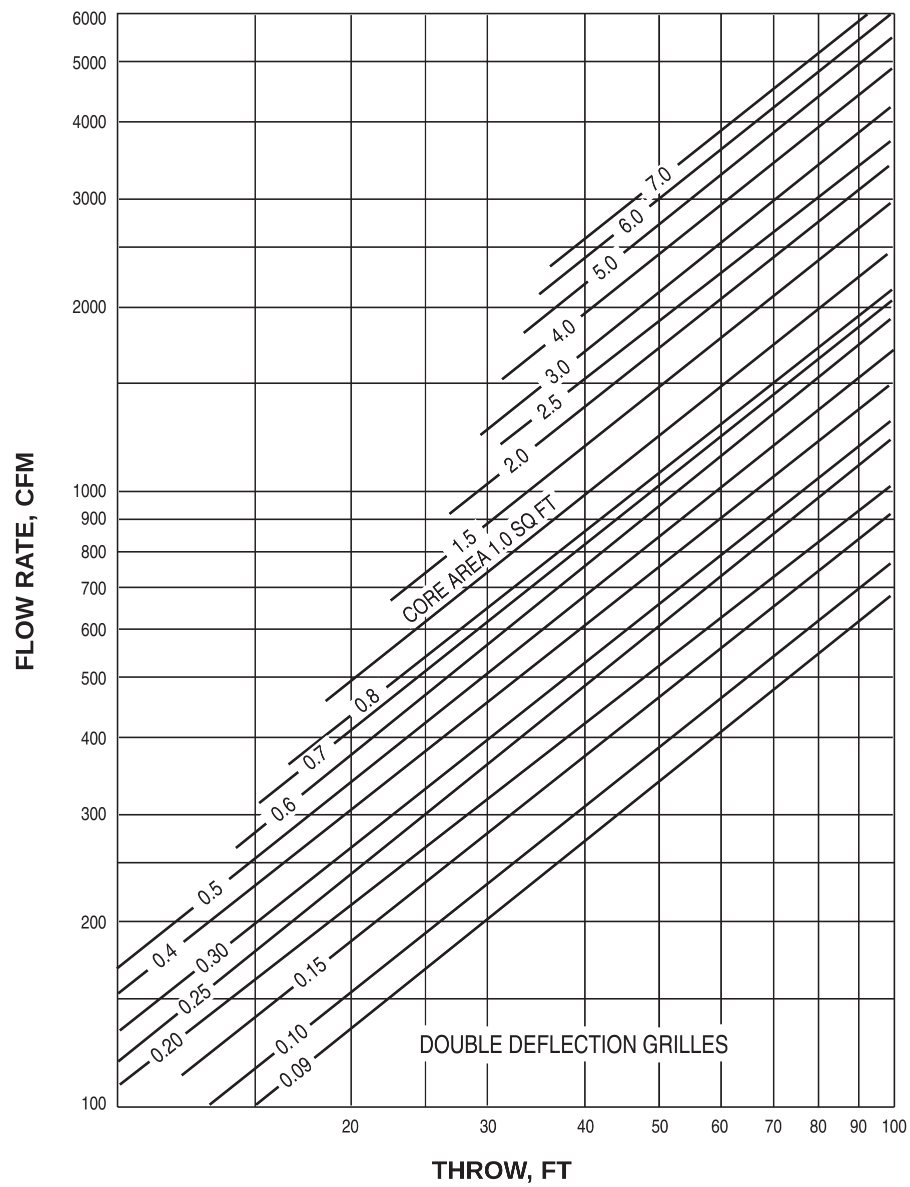
Tolerable Velocities

Air velocities from 50 to 80 fpm are acceptable in areas of light activity.

Calculated jet velocity is the centerline velocity of the jet. The average velocity away from the center of the jet will be much less.

It is undesirable to blow air directly on a worker at a velocity greater than 200 fpm for a long period of time.

Double Deflection Grilles and Registers
Supply Grille and Register Selection for Commercial Applications



THROW IS BASED ON 75 FPM TERMINAL VELOCITY. (Vt)

Fig. 9. Grille and Register Selection for Commercial Applications – Double Deflection Grilles with a Damper, 3/4" Blade Spacing.

For other terminal velocities multiply by factor.					
Vt 50	**Vt 75**	**Vt 100**	**Vt 150**	**Vt 200**	**Vt 300**
1.5	1.0	0.75	0.50	0.375	0.25

Vt = Terminal Velocity.

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Adjustable Vane Supply Single and Double Deflection Grilles

Outlet Area In Square Feet

Example: Listed size 28 x 28 @ 0°, Ao area factor = 3.54 square feet.

LISTED WIDTH	LISTED HEIGHT																								AO		
	4	5	6	8	10	12	14	16	18	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48	0°	22.5°	45°
	8		6																						0.14	0.12	0.10
	12	10	8																						0.18	0.16	0.14
	16	12	10																						0.24	0.21	0.18
		14		8																					0.26	0.22	0.20
	18		12																						0.29	0.25	0.22
			14	10																					0.34	0.30	0.26
	24		16	12																					0.39	0.34	0.30
	26				10																				0.41	0.36	0.31
	30		18	14																					0.44	0.38	0.33
		24	20		12																				0.50	0.44	0.38
	36	28	22	16																					0.54	0.47	0.41
	38		24	18	14	12																			0.61	0.53	0.46
			30		18																				0.77	0.67	0.58
			34	24	20	16	14																		0.84	0.73	0.64
			38	28	22	18	16																		0.93	0.81	0.71
				30	24	20																			1.03	0.90	0.78
						22	18	16																	1.12	0.97	0.84
				36	30	24	20	18																	1.26	1.09	0.95
					32	28	24	20	18																1.43	1.24	1.08
					36	30	26	22	20																1.58	1.37	1.19
						32		24																	1.70	1.48	1.29
									22	20															1.77	1.54	1.34
						36	30	26	24	22															1.90	1.65	1.44
						40		30	26	24	22														2.16	1.87	1.63
						42	36																		2.22	1.93	1.68
							40	34	30		24														2.41	2.09	1.82
								36	32	28	26	24													2.58	2.24	1.95
							46	40	36	32															2.92	2.53	2.21
							48					28	26												3.04	2.64	2.30
									40	36	32	30													3.24	2.81	2.46
									42				28												3.39	2.94	2.57
										40	36		30	28											3.54	3.07	2.68
										42				30											3.79	3.29	2.87
										44	40	36													3.90	3.39	2.96
										48		38	34		30										4.07	3.53	3.08
											46	42		36											4.57	3.96	3.46
													40			32									4.65	4.04	3.52
														38	36										4.91	4.26	3.72
												48		42	38	36	34								5.23	4.54	3.96
													48		40	38	36								5.58	4.84	4.22
														46	42		38	36							5.91	5.13	4.48
																	42		38						6.60	5.72	5.00
															48		44	42							6.91	5.99	5.23
																48	46		42	40					7.32	6.35	5.55
																		48	46	44	42				8.09	7.02	6.12
																					46	44			8.89	7.71	6.73
																							46		9.72	8.44	7.36
																								48	10.60	9.20	8.03

Table 5.

Grille Air Measurement

cfm = AoVo (12)

Ao = Outlet Area, sq. ft.

Vo = Outlet Velocity, fpm.

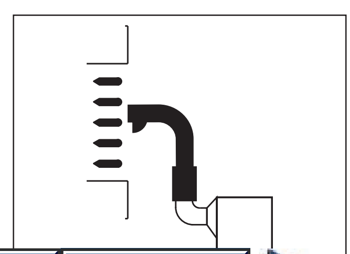


Fig. 10. Using 2220A Velometer Tip to Measure Vo Outlet Velocity.

High Capacity Grilles and Registers Supply Grille and Register Selection for High Capacity Industrial Applications

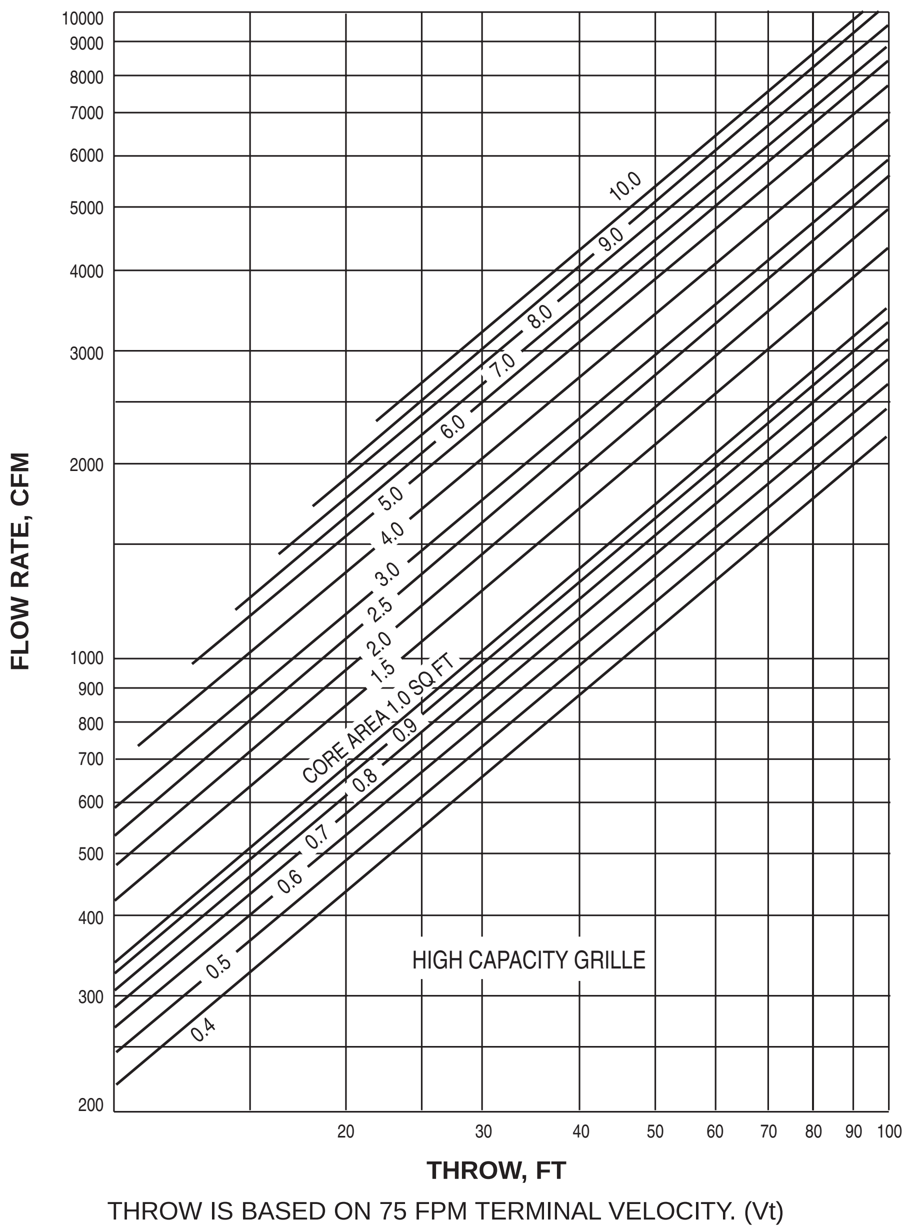


Fig. 11. Grille and Register Selection for Industrial Applications – Double Deflection Grilles with a Damper, 1 1/2" Blade Spacing.

For other terminal velocities multiply by factor.				
Vt 50	**Vt 75**	**Vt 100**	**Vt 150**	**Vt 200**
2.0	1.3	1.0	0.7	0.5

Vt = Terminal Velocity.

Fixed Vane Return Single Deflection Grilles

Outlet Area In Square Feet

Example: Listed size 28 x 28,
Ao area factor = 4.85 square feet.

LISTED WIDTH \ LISTED HEIGHT	4	5	6	8	10	12	14	16	18	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48	AO
	8		6																						0.23
	12	10	8																						0.30
	16	12	10																						0.37
		14		8																					0.40
	18		12																						0.45
	24		16	12																					0.59
	26				10																				0.62
	28		18	14																					0.67
		24	20	16	12																				0.74
	38		24	18	14	12																			0.89
			34	24	20	16	14																		1.22
			38	28	22	18	16																		1.34
				30	24	20																			1.49
				30		22	18	16																	1.58
				36	30	24	20	18																	1.78
					32	28	24	20	18																2.01
					36	30	26	22	20																2.23
						36	30	26	24	20															2.48
							36	30	26	24	22														3.00
							40	34	30		24														3.34
								36	32	28	26	24													3.56
							46	40	36	32															4.01
							48					28	26												4.19
									40	36	32	30	28												4.46
										40	36		30	28											4.85
										44	40	36		30											5.35
										48		38	34		30										5.57
											46			38	36	32									6.34
												48		48	38	36	34								7.13
													48	46	42		38	36							8.02
															48		42		38						8.94
																48	46	42		40					9.90
																		48	46	44	42				10.92
																					46	44			11.98
																							46		13.10
																								48	14.26

Table 6.

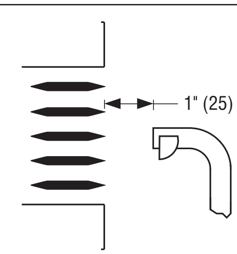
Grille Air Measurement

cfm = AoVo (13)

Ao = Outlet Area, sq. ft.

Vo = Outlet Velocity, fpm.

Fig. 12. Using 2220A Velometer Tip to Measure Vo Outlet Velocity.



Noise Level in a Ventilated Space

Noise in a ventilated space may come from a wide variety of sources. Noises in the room can result from air flowing through ceiling diffusers or sidewall grilles. Air outlets also radiate noise generated in the duct system, such as fan noise, air flowing around dampers, turning vanes, or through terminal units. Noise from outside the room may be machinery noise, or traffic noise. Our interest in this discussion is limited to the noise generated by grilles, ceiling diffusers, and noise emanating from air supply or return outlets.

It is generally assumed that it is desirable to have completely silent air systems. Although this is necessary for some situations, such as concert halls or live theatre, it is not always the case. For example, in office areas without suitable masking noise, it is often necessary to provide background music. A comfortable background noise level will mask sudden disruptive noises such as from traffic or garbage collection. Another important effect of background noise is that it provides a degree of speech privacy so there is no need to talk behind closed doors.

The ideal background noise should not be loud enough to interfere with normal speech. It should be well balanced, not all high pitched sounds or all low sounds.

For example, a low duct noise will complement higher octave diffuser noise and provide a balanced pleasant acoustic background.

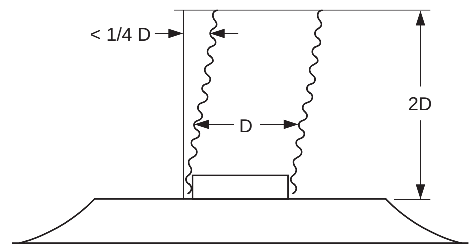
Noise from the duct system may be falsely attributed to the performance of the diffuser. Removal of the diffuser core quickly determines whether the offending noise is from the duct system or from the diffuser. Duct system noise should be 5 dB less than the diffuser dB rating.

The level of noise is measured in decibels. Air outlet manufacturers catalog noise values in NC numbers for air flowing through their products. The more air handled, the higher the NC value. It is generally assumed that an average office or ventilated space will have a room attenuation of 10 dB so suppliers' catalogs give product NC numbers as Sound Pressure Level with the 10 dB already deducted.

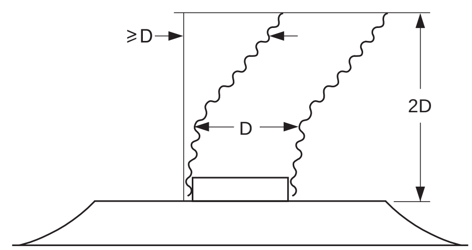
Diffuser Inlet Conditions

Performance data as shown in this catalog is obtained in accordance with ANSI/ASHRAE Standard 70 – 2006. This standard provides the laboratory method for testing and rating the performance of air outlets and inlets. The standard does not however, take into effect the real world conditions often encountered in typical installations. For example, it assumes a uniform velocity distribution throughout the grille or diffuser neck. This is obviously not always the case in practice, with the use of flexible duct, elbows, inlet dampers, etc... the velocity could vary substantially. These conditions could cause greatly increased turbulence and substantially higher sound levels than those indicated by the tested catalog data. Also, the sound levels are for only one diffuser, multiple diffusers in the same area will increase the resultant sound level.

Inlet conditions are crucial in the proper performance of an outlet. With the common use of flexible duct, this is often overlooked and can result in significant increases in sound levels. If the flexible duct connection is misaligned less than 1/4 the diffuser neck diameter over a connection length equal to twice the inlet diameter, no significant change in sound levels will occur. However, if the connection is misaligned equal to or more than the neck diameter over a distance of twice the inlet diameter, the resultant sound levels can increase by as much as 12 dB.



Good Inlet Conditions
(Sound levels as catalogued)



Poor Inlet Conditions
(Sound levels up to 12 dB higher than catalog)

Balancing dampers are another significant source of noise in any HVAC system. Ideally, balancing dampers should be located as far from the air outlet as possible, (5 to 10 duct diameters), and acoustic duct should be used between them. The use of volume dampers on diffuser inlets can significantly increase sound levels due to the effect on the air turbulence between the damper and the diffuser as well as the resultant increase in pressure drop.

I

ENGINEERING GUIDE

Diffuser Damper Sound Correction

The following table provides an addition to be added for ceiling diffusers with neck mounted opposed blade dampers.

Table 7.
Diffuser Damper Sound Correction.

	Pressure Drop – In. w.g.		
Damper throttling effect	0.05"	0.15"	0.25"
Approximate damper opening	3/4	2/3	1/2
NC add to single outlet sound rating	5	10	15

The following table provides an addition to be added for linear diffusers based on the damper pressure ratio. This is the pressure drop across the partially closed damper divided by the pressure drop across the fully open damper.

Table 8. Decibels to Be Added to Diffuser Sound Rating to Allow for Throttling of Volume Damper

	Damper Pressure Ratio					
	1.5	**2**	**2.5**	**3**	**4**	**6**
Location of Volume Damper	**dB to be added to Diffuser Sound Rating**					
In neck of linear diffuser	5	9	12	15	18	24
In inlet of plenum of linear diffuser	2	3	4	5	6	9
In supply duct at least 5 ft. (1.5 m) from inlet plenum of linear diffuser	0	0	0	2	3	5

Air extractors are another commonly used accessory that must be applied properly in order to avoid a detrimental effect on the system design. They should only be used when the duct is wide enough to allow the device to open to its maximum position without causing undue restriction of the airflow in the duct. Otherwise, they could limit downstream airflow, increase duct velocity and increase sound levels.

Equalizing grids can have positive impact on diffuser inlet conditions and reduce the resultant sound levels if they are used properly. A poor inlet condition that results in a noise problem can sometimes be helped substantially by using equalizing grids in the neck of the diffusers or at the branch take off to the diffuser. However, an equalizing grid will not help a good inlet condition and can in fact add 2 - 3 dB to the diffuser sound levels in ideal conditions.

Flow generated noise should be minimized wherever possible by locating elbows or branch takeoffs at least 4 to 5 duct diameters from each other. This will also help to reduce sound transmission from one room to another through the duct.

Sound Pressure Level

The noise we hear in a space is not the absolute "Sound Power Level" noise generated by these sources, but it is the Sound Power Level minus the attenuating value of the space. This is called "Sound Pressure Level". The space attenuation is due to the sound absorptive value of ceiling tiles, walls, drapes, windows, floor, rugs, furniture, people, etc...

Sound Pressure Levels for Evenly Distributed Ceiling Diffusers

Where there are four or more similar ceiling diffusers, evenly distributed about the room, Lp may be calculated using formula 15. Where there are rows of linear diffusers, use the sound power level of a single section for Lws and the number of sources as the number of sections in the array.

Lpt, 5 ft. = Lws - 5 log X - 28 log h + 1.3 log N - 3 log f + 31 dB (15)

Where: Lpt = The average sound pressure level (+ or - 1 dB) in a plane 5 feet off the floor, in dB re 20 µPa.

Lws = Sound power level of a single outlet (outlet sound power plus duct noise and any noise created at the air terminal) in dB re 10^{-12} watts.
X = Ratio of floor area served by each outlet to the square of the ceiling height.
h = Ceiling height in feet.
N = Number of ceiling outlets in the room (4 or more).
f = Octave band center frequency in Hz.

Estimating Sound Pressure Levels

Lp = Lw - 5 log V - 3 log f - 10 log r + 25 dB (14)

Where: Lp = Room sound pressure level at a reference location in dB re 20 µPa.

Lw = Source sound power level in dB re 10^{-12} watts.
V = Room volume in cubic feet.
f = Octave band center frequency in Hz.
r = Distance from the sound source to reference location in feet.

This equation is for a single noise source in the room. The total sound pressure level is obtained by adding the Lp at r distance from the reference location for each additional noise source, on an energy basis.

Fig. 13. Combining Two Sound Levels.

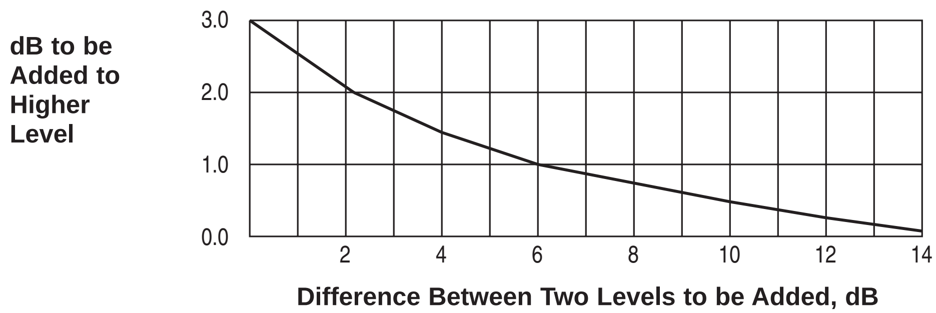


Table 9. Combining Several Sound Levels.

Number of Multiple Outlets	2	3	4	5	6	8	10
Increase in Noise Level, dB	3	5	6	7	8	9	10

RC (Room Criterion) Curves

It has been the practice to use NC curves to specify the noise level in an occupied space. This simple one number method of specifying acoustic conditions has been used successfully for many projects. However, there are some HVAC duct values specified which, although the installation may meet the NC values indicated, are still not acceptable. It has been found that predominant low octave band noise which does not exceed the specified NC curve can generate a low rumble. This is difficult if not impossible to correct once the installation is complete. At the other end of the scale, a predominant high octave band noise may follow the specified NC curve but will generate a hissing sound which can not be tolerated. To try to prevent system designs which result in such conditions, a revised set of curves has been developed called RC curves.

Chapter 42 of the ASHRAE Fundamentals Handbook recommends the use of RC curves in the design of air conditioning ventilating systems as a tool to help prevent acoustical disasters. RC curves eliminate the 8000 Hz octave band but use the 31.5 Hz octave band and the 16 Hz band to avoid the possibility of low frequency fan noise. An RC designation is based on a speech interference rating, a dB rating and a letter of the alphabet to describe the quality of the sound.

Determining an RC Noise Rating

1. Plot the octave band sound pressure level spectrum on an RC chart.
2. Calculate the Speech Interference Level (SIL) by taking the arithmetic average in dB of the sound level reading in the 500, 1000, 2000 Hz octave bands.
3. Mark the SIL on the 1000 Hz band line.
4. Draw a line with a slope upward to the left of 5 dB per octave through the SIL point, from the 4000 to 31.5 Hz octave band lines. This has established a reference line for evaluating the sound quality of the sound spectrum.
5. Draw a line from the 500 to 31.5 Hz octave band line parallel to, and 5 dB above, the line drawn in step 4. Draw a second line from the 1000 to 4000 Hz octave band 3 dB above the lines drawn in step 4. If the sound spectrum does not extend above these boundary lines the RC noise rating is labeled neutral (N).
6. The RC noise rating is the value of the SIL point on the 1000 Hz band line.
7. The quality of the sound can be determined by the manner in which the sound spectrum goes beyond the boundary limits. This is specified alphabetically as follows:

Neutral Spectrum (N)
The plotted levels in the octave bands must not exceed the levels of the boundary lines constructed as above. Example Fig. 14.

Rumbly Spectrum (R)
The levels of the octave bands less than 500 Hz exceed the lower octave band boundary line levels. Example Fig. 15.

Hissy Spectrum (H)
The levels in the octave bands higher than 1000 Hz exceed the higher octave band boundary line level. Example Fig 16.

Tonal Spectrum (T)
A sharp peak in the sound spectrum in a particular octave band, which is 3 dB or more above a line joining the readings of the octave bands on either side of the sound spike.

Acoustically Induced Perceptible Vibration (RV)
If the sound spectrum reaches the cross- hatched region on the RC chart, the sound energy could be sufficient to induce vibration in walls and ceilings of light building structures. Sound pressure readings in the 16 Hz octave band are important because they reveal any HVAC noises of significant energy. Besides creating a deep rumbling noise, these can induce audible rattles in light fixtures, air diffusers and grilles. Example Fig 17.

RC Noise Ratings

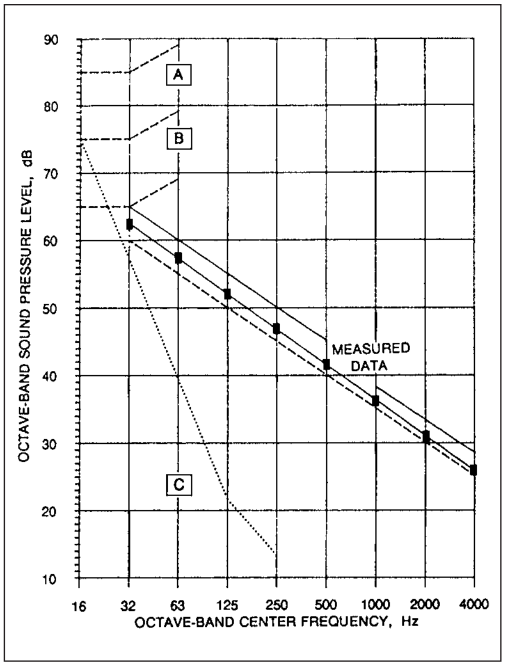


Fig. 14.
Neutral RC 35 (N).

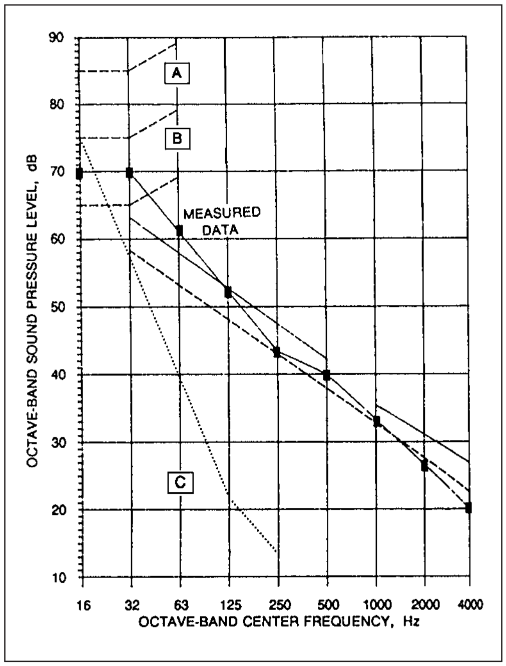


Fig. 15.
Rumbly RC 33 (R).

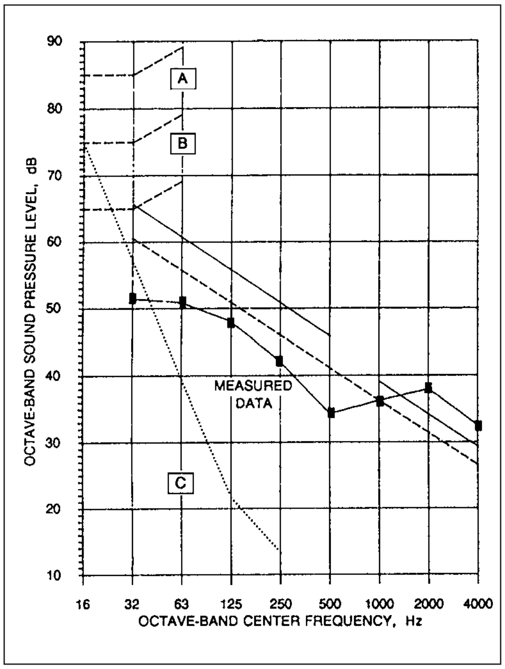


Fig. 16.
Hissy RC 37 (H).

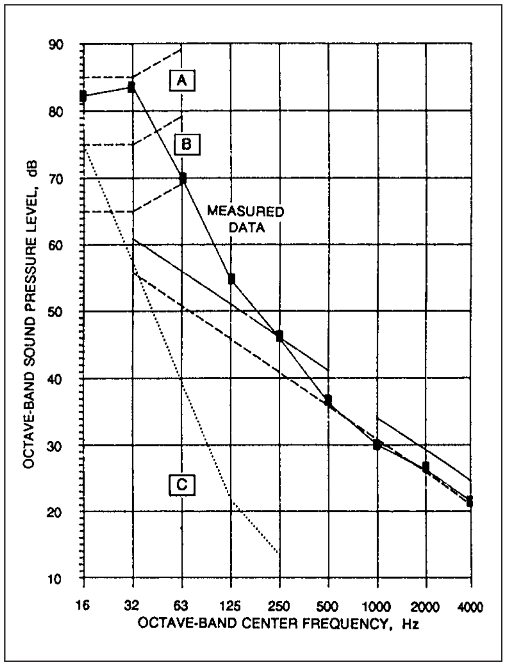


Fig. 17.
Rumbly and Induced Vibration RC 31 (RV).

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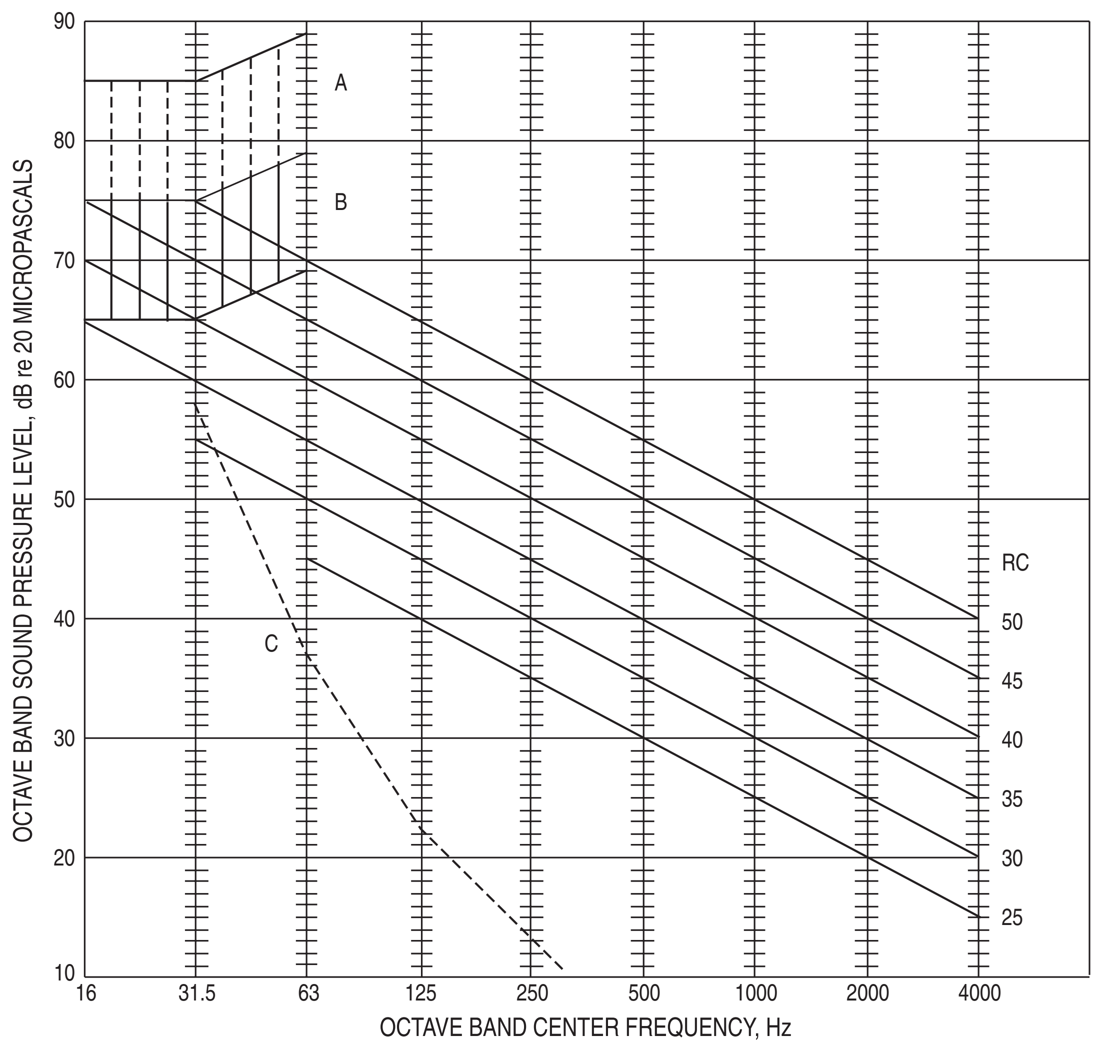
Acceptable HVAC Noise Levels

Room noise levels must not mask sounds that people want to hear nor be of an obtrusive or annoying character. The RC system of rating noise levels should be used where a neutral unobtrusive background sound is desired. If rumbles, hisses and tonal noise is acceptable, then the alternate NC method of specifying sound can be used.

RC Curves



Region A: High probability that noise-induced vibration levels in light wall and ceiling construction will be clearly felt; anticipate audible rattles in low mass fixtures, doors, windows, etc.

Region B: Noise-induced vibration levels in light wall and ceiling constructions may be moderately felt; slight possibility of rattles in low mass fixtures, doors, windows, etc.

Region C: Below threshold of hearing for continuous noise.

Fig.18. RC (Room Criteria) Curves for Specifying Design Levels to give a Balanced Sound Spectrum.

Alternate NC Noise Criteria Curves

Noise Criteria, or NC numbers provide a single number method of specifying a noise spectrum. To determine a Noise Criteria number, the Sound Pressure Level in each octave band is plotted on an NC Chart. The resulting NC number is the number of the maximum NC curve on the chart, tangent to the plotted data.

NC Curves

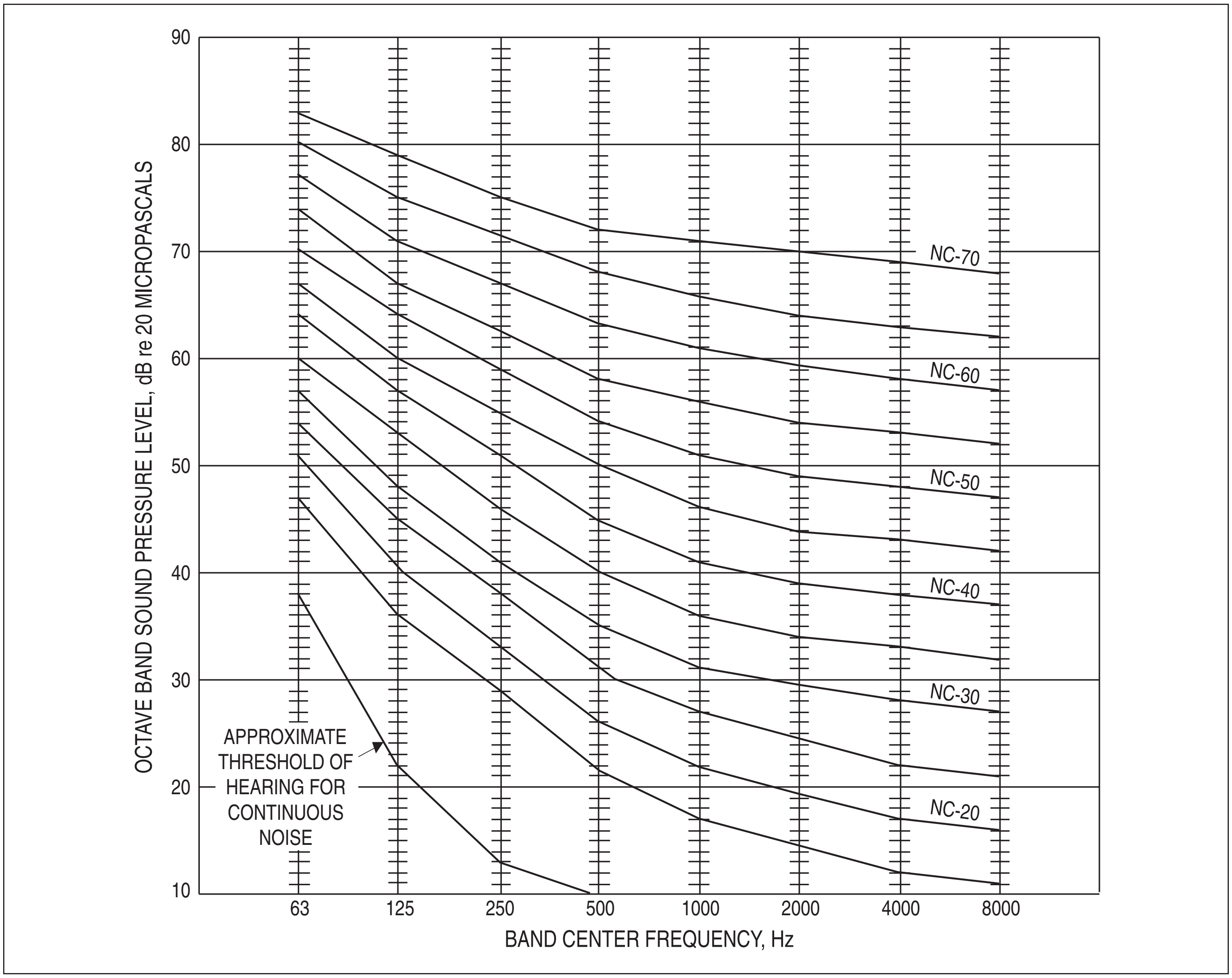


Fig. 19. Noise Criteria Curves for Specifying Desired Sound Levels.

Table 10. Acceptable HVAC Noise Levels in Unoccupied Rooms.

	Preferred	Alternate
Private Residences	RC 25-30 (N)	
Apartments	RC 30-35 (N)	NC 30-35
Hotel/Motel		
Bedrooms/Suites	RC 30-35 (N)	NC 30-35
Meeting/Banquet Rooms	RC 30-35 (N)	NC 30-35
Halls, Lobbies, Corridors	RC 35-40 (N)	NC 35-40
Service Areas	RC 40-45 (N)	NC 40-45
Offices		
Executive	RC 25-30 (N)	
Conference Rooms	RC 25-30 (N)	
Private	RC 30-35 (N)	
Open-plan Areas	RC 35-40 (N)	NC 35-40
Business Machines/Computers	RC 40-45 (N)	NC 40-45
Public Circulation	RC 40-45 (N)	NC 40-45
Hospitals		
Private Rooms	RC 25-30 (N)	
Wards	RC 30-35 (N)	
Operating Rooms	RC 25-30 (N)	
Laboratories	RC 35-40 (N)	NC 35-40
Corridors	RC 30-35 (N)	NC 30-35
Public Areas	RC 35-40 (N)	NC 35-40
Churches	RC 30-35 (N)	
Schools		
Lecture/Classrooms	RC 25-30 (N)	
Open Plan Classrooms	RC 35-40 (N)	NC 35-40
Libraries	RC 35-40 (N)	
Courtrooms	RC 35-40 (N)	
Legitimate Theaters	RC 20-25 (N)	
Movie Theaters	RC 30-35 (N)	NC 30-35
Restaurants	RC 40-45 (N)	NC 40-45
Concert and Recital Halls	RC 15-20 (N)	
Recording Studios	RC 15-20 (N)	
TV Studios	RC 20-25 (N)	

ADPI

Air Diffusion Performance Index

Dr. Ralph Nevins has made a thorough study of room comfort conditions under the following circumstances. He has tested room air velocity and temperature patterns using a variety of air supply systems, each with different room heating and cooling loads. The air supply systems tested were:

Floor grilles with non-spreading and spreading air patterns.
Sill grilles with vertical blow, 22 1/2° and 45° upward blow.
High sidewall grilles.
2 slot and 4 slot ceiling diffusers to Vt of 50 and 100 fpm.
Groups of 4 and 8 troffer diffusers.
Circular ceiling diffusers.
Square perforated and louvered ceiling diffusers.

Airflow rates – cfm/sq ft of floor – 0.5, 1, 2, 3, 4, 5
Room heating loads – Btuh/sq. ft. of floor – min, 17.5, 35, 50, 70

Dr. Nevins developed Air Diffusion Performance Index numbers for each different ventilating system. These ADPI numbers were based on criteria which he judged would satisfy the comfort expectations of most people.

The **Air Diffusion Performance Index** is the percentage of measurements of air velocity and temperature, made on a horizontal and vertical grid pattern in an occupied space, which meet the following conditions:

- room air velocities 70 fpm or less.
- temperature difference from floor to head height within -3 degrees or +2 degrees F of the room ambient temperature.

Curves were plotted for each system, ADPI against T_{50}/L.
These curves give the designer a tool which enables him to pick the best diffuser system for the project.

T_{50} = the distance in feet from the diffuser outlet to the place where the centerline jet velocity is 50 fpm.
L = the desired length of throw to give proper air distribution.

Comfort Index (APDI) vs. Throw Ratio (T_{50} / L)

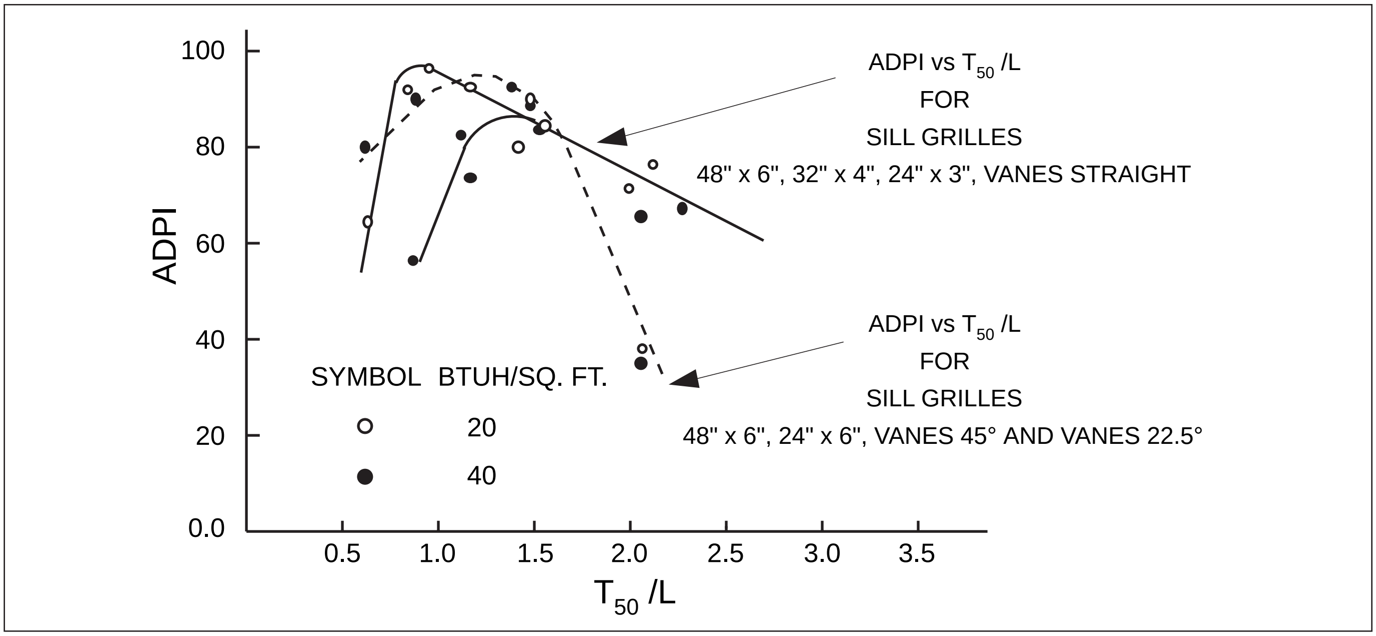


Fig. 20. Sill Grilles.

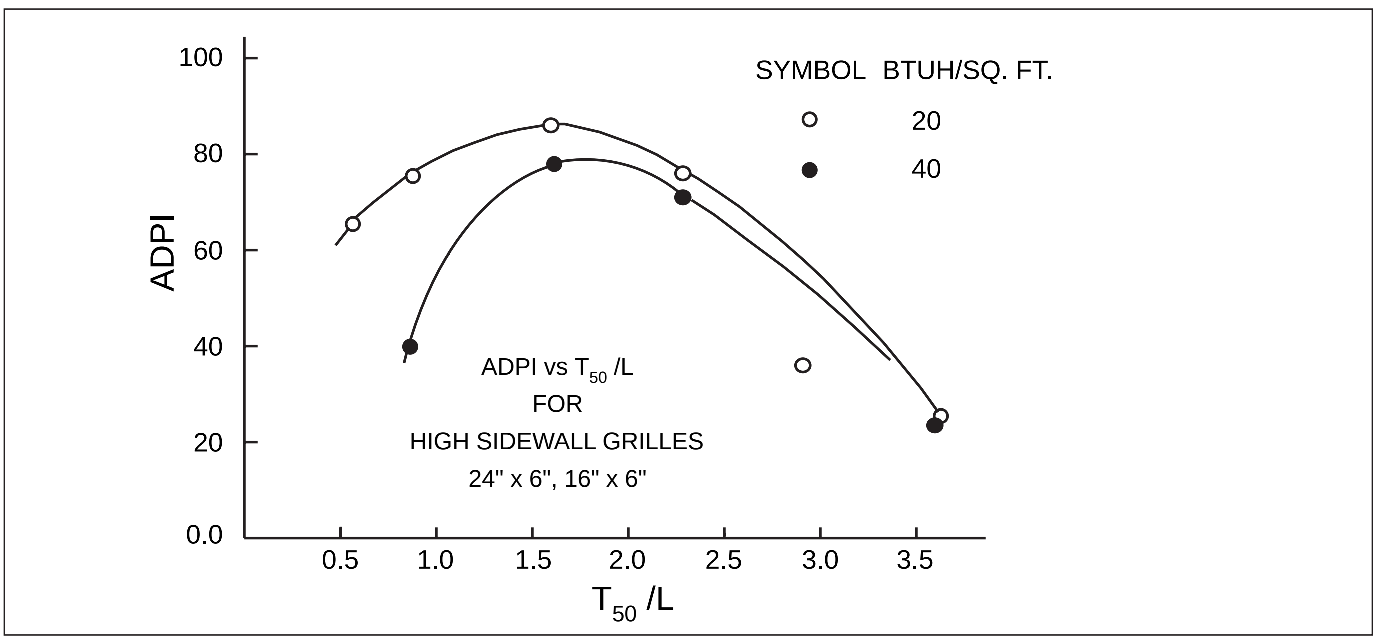


Fig. 21. High Sidewall Grilles.

Comfort Index (APDI) vs. Throw Ratio (T_{50} / L)

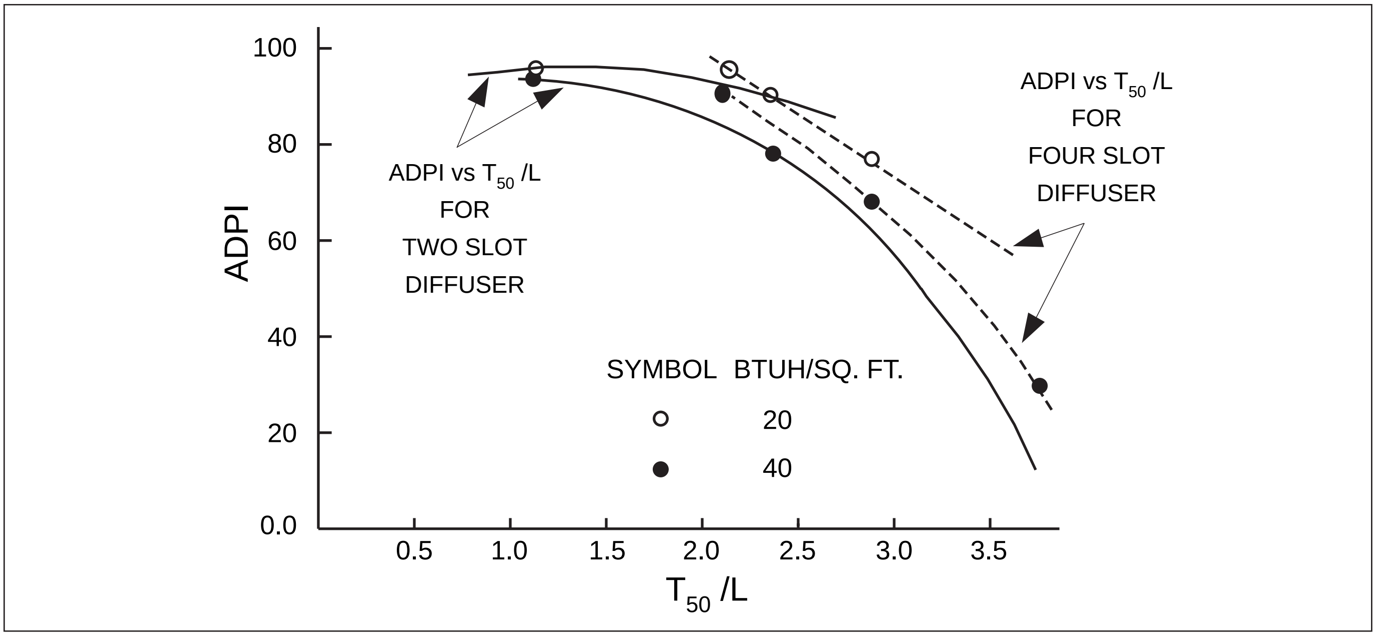


Fig. 22. 2 and 4 Slot Diffusers.

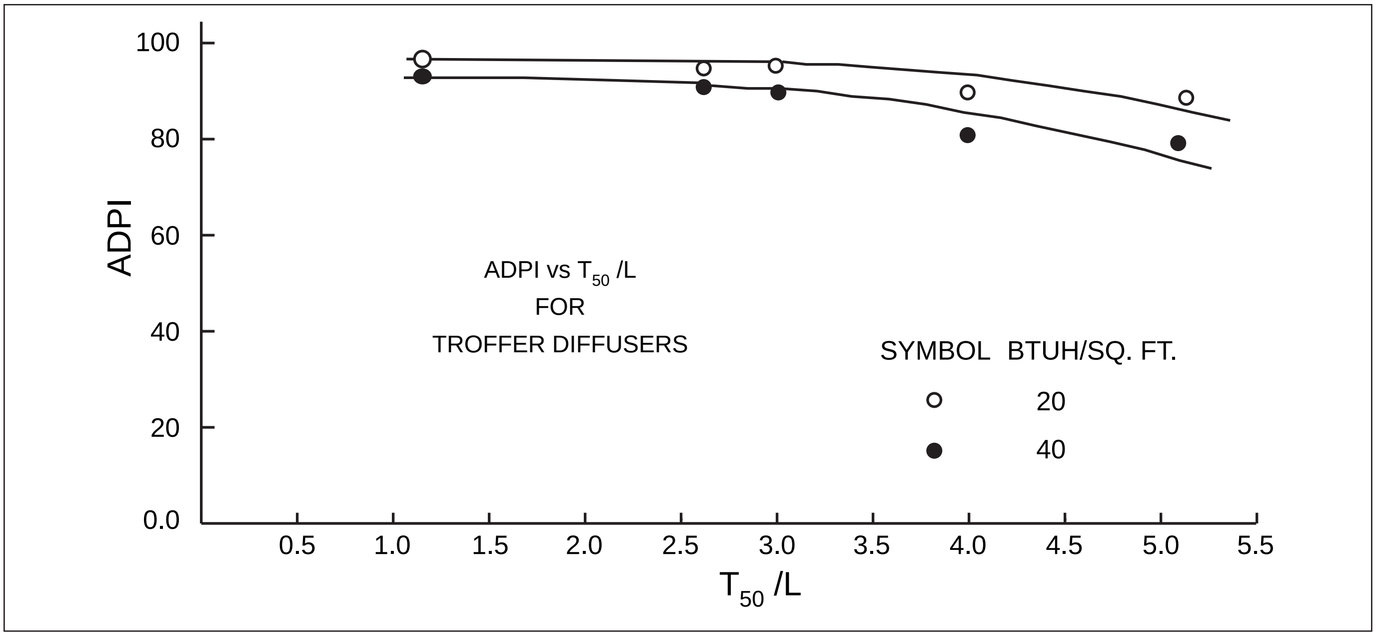


Fig. 23. Troffer Diffusers.

Comfort Index (APDI) vs. Throw Ratio (T_{50} / L)

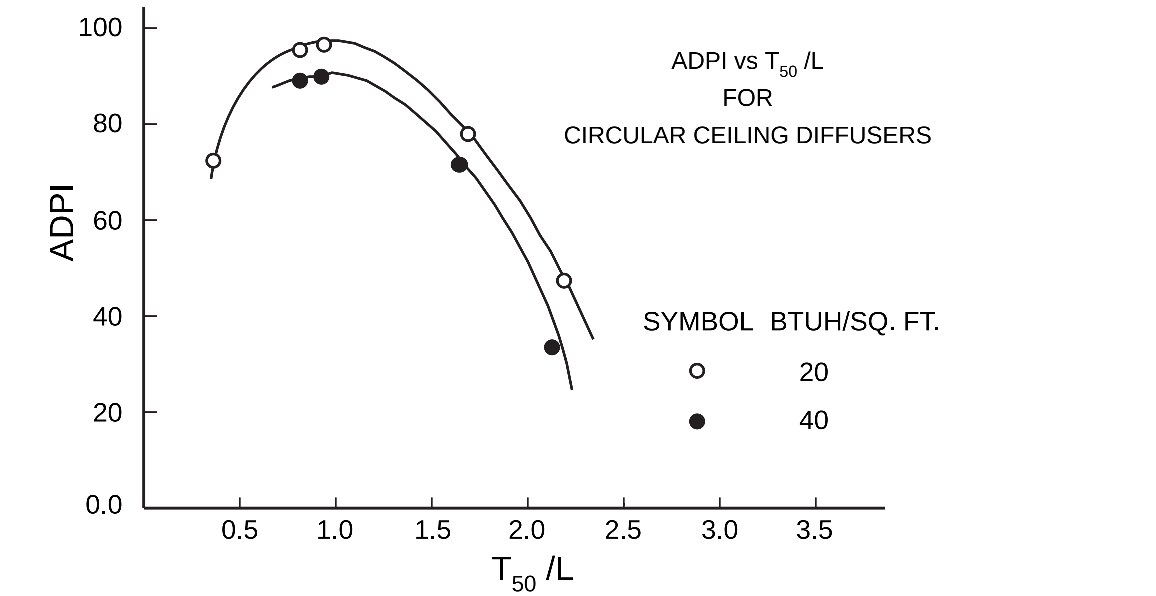


Fig. 24. Round Ceiling Diffusers.

Comfort Index (APDI) vs. Throw Ratio (T_{50} / L)

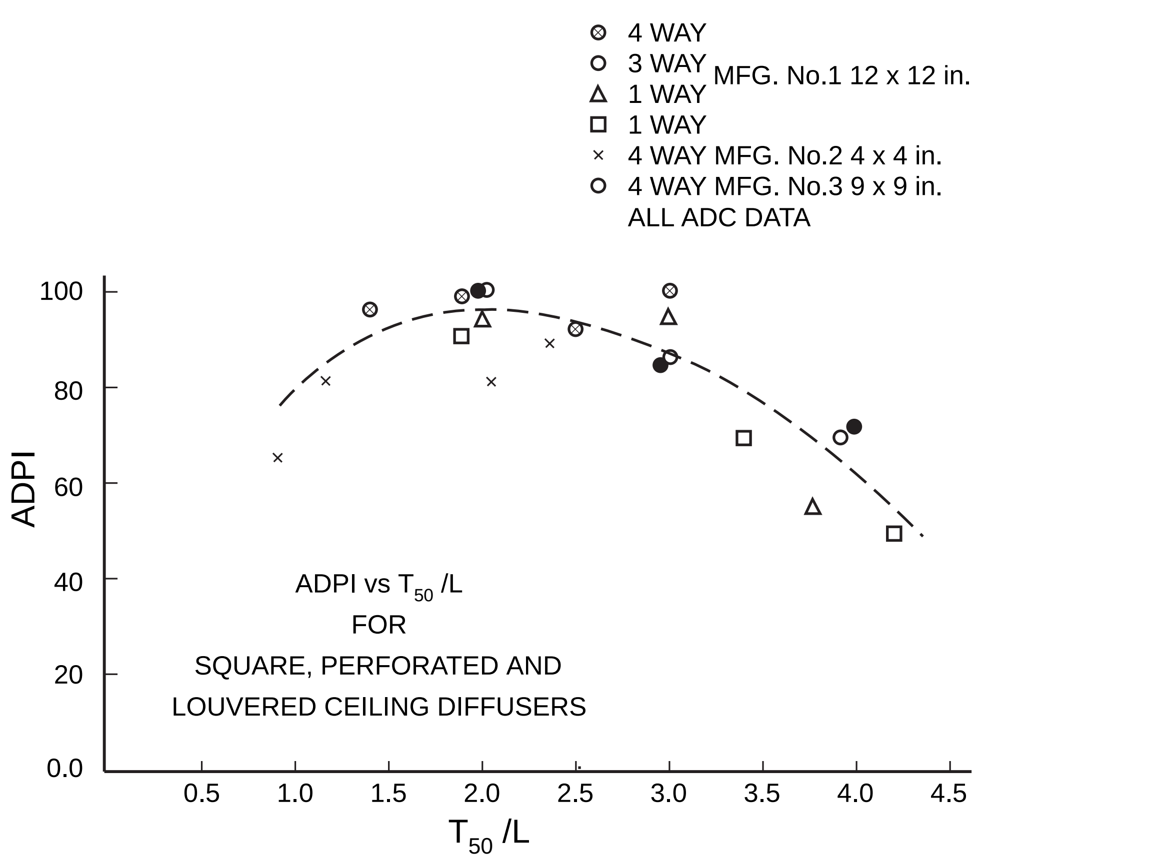


Fig. 25. Square, Perforated and Louvered Ceiling Diffusers. Room loads from 11 to 51 Btuh/sq. ft.

Table 11.

Room Length 'L' for Several Diffuser Types	
Diffuser Type	**Length 'L', Distance**
High Side Wall Grille	- to wall perpendicular to jet.
Circular Ceiling Diffuser	- to closest wall, or intersecting jet.
Sill Grille	- length of room in direction of throw.
Ceiling Slot Diffuser	- to wall or mid-plane between outlets.
Light Troffer Diffuser	- to mid-plane between outlets + distance ceiling to top of occupied zone.
Perforated Louvered Ceiling Diffusers	- to wall or mid-plane between outlets.

Table 12. ADPI Selection Guide

Terminal Device	Room Load Btuh/ft	T50/L	Maximum ADPI	For ADPI Greater Than	T50/L
High Side Wall Grilles	80	1.8	68	—	—
	60	1.8	72	70	1.5 - 2.2
	40	1.6	78	70	1.2 - 2.3
	20	1.5	85	80	1.0 - 1.9
Circular Ceiling Diffusers	80	0.8	76	70	0.7 - 1.3
	60	0.8	83	80	0.7 - 1.2
	40	0.8	88	80	0.5 - 1.5
	20	0.8	93	90	0.7 - 1.3
Sill Grille Straight Vanes	80	1.7	61	60	1.5 - 1.7
	60	1.7	72	70	1.4 - 1.7
	40	1.3	86	80	1.2 - 1.8
	20	0.9	95	90	0.8 - 1.3
Sill Grille Spread Vanes	80	0.7	94	90	0.8 - 1.5
	60	0.7	94	80	0.6 - 1.7
	40	0.7	94	—	—
	20	0.7	94	—	—
Ceiling Slot Diffuser	80	0.3*	85	80	0.3 - 0.7
	60	0.3*	88	80	0.3 - 0.8
	40	0.3*	91	80	0.3 - 1.1
	20	0.3*	92	80	0.3 - 1.5
Light Troffer Diffuser	60	2.5	86	80	<3.8
	40	1.0	92	90	<3.0
	20	1.0	95	90	<4.5
Perforated & Louvered Ceiling Diffusers**	11 -51	2.0	96	90	1.4 - 2.7
				80	1.0 - 3.4
Air Distributing Ceilings	80	—	57	—	—
	60	—	68	—	—
	40	—	78	—	—
	20	—	88	—	—

* T_{100}/ L (** Square face)

Pressure Measurement

Concepts of Pressure. Pressure is force per unit area. This may also be defined as energy per unit volume of fluid. There are three categories of pressure — Total Pressure, Static Pressure and Velocity Pressure. They are all associated with air handling. Unit of pressure is expressed in inches of water, designated **in. w.g.**

Static Pressure is the normal force per unit area at a small hole in the wall of a duct or other boundaries. It is a function of air density and degree of compression. It may be thought of as the pressure in a tire or in a balloon which extends in all directions.

Velocity Pressure is the force per unit area capable of causing an equivalent velocity in moving air. Velocity pressure is a function of air density and velocity. At standard air density, the relationship between velocity pressure and velocity is expressed in the following formula:

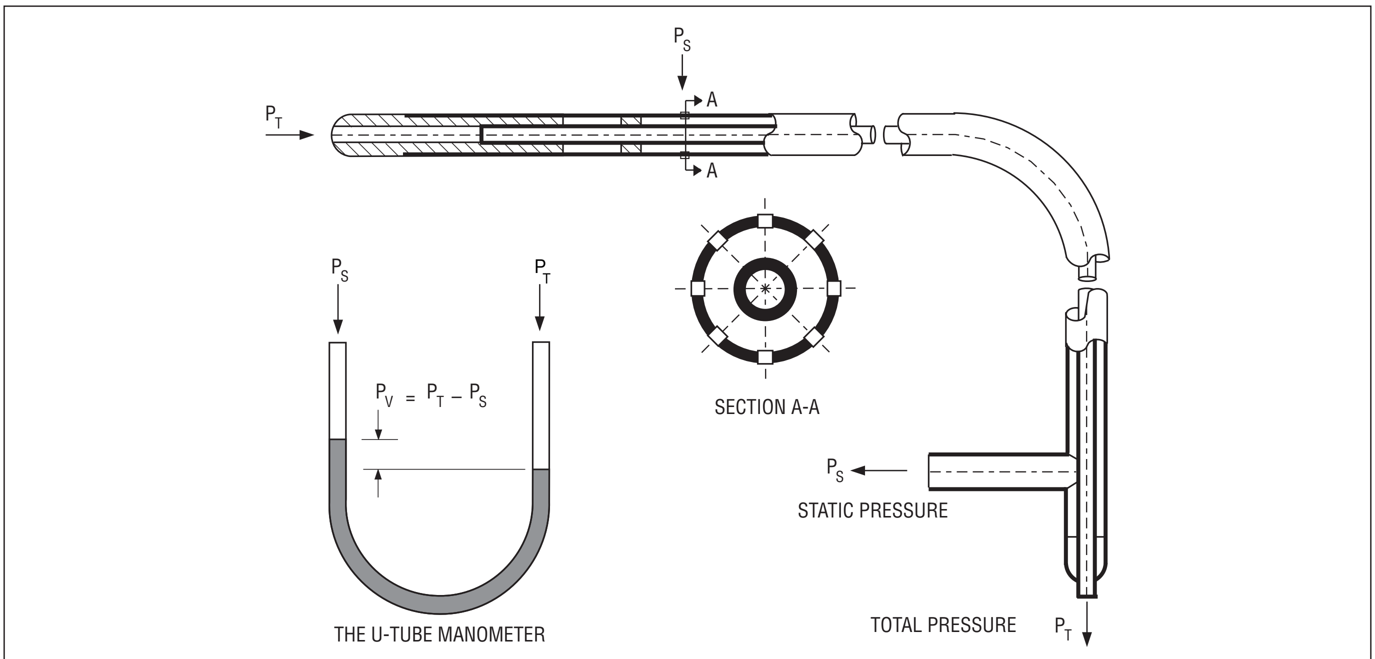
$$Pv = \left(\frac{V}{4005}\right)^2 \quad \text{or } V = 4005\sqrt{Pv}$$

Where: V = Air Velocity (FPM)
Pv = Velocity Pressure (in. w.g.)

Total Pressure, as its name implies, is the sum of static pressure and velocity pressure.

The Pitot Static Tube is an instrument used to measure pressure and velocities as illustrated below. It is constructed of two tubes. The inner, or impact tube, senses the total pressure as the impact opening faces upstream. The outer tube senses only the static pressure, which communicates with the airstream through small holes in its wall.

The U-Tube Manometer connects both parts of the Pitot Static Tube. The manometer functions as a subtracting device to give a reading of velocity pressure.



CONVERSION CHART for converting *VELOCITY PRESSURE* in inches of water to *VELOCITY* in feet per minute

Note: This chart is based upon standard air conditions of 70° Fahrenheit and 29.92 inches of mercury (barometric pressure), and assumes that the airflow is essentially non-compressible (under 10 inches of water pressure); as reflected by the following formula.

$$\text{Velocity (fpm)} = 4005 \sqrt{\text{Velocity Pressure in inches of water}}$$

VP	V	VP	V	VP	V	VP	V	VP	V	VP	V	VP	V	VP	V	VP	V	VP	V	VP	V
.001”	127	.062”	996	.123”	1404	.184”	1718	.245”	1982	.306”	2215	.367”	2426	.77”	3514	1.38”	4705	1.99”	5651	2.60”	6458
.002”	179	.063”	1004	.124”	1410	.185”	1723	.246”	1987	.307”	2219	.368”	2429	.78”	3537	1.39”	4722	2.00”	5664	2.61”	6470
.003”	219	.064”	1012	.125”	1416	.186”	1727	.247”	1991	.308”	2223	.369”	2433	.79”	3560	1.40”	4739	2.01”	5678	2.62”	6482
.004”	253	.065”	1020	.126”	1422	.187”	1732	.248”	1995	.309”	2226	.370”	2436	.80”	3582	1.41”	4756	2.02”	5692	2.63”	6495
.005”	283	.066”	1029	.127”	1427	.188”	1737	.249”	1999	.310”	2230	.371”	2439	.81”	3604	1.42”	4773	2.03”	5706	2.64”	6507
.006”	310	.067”	1037	.128”	1433	.189”	1741	.250”	2003	.311”	2233	.372”	2443	.82”	3625	1.43”	4790	2.04”	5720	2.65”	6519
.007”	335	.068”	1045	.129”	1439	.190”	1746	.251”	2007	.312”	2236	.373”	2445	.83”	3657	1.44”	4806	2.05”	5734	2.66”	6532
.008”	358	.069”	1052	.130”	1444	.191”	1750	.252”	2011	.313”	2239	.374”	2449	.84”	3669	1.45”	4823	2.06”	5748	2.67”	6544
.009”	380	.070”	1060	.131”	1449	.192”	1755	.253”	2015	.314”	2242	.375”	2453	.85”	3690	1.46”	4840	2.07”	5762	2.68”	6556
.010”	400	.071”	1067	.132”	1455	.193”	1759	.254”	2019	.315”	2245	.376”	2456	.86”	3709	1.47”	4856	2.08”	5776	2.69”	6569
.011”	420	.072”	1075	.133”	1461	.194”	1764	.255”	2023	.316”	2248	.377”	2459	.87”	3729	1.48”	4873	2.09”	5790	2.70”	6581
.012”	439	.073”	1082	.134”	1466	.195”	1768	.256”	2027	.317”	2251	.378”	2462	.88”	3758	1.49”	4889	2.10”	5804	2.71”	6593
.013”	457	.074”	1089	.135”	1471	.196”	1773	.257”	2031	.318”	2254	.379”	2466	.89”	3779	1.50”	4905	2.11”	5817	2.72”	6605
.014”	474	.075”	1097	.136”	1477	.197”	1777	.258”	2035	.319”	2257	.380”	2469	.90”	3800	1.51”	4921	2.12”	5831	2.73”	6617
.015”	491	.076”	1104	.137”	1482	.198”	1782	.259”	2039	.320”	2260	.381”	2472	.91”	3821	1.52”	4938	2.13”	5845	2.74”	6629
.016”	507	.077”	1111	.138”	1488	.199”	1787	.260”	2042	.321”	2264	.382”	2475	.92”	3842	1.53”	4954	2.14”	5859	2.75”	6641
.017”	522	.078”	1119	.139”	1493	.200”	1791	.261”	2046	.322”	2268	.383”	2479	.93”	3863	1.54”	4970	2.15”	5872	2.76”	6654
.018”	537	.079”	1125	.140”	1498	.201”	1795	.262”	2050	.323”	2272	.384”	2481	.94”	3884	1.55”	4986	2.16”	5886	2.77”	6666
.019”	552	.080”	1133	.141”	1504	.202”	1800	.263”	2054	.324”	2276	.385”	2485	.95”	3904	1.56”	5002	2.17”	5899	2.78”	6678
.020”	566	.081”	1140	.142”	1509	.203”	1804	.264”	2058	.325”	2280	.386”	2488	.96”	3924	1.57”	5018	2.18”	5913	2.79”	6690
.021”	580	.082”	1147	.143”	1515	.204”	1809	.265”	2062	.326”	2284	.387”	2491	.97”	3945	1.58”	5034	2.19”	5927	2.80”	6702
.022”	594	.083”	1154	.144”	1520	.205”	1813	.266”	2066	.327”	2289	.388”	2495	.98”	3965	1.59”	5050	2.20”	5940	2.81”	6714
.023”	607	.084”	1161	.145”	1525	.206”	1818	.267”	2070	.328”	2293	.389”	2499	.99”	3985	1.60”	5066	2.21”	5954	2.82”	6725
.024”	620	.085”	1167	.146”	1530	.207”	1822	.268”	2074	.329”	2297	.390”	2501	1.00”	4005	1.61”	5082	2.22”	5967	2.83”	6737
.025”	633	.086”	1175	.147”	1536	.208”	1827	.269”	2078	.330”	2301	.40”	2533	1.01”	4025	1.62”	5098	2.23”	5981	2.84”	6749
.026”	645	.087”	1181	.148”	1541	.209”	1831	.270”	2081	.331”	2304	.41”	2563	1.02”	4045	1.63”	5114	2.24”	5994	2.85”	6761
.027”	658	.088”	1188	.149”	1546	.210”	1835	.271”	2085	.332”	2308	.42”	2595	1.03”	4064	1.64”	5129	2.25”	6008	2.86”	6773
.028”	670	.089”	1193	.150”	1551	.211”	1839	.272”	2089	.333”	2311	.43”	2626	1.04”	4084	1.65”	5144	2.26”	6021	2.87”	6785
.029”	682	.090”	1201	.151”	1556	.212”	1844	.273”	2093	.334”	2315	.44”	2656	1.05”	4103	1.66”	5160	2.27”	6034	2.88”	6797
.030”	694	.091”	1208	.152”	1561	.213”	1848	.274”	2097	.335”	2318	.45”	2687	1.06”	4123	1.67”	5175	2.28”	6047	2.89”	6809
.031”	705	.092”	1215	.153”	1567	.214”	1853	.275”	2101	.336”	2322	.46”	2716	1.07”	4142	1.68”	5191	2.29”	6060	2.90”	6820
.032”	716	.093”	1222	.154”	1572	.215”	1857	.276”	2105	.337”	2325	.47”	2746	1.08”	4162	1.69”	5206	2.30”	6074	2.91”	6832
.033”	727	.094”	1228	.155”	1577	.216”	1862	.277”	2119	.338”	2329	.48”	2775	1.09”	4181	1.70”	5222	2.31”	6087	2.92”	6844
.034”	738	.095”	1234	.156”	1582	.217”	1866	.278”	2113	.339”	2332	.49”	2804	1.10”	4200	1.71”	5237	2.32”	6100	2.93”	6855
.035”	749	.096”	1241	.157”	1587	.218”	1870	.279”	2116	.340”	2335	.50”	2832	1.11”	4219	1.72”	5253	2.33”	6113	2.94”	6867
.036”	759	.097”	1247	.158”	1592	.219”	1875	.280”	2119	.341”	2338	.51”	2860	1.12”	4238	1.73”	5268	2.34”	6126	2.95”	6879
.037”	770	.098”	1254	.159”	1597	.220”	1879	.281”	2123	.342”	2342	.52”	2888	1.13”	4257	1.74”	5283	2.35”	6139	2.96”	6890
.038”	780	.099”	1260	.160”	1602	.221”	1883	.282”	2127	.343”	2345	.53”	2916	1.14”	4276	1.75”	5298	2.36”	6152	2.97”	6902
.039”	791	.100”	1266	.161”	1607	.222”	1887	.283”	2131	.344”	2349	.54”	2943	1.15”	4295	1.76”	5313	2.37”	6165	2.98”	6913
.040”	801	.101”	1273	.162”	1612	.223”	1892	.284”	2135	.345”	2352	.55”	2970	1.16”	4314	1.77”	5328	2.38”	6179	2.99”	6925
.041”	811	.102”	1279	.163”	1617	.224”	1896	.285”	2139	.346”	2356	.56”	2997	1.17”	4332	1.78”	5343	2.39”	6191	3.00”	6937
.042”	821	.103”	1285	.164”	1622	.225”	1900	.286”	2143	.347”	2360	.57”	3024	1.18”	4350	1.79”	5359	2.40”	6204	3.01”	6948
.043”	831	.104”	1292	.165”	1627	.226”	1905	.287”	2147	.348”	2363	.58”	3050	1.19”	4368	1.80”	5374	2.41”	6217	3.02”	6960
.044”	840	.105”	1298	.166”	1632	.227”	1909	.288”	2151	.349”	2366	.59”	3076	1.20”	4386	1.81”	5388	2.42”	6230	3.03”	6971
.045”	849	.106”	1304	.167”	1637	.228”	1913	.289”	2154	.350”	2369	.60”	3102	1.21”	4405	1.82”	5403	2.43”	6243	3.04”	6983
.046”	859	.107”	1310	.168”	1642	.229”	1917	.290”	2157	.351”	2372	.61”	3127	1.22”	4423	1.83”	5418	2.44”	6256	3.05”	6994
.047”	868	.108”	1316	.169”	1646	.230”	1921	.291”	2161	.352”	2376	.62”	3153	1.23”	4442	1.84”	5433	2.45”	6269	3.06”	7006
.048”	877	.109”	1322	.170”	1651	.231”	1925	.292”	2164	.353”	2379	.63”	3179	1.24”	4460	1.85”	5447	2.46”	6281	3.07”	7017
.049”	887	.110”	1328	.171”	1656	.232”	1929	.293”	2168	.354”	2383	.64”	3204	1.25”	4478	1.86”	5462	2.47”	6294	3.08”	7028
.050”	896	.111”	1334	.172”	1661	.233”	1933	.294”	2171	.355”	2386	.65”	3229	1.26”	4495	1.87”	5477	2.48”	6307	3.09”	7040
.051”	904	.112”	1340	.173”	1666	.234”	1937	.295”	2175	.356”	2389	.66”	3254	1.27”	4513	1.88”	5491	2.49”	6319	3.10”	7051
.052”	913	.113”	1346	.174”	1670	.235”	1941	.296”	2179	.357”	2393	.67”	3279	1.28”	4531	1.89”	5506	2.50”	6332	3.11”	7063
.053”	922	.114”	1352	.175”	1675	.236”	1945	.297”	2182	.358”	2396	.68”	3303	1.29”	4549	1.90”	5521	2.51”	6345	3.12”	7074
.054”	931	.115”	1358	.176”	1680	.237”	1950	.298”	2186	.359”	2400	.69”	3327	1.30”	4566	1.91”	5535	2.52”	6358	3.13”	7085
.055”	939	.116”	1364	.177”	1685	.238”	1954	.299”	2189	.360”	2403	.70”	3351	1.31”	4583	1.92”	5550	2.53”	6370	3.14”	7097
.056”	948	.117”	1370	.178”	1690	.239”	1958	.300”	2193	.361”	2406	.71”	3375	1.32”	4601	1.93”	5564	2.54”	6383	3.15”	7108
.057”	956	.118”	1376	.179”	1695	.240”	1962	.301”	2197	.362”	2410	.72”	3398	1.33”	4619	1.94”	5579	2.55”	6395	3.16”	7119
.058”	964	.119”	1382	.180”	1699	.241”	1966	.302”	2200	.363”	2413	.73”	3422	1.34”	4636	1.95”	5593	2.56”	6408	3.17”	7131
.059”	973	.120”	1387	.181”	1704	.242”	1970	.303”	2204	.364”	2416	.74”	3445	1.35”	4653	1.96”	5608	2.57”	6420	3.18”	7142
.060”	981	.121”	1393	.182”	1709	.243”	1974	.304”	2208	.365”	2420	.75”	3468	1.36”	4671	1.97”	5623	2.58”	6433	3.19”	7153
.061”	989	.122”	1399	.183”	1713	.244”	1978	.305”	2212	.366”	2423	.76”	3491	1.37”	4688	1.98”	5637	2.59”	6445	3.20”	7164

Definitions and Formulae

CFM = Cubic Feet per Minute
FPM = Feet per Minute (Velocity)
Ak = Area Factor Expressed in Square Feet
TP = Total Pressure Expressed in Inches of Water
SP = Static Pressure Expressed in Inches of Water
VP = Velocity Pressure Expressed in Inches of Water
VP = $(FPM \div 4005)^2$
ΔP = Differential Pressure
ΔP_S = Static Differential Pressure
ΔP_T = Total Differential Pressure

CFM = FPM x Ak
FPM = CFM ÷ Ak
VP = TP - SP
TP = SP + VP
SP = TP - VP
$\Delta P_T = TP_1 - TP_2$
$\Delta P_s = SP_1 - SP_2$

Thermal Linear Type Grille Expansion

ΔT Temperature Differential (°F)	Expansion (inches/ft.)		
	Aluminum	Steel	Copper
0	0	0	0
10	.00156	.00076	.00112
20	.00313	.00152	.00224
30	.00469	.00228	.00336
40	.00625	.00304	.00448
50	.00782	.00380	.00560
60	.00938	.00456	.00672
70	.01094	.00532	.00784
80	.01250	.00608	.00896
90	.01407	.00684	.01008
100	.01563	.00760	.01120

Measures of Force and Pressure

Dyne = force necessary to accelerate a 1 gram mass 1 centimeter per second squared = 0.000072 poundal.

Poundal = force necessary to accelerate a 1 pound mass 1 foot per second squared = 13,825.5 dynes = 0.138255 newtons.

Newton = force needed to accelerate a 1 kilogram mass 1 meter per second squared.

Pascal (pressure) = 1 newton per square meter = 0.020885 pound per square foot.

Atmosphere (air pressure at sea level) = 2,116.102; pounds per square foot = 14.6952; pounds per square inch = 1.0332; kilograms per square centimeter = 101,323 newtons per square meter.

Equivalent Measures of Pressure

1 lb. per square inch =
- 144 lbs. per sq. ft.
- 2.036 in. Mercury at 32°F.
- 2.311 ft. Water at 70°F.
- 27.74 in. Water at 70°F.

1 ounce per square inch =
- 1272 in. Mercury at 32°F.
- 1.733 in. Water at 70°F.

1 Atmosphere =
- 14.696 lbs. per sq. in.
- 2116.3 lbs. per sq. ft.
- 33.96 ft. Water at 70°F.
- 29.92 in. Mercury at 32°F.

1 inch Water at 70°F =
- .03609 lb. per sq. in.
- .5774 oz. per sq. in.
- 5.196 lbs. per sq. ft.

1 foot Water at 70°F =
- .433 lbs. per sq. in.
- 62.31 lbs. sq. ft.

1 inch Mercury at 32°F =
- .491 lbs. per sq. in.
- 7.86 oz. per sq. in.
- 1.136 ft. Water at 70°F.
- 13.63 in. Water at 70°F.

Sheet Metal Thickness (Inches) and Weight (Lbs./Sq. Ft.)

Gauge No.	Steel		Galvanized Steel		Aluminum	
	Thickness	Weight	Thickness	Weight	Thickness	Weight
3	.2391	10.000			.2294	3.23
4	.2242	9.375			.2043	2.88
5	.2092	8.750			.1819	2.56
6	.1943	8.125			.1620	2.29
7	.1793	7.500			.1443	2.04
8	.1644	6.875	.1681	7.031	.1285	1.81
9	.1495	6.250	.1532	6.406	.1144	1.61
10	.1345	5.625	.1382	5.781	.1019	1.44
11	.1196	5.000	.1233	5.156	.0907	1.28
12	.1046	4.375	.1084	4.531	.0808	1.14
13	.0897	3.750	.0934	3.906	.0720	1.02
14	.0747	3.125	.0785	3.281	.0641	.905
15	.0673	2.812	.0710	2.969	.0571	.806
16	.0598	2.500	.0635	2.656	.0508	.717
17	.0538	2.250	.0575	2.406	.0453	.639
18	.0478	2.000	.0516	2.156	.0403	.569
19	.0418	1.750	.0456	1.906	.0359	.507
20	.0359	1.500	.0396	1.656	.0320	.452
21	.0329	1.375	.0366	1.531	.0285	.402
22	.0299	1.250	.0336	1.406	.0254	.357
23	.0269	1.125	.0306	1.281	.0226	.319
24	.0239	1.000	.0276	1.156	.0201	.284
25	.0209	.875	.0247	1.031	.0179	.253
26	.0179	.750	.0217	.906	.0159	.224
27	.0164	.688	.0202	.844	.0142	.200
28	.0149	.625	.0187	.781	.0126	.178
29	.0135	.562	.0172	.719	.0113	.159
30	.0120	.500	.0157	.656	.0100	.141
31	.0105	.438	.0142	.594	.0089	.126
32	.0097	.406	.0134	.563	.0080	.113

† Steel – U.S. Standard (Revised)
Galvanized – Galvanized Gauge No.
Aluminum – American Gauge and Brown & Sharpe

Round Duct Area and Circumference

Dia. In Inches	Area Sq. Ft.	Circum. Inches
1	.00545	3.142
2	.0218	6.283
3	.0491	9.425
4	.0873	12.57
5	.1364	15.71
6	.1963	18.85
7	.2673	21.99
8	.3491	25.13
9	.4418	28.27
10	.5454	31.42
11	.6600	34.56
12	.7854	37.70
13	.9218	40.84
14	1.069	43.98
15	1.227	47.12
16	1.396	50.27
17	1.576	53.41
18	1.767	56.55
19	1.969	59.69
20	2.182	62.83
21	2.405	65.97
22	2.640	69.12
23	2.885	72.26
24	3.142	75.40
25	3.409	78.54
26	3.687	81.68
27	3.976	84.82
28	4.276	87.96
29	4.587	91.11
30	4.909	94.25
31	5.241	97.39
32	5.585	100.5
33	5.940	103.7
34	6.305	106.8
35	6.681	110.0
36	7.069	113.1
37	7.467	116.2
38	7.876	119.4
39	8.296	122.5
40	8.727	125.7
41	9.168	128.8
42	9.621	131.9
43	10.08	135.1
44	10.56	138.2
45	11.04	141.4
46	11.54	144.5
47	12.05	147.7
48	12.57	150.8
49	13.09	153.9
50	13.64	157.1

Common Fractions Reduced to Decimals

8ths	16ths	32ds	64ths		8ths	16ths	32ds	64ths		8ths	16ths	32ds	64ths	
			1	.015625				23	.359375				45	.703125
		1	2	.03125	3	6	12	24	.375			23	46	.71875
			3	.046875				25	.390625				47	.734375
	1	2	4	.0625			13	26	.40625	6	12	24	48	.75
			5	.078125				27	.421875				49	.765625
		3	6	.09375		7	14	28	.4375			25	50	.78125
			7	.109375				29	.453125				51	.796875
1	2	4	8	.125			15	30	.46875		13	26	52	.8125
			9	.140625				31	.484375				53	.828125
		5	10	.15625	4	8	16	32	.5			27	54	.84375
			11	.171875				33	.515625				55	.859375
	3	6	12	.1875			17	34	.53125	7	14	28	56	.875
			13	.203125				35	.546875				57	.890625
		7	14	.21875		9	18	36	.5625			29	58	.90625
			15	.234375				37	.578125				59	.921875
2	4	8	16	.25			19	38	.59375		15	30	60	.9375
			17	.265625				39	.609375				61	.953125
		9	18	.28125	5	10	20	40	.625			31	62	.96875
			19	.296875				41	.640625				63	.984375
	5	10	20	.3125			21	42	.65625	8	16	32	64	1.
			21	.328125				43	.671875					
		11	22	.34375		11	22	44	.6875					

Mathematical Formulae

To find the CIRCUMFERENCE of a:
Circle — Multiply the diameter by 3.14159265 (usually 3.1416).

To find the AREA of a:
Circle — Multiply the square of the diameter by .785398 (usually .7854).
Rectangle — Multiply the length of the base by the height.
Sphere (surface) — Multiply the square of the radius by 3.1416 and multiply by 4.
Square — Square the length of one side.
Trapezoid — Add the two parallel sides, multiply by the height and divide by 2.
Triangle — Multiply the base by the height and divide by 2.

To find the VOLUME of a:
Cone — Multiply the square of the radius of the base by 3.1416, multiply by the height, and divide by 3.
Cube — Cube the length of one edge.
Cylinder — Multiply the square of the radius of the base by 3.1416 and multiply by the height.
Pyramid — Multiply the area of the base by the height and divide by 3.
Rectangular Prism — Multiply the length by the width by the height.
Sphere — Multiply the cube of the radius by 3.1416, multiply by 4 and divide by 3.

Metric Guide Conversion Factors

Quantity	Imperial Unit	Metric Unit		From Imperial To Metric Multiply By:	From Metric To Imperial Multiply By:
Area	square foot	square meter	(m^2)	0.0929	10.764
	square inch	square millimeter	(mm^2)	645.16	.00155
Density	pounds per cubic foot	kilograms per cubic meter	(kg/M^3)	16.018	.0624
Energy	British thermal unit (BTU)	joule	(J)	1055.056	.000948
	kilowatt hour	megajoule	(MJ)	3.6	.2778
	watts per second	joule	(J)	1.0	1.0
	horsepower hour	megajoule	(MJ)	2.6845	.3725
Force	ounce force	newton	(N)	.278	3.597
	pound force	newton	(N)	4.4482	.2248
	kilogram force	newton	(N)	9.8067	.102
Heat	BTU per hour	watt	(W)	.2931	3.412
	BTU per pound	joules per kilogram	(J/kg)	2326.0	.00043
Length	inch	millimeter	(mm)	25.4	.0394
	foot	millimeter	(mm)	304.8	.00328
	foot	meter	(m)	.3048	3.2808
	yard	meter	(m)	.9144	1.0936
Mass (weight)	ounce (avoirdupois)	gram	(g)	28.350	.0353
	pound (avoirdupois)	kilogram	(kg)	.4536	2.2046
Power	horsepower	kilowatt	(kW)	.7457	1.341
	horsepower (boiler)	kilowatt	(kW)	9.8095	.1019
	foot pound - force per minute	watt	(W)	.0226	44.254
	ton of refrigeration	kilowatt	(kW)	3.517	.2843
Pressure	inch of water column	kilopascal	(kPa)	.2486	4.0219
	foot of water column	kilopascal	(kPa)	2.9837	.3352
	inch of mercury column	kilopascal	(kPa)	3.3741	.2964
	ounces per square inch	kilopascal	(kPa)	.4309	2.3206
	pounds per square inch	kilopascal	(kPa)	6.8948	.145
Temperature	Fahrenheit	Celsius	(°C)	5/9(°F-32)	(9/5°C)+32
Torque	ounce - force inch	millinewton-meter	(mN.m)	7.0616	.1416
	pound - force inch	newton-meter	(N.m)	.1130	8.8495
	pound - force foot	newton-meter	(N.m)	1.3558	.7376
Velocity	feet per second	meters per second	(m/s)	.3048	3.2808
	feet per minute	meters per second	(m/s)	.00508	196.85
	miles per hour	meters per second	(m/s)	.44704	2.2369
Volume (capacity)	cubic foot	liter	(l)	28.3168	.03531
	cubic inch	cubic centimeter	(cm^2)	16.3871	.06102
	cubic yard	cubic meter	(m^3)	.7646	1.308
	gallon (U.S.)	liter	(l)	3.785	.2642
	gallon (imperial)	liter	(l)	4.546	.2120
Volume (flow)	cubic feet per minute (cfm)	liters per second	(l/s)	.4719	2.119
	cubic feet per minute (cfm)	cubic meters per second	(m^3/s)	.0004719	2119.0
	cubic feet per hour (cfh)	milliliters per second	(ml/s)	7.8658	.127133
	gallons per minute (U.S.)	liters per second	(l/s)	.06309	15.850
	gallons per minute (imperial)	liters per second	(l/s)	0.7577	13.198