AIR DISTRIBUTION ENGINEERING

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Engineering Terms & Definitions

ADPI (Air Diffusion Performance Index): Statistically relates the space conditions of local temperatures and velocities to occupant's thermal comfort. The design goal in an office environment is to maintain high comfort levels by obtaining high ADPI values. In order to obtain high ADPI values, we must define a few terms for the selection procedure.

CFM: Cubic feet per minute - a measurement of volumetric air flow rate.

Clo: A unit used to express the thermal insulation provided by garments and clothing ensembles, where 1 clo = 0.155 $m^2 \cdot °C/W$ (0.88 $ft^2 \cdot h \cdot °F/Btu$).

Coanda Effect: In the 1930's the Romanian aerodynamicist Henri-Marie Coanda (1885-1972) observed that a stream of air (or other fluid) emerging from a nozzle tends to follow a nearby curved or flat surface, if the curvature of the surface or angle the surface makes with the stream is not too sharp.

Entrainment: To draw in and transport (as solid particles or gas) by the flow of a fluid.

FPM: Feet per minute - measure of air velocity.

Insulation, Clothing / Ensemble (Icl): The resistance to sensible heat transfer provided by a clothing ensemble. Expressed in clo-units.

L/s: Liters per second: A measurement of volumetric air flow rate.

LwDIS: Discharge sound power level.

LwRAD: Radiated sound power level.

Met: A unit used to describe the energy generated inside the body due to metabolic activity. It is defined as 58.2 W/m^2 (18.4 $Btu/h \cdot ft^2$) which is equal to the energy produced per unit surface area of an average person, seated at rest. The surface area of an average person is 1.8 m^2 (19 ft^2).

Metabolic Rate (M): Rate of energy production of the body by metabolism, which varies with activity. Expressed in met units in this standard.

NC: Noise criterion represented by NC curves that were developed to represent lines of equal hearing perception in all bands and at varying sound levels. Most air terminal products are currently specified and reported as a single number NC rating.

Occupied Zone: The region normally occupied by people within a space, generally considered to be between the floor and 1.8m (6ft) above the floor and more than 1.0 m (3.3ft) from outside walls/windows or fixed heating, ventilating or air conditioning equipment and 0.3m (1ft) from internal walls.

RC: Room Criteria represented by both a numerical value and a letter "quality" rating. The number represents the spectrum's speech interference level (SIL), and is obtained by taking the arithmetic average of the noise levels in the 500-, 1000- and 2000- Hz octave bands. The letter denotes the sound's "quality" as it might subjectively be described by an observer.

RH: The amount of water vapor actually in the air divided by the amount of water vapor the air can hold. Relative humidity is expressed as a percentage and can be computed in a variety of ways. One way is to divide the actual vapor pressure by the saturation vapor pressure and then multiply by 100 to convert to a percent.

SCIM: Abbreviation for Standard Cubic Inches per Minute with Standard conditions are defined as 14.7 psia and 60°F.

Sound Power Level (Lw or Pwl): The level, in dB as a ratio relative to some reference level, at which a source produces sound, usually given in octave bands. The equation is as follows:

Sound Power Level (PWL) = $10 \cdot \log10 \cdot (Wsource/Wref)$ dB where: Wref is 10^{-12} Watts and Wsource is sound power in Watts.

Sound Pressure Level (lp or SPL): The level of sound energy, measured in dB, at a specific location. The frequency range of the measurement or calculation must be indicated along with the sound level in dB. The equation is as follows :

Sound Pressure Level (SPL) = $10 \cdot \log10 \cdot (P^2/Pref^2)$ dB, where: Pref = $2 \cdot 10^{-5}$ Pa and P = sound pressure in Pa.

"WG: Inches of water gage - measure of pressure.

Air Distribution Basics

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Primary Jet

Extensive studies have shown that the air jet from free round openings, grilles, perforated panels, ceiling diffusers, and other outlets is defined by the "Throw," "Drop," and "Spread,".

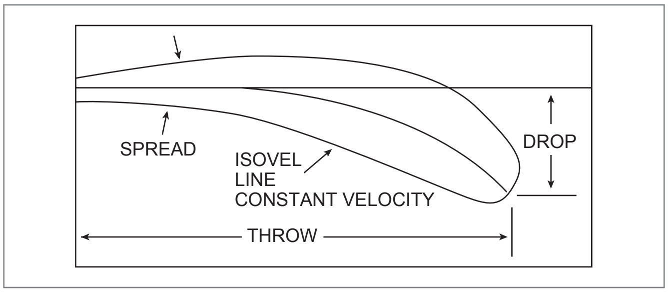
- **Throw** - The horizontal or vertical axial distance an airstream travels after leaving an air outlet before the maximum stream velocity is reduced to a specific terminal value (150, 100, 50 fpm).
- **Drop** - The vertical distance that the lower edge of a horizontally projected airstream drops between the outlet and the end of its throw.
- **Spread** - The divergence of the airstream in a horizontal or vertical plane after it leaves the outlet.

The primary air jet is defined as the air delivered to a room through the supply duct plus the entrained room air that lies within an envelope of arbitrary velocity, usually taken as some value between 50 and 150 fpm. The jet discharged from the free opening has four zones of expansion. The centerline velocity of this primary jet in any zone is related to the initial velocity (See Figure: "Throw & Drop of Free Air Jet"). Regardless of the type of outlet, the airstream will tend to assume a circular shape in free space, provided the jet does not impinge on or flow along a boundary or surface.

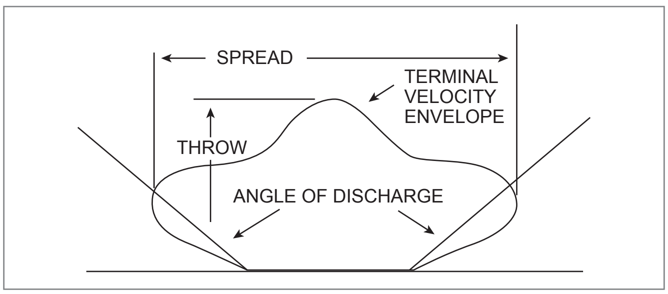
In many cases, isothermal jet performance can be analyzed to determine the following.

- Angle of Divergence or Spread
- Velocity Patterns Along the Jet Axis
- Velocity Profile at Any Cross-section in the Zone of Engineering Significance
- Entrainment Ratios in the Same Zone

THROW & DROP OF FREE AIR JET



SPREAD OF FREE AIR JET

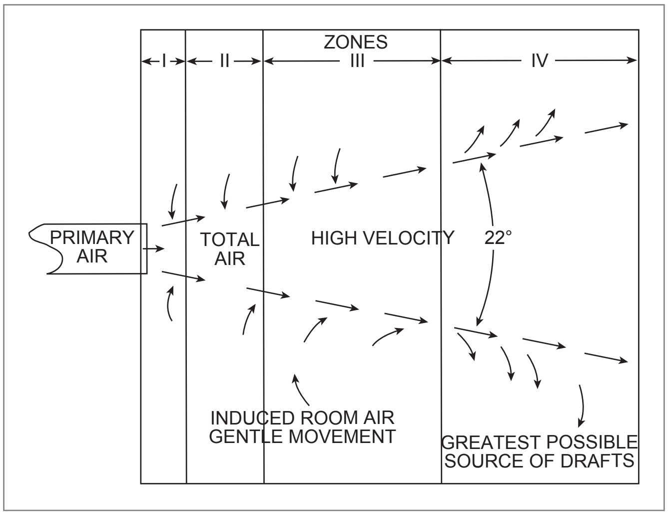


The angle of divergence is usually well defined near the outlet face with the boundary contours becoming billowy and poorly defined as the velocity nears the terminal velocity of 100-50 fpm (See Figure: "Spread of Free Air Jet"). External forces, local eddies, and surges easily influence the boundary.

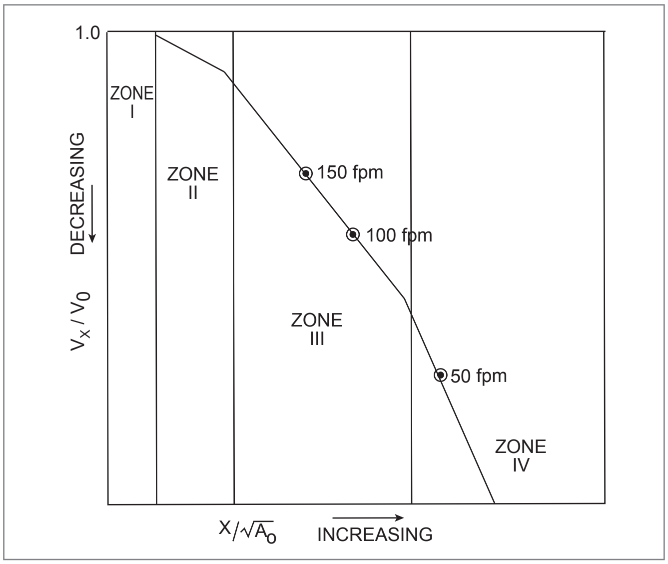
The zones of expansion for an isothermal jet can be described as follows. (See Figures: "Expansion Zones & Centerline Velocities of Axial & Radial Jets)

Zone 1 - A short zone, extending about four duct diameters from the outlet face (or vena contract for orifice discharge) in which the maximum velocity of the airstream, or the centerline velocity, remains practically unchanged.

EXPANSION ZONES



CENTERLINE VELOCITIES OF AXIAL & RADIAL JETS



V_o - Outlet Velocity *$\sqrt{A_o}$ - Outlet Area*
X - Throw Distance *V_x - Terminal Velocity*

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Zone 2 - A transition zone, extending about eight duct diameters for round outlets or for rectangular outlets of small aspect ratio, over which maximum velocities vary inversely as the square root of the distance from the outlet. For rectangular outlets of large aspect ratio, this zone is elongated and extends from about four widths to a distance approximately equal to the width multiplied by four times the aspect ratio.

Zone 3 - A long zone of major engineering importance in which the maximum velocities vary inversely as the distance from the outlet. This zone is often called the zone of fully established turbulent flow and may be 25-100 duct diameters long, depending on the shape and area of the outlet, the initial velocity and the dimensions of the space into which the outlet discharges.

Zone 4 - A terminal zone, in the case of confined spaces, in which the maximum velocity decreases at an increasing rate, or in the case of large spaces free from wall effects, the maximum velocity decreases rapidly in a few duct diameters to a velocity below 50 fpm. Measured angles of divergence usually range from 20° to 24° with an average value of 22°.

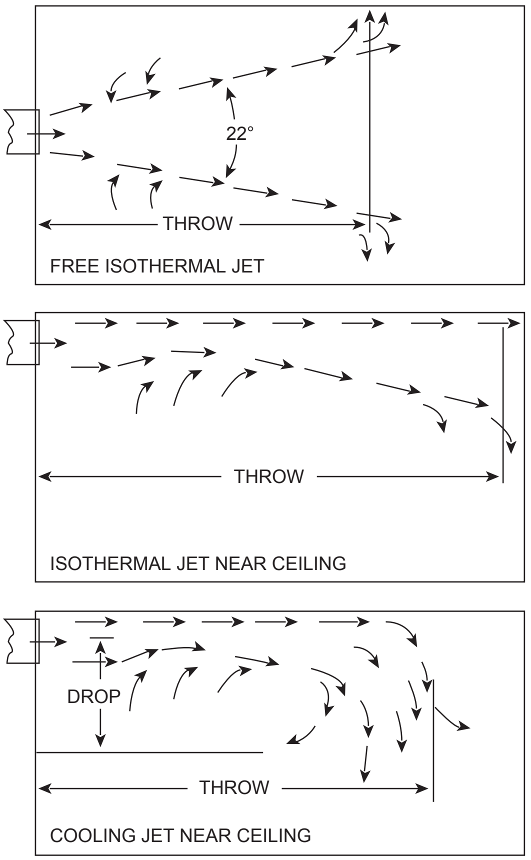
Entrainment
As the primary jet discharges or exits from the outlet, it begins to induce the room air into the jet stream, exchanging the momentum of the jet to increase momentum in the room air. The entrainment increases the volumetric flow at a given section of the jet and increases the width of the jet causing spread.

Jet Interference
Jets discharging near obstructions, walls, ceilings, or near other jets may be influenced in several ways. Jets flowing parallel to a ceiling or wall will attach to the surface, so the maximum velocity will exist near the surface. Entrainment takes place along the surface of the half cone, or the free surface of the jet, which is known as the “Coanda” effect. The Coanda effect, or surface effect, creates a low-pressure region between the mass airflow and the ceiling or wall. The resulting higher pressure on the room side holds the airstream to the surface, which increases throw and decreases drop (See Figure: “Jet Patterns”). As catalog data includes this effect, throw for most ceiling diffusers should be reduced by 30% if no ceiling is present.

The interior layout, local sources of heat gain or loss, and outlet performance factors, all influence the final selection of an air distribution system. These all must be considered in selecting the type and location of the supply outlets. In an attempt to control room air motion, some basic principles of air delivery to a space must be understood.

Primary Air
Primary air is the starting point when laying out or investigating room air motion. By definition, primary air is the mixture of air supplied to the outlet and induced room air within an envelope of velocities greater than 150 fpm. The primary pattern can be completely defined by high velocity envelopes taken through two perpendicular planes. Data obtained isothermally (no temperature difference between the supply and room air) down to a velocity of 150 fpm apply equally well for heating and cooling.

JET PATTERNS



Total Air
Total air is the mixture of primary and entrained room air, which is under the influence of the outlet conditions. Normally, total air has a relatively high velocity, but it does not have a sharply defined lower limit. Even though the total air follows the general pattern indicated by the primary air, its spread and travel may not be in proportion to that of the primary air. Other factors, such as ceiling height, obstructions, and internal and external loads disturb the orderly course of the airstream.

Temperature differences between total air and room air produce a buoyancy effect, which causes cool total air to drop and warm total air to rise. The most complete mixing of total air and room air occurs during isothermal conditions. Consequently, the type of outlet and location reduce buoyancy effects and increase the travel of total air (restricts drop) during cooling, when cool air is induced and mixed rapidly with the primary air. This also reduces stratification during heating, when warm air is induced and mixed rapidly with the primary air. In addition to the outlet type and location, buoyancy effects are dependent on the temperature differential between supply air and room air. Total air has a tendency to “scrub” surfaces near diffusers or

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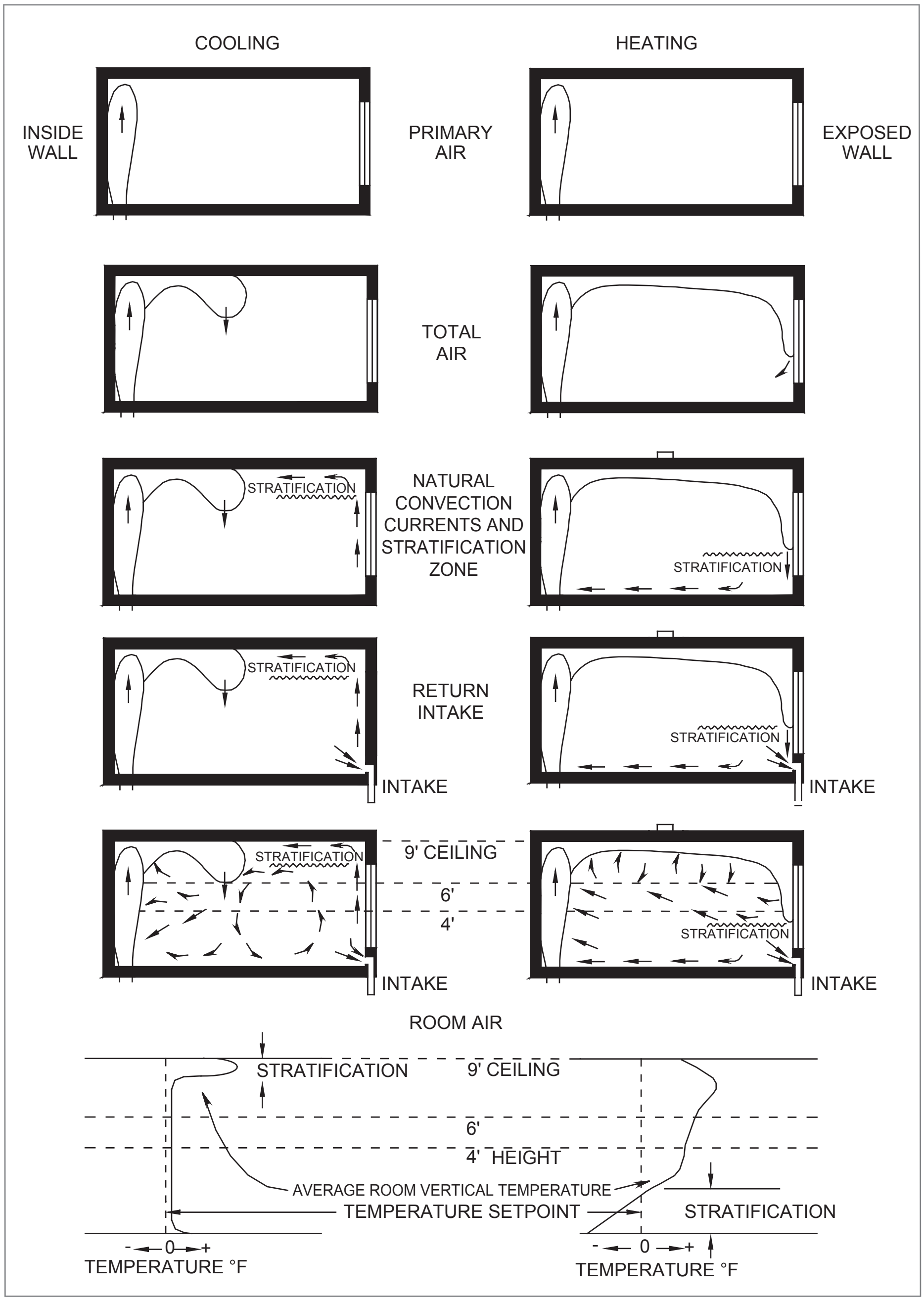
air outlets. A near perfect situation can be envisioned where total air covers all of the walls and ceiling with a thin film. The occupied space would then be completely enclosed within an envelope of conditioned air. From the smudging standpoint, effective air distribution usually requires that the surfaces be used, no matter how desirable it might be to keep the total air off the surfaces.

Since the total air within a confined space is affected by factors other than the outlet conditions, it is not subject to complete analytical treatment. However, it is possible to estimate the total air characteristics for cooling and heating within a free space, which is a condition where the airstream does not come in contact with confining walls or ceiling until the airstream has been reduced to a low velocity.

Natural Convection Currents

Currents are created by buoyancy effects, which are caused by differences in temperature between room air and the air in contact with a warm or cold surface. Air in contact with a warm surface will rise; the air in contact with a cold surface will fall. These currents are caused not only by the windows and walls, but also by internal loads such as people, lights, machines, etc. In most cases, natural convection currents will not only affect the room air motion, but they can also play a major role in the comfort of a space. For example, convection currents falling down at the outside wall during the heating season can produce a feeling of coolness.

AIR MOTION CHARACTERISTICS OF OUTLETS IN OR NEAR FLOOR WITH NON-SPREADING VERTICAL JET



Stratification layers, as shown to the left, actually exist in practice and in many tests. A similar situation often occurs in practice, which is identified by a region where a layer of smoke, for visualization, will "hang" for some time. Whether a stagnant layer actually exists is not important, but the concept of a stagnant layer and zone leads to a better understanding of air distribution. It may be noted that the natural convection currents form a stagnant zone between the stagnant layer and the ceiling during cooling and between the stagnant layer and the floor during heating.

Studies have shown that with well mixed air supply systems (typical ceiling supply and return systems), with less than 1.5 cfm/ft^2, the primary driver of room air motion is local convection sources, not the diffuser induced air currents. Room air motion is primarily dependent on room load, not diffuser air supply rate. As a result, room air motion levels with VAV (Variable Air Volume) systems at part load are comparable to those of CAV (Constant Air Volume) systems at the same load condition.

The temperature gradient curves, as shown left, emphasize how some of the factors discussed are interrelated and how they affect the space temperature distribution. Where the air motion is uniform (between the total air and stagnant layer), the temperatures are approximately equal and uniform (as indicated by the almost vertical portion of the gradient). As the stagnant

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layer is crossed, the temperatures in the stagnant zone vary considerably. The gradients in the stagnant zones show that the air is stratified in layers of increasing temperatures with an increase in space height.

The stagnant zone depends on natural currents, magnitude of the heating or cooling load, space construction and volume, and the area of exposure of the load, as in the case of internal loading. Consequently, the magnitude of the temperature variations between levels will be smaller in mild climates than in severe climates, as well as in spaces having exposed walls with greater resistance to heat flow with minimum internal loads. Hence, with no loading, the temperature gradient curves would be vertical, indicating that all of the air temperatures in the conditioned space are equal to the control temperature.

Return Intake
Return intakes typically affect only the air motion within its immediate vicinity. Even the natural convection currents may possess enough energy to overcome the draw of the intake. This does not mean that the return location is not important, but only that it has little effect on the room air motion.

Room Air
The room air motion diagram is completed when the remaining room air is shown to drift back toward the primary air and total air envelopes. The highest air motion in the space is in and near the primary and total air, while the most uniform air motion is between the total air and the stagnant layer. The lowest air motion is in the stagnant zone.

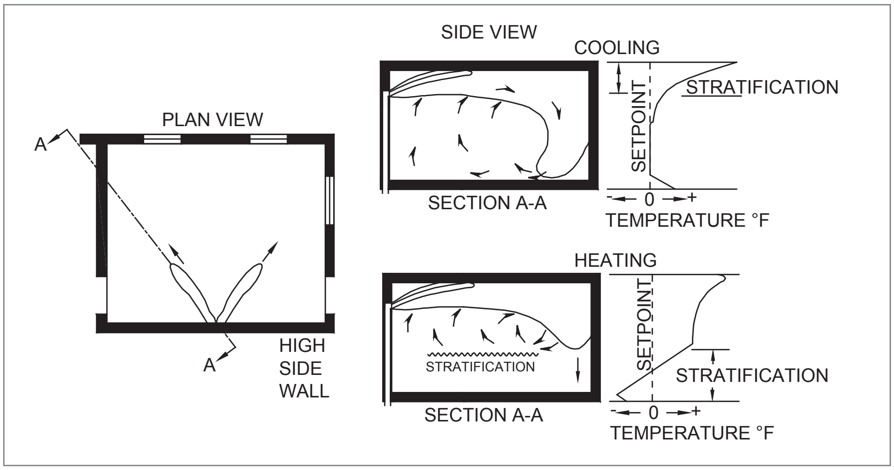
Performance of Supply Outlets
Construction features of supply outlets vary significantly to accommodate several different applications. The following are examples of typical applications and the performance criteria required from the supply outlets as defined by Straub et. al.

Group A: Horizontal Discharge Near Ceiling - This group includes high sidewall grilles, sidewall diffusers, ceiling diffusers and linear ceiling diffusers. The figure “Air Motion Characteristics of High Sidewall, Group A Grilles” shows a primary air envelope or isovel with a two-jet pattern for a high sidewall ceiling outlet. Outlet vane settings may cause a discharge in one, two, or three jets; however, the overall effect in each is the same. During cooling, the total air drops into the occupied zone at a distance from the outlet, which depends on air quantity, supply velocity, temperature difference between supply and room air, deflection setting, ceiling effect and type, and amount of loading in the space. Also revealed, for cooling, is a potential overthrow condition, which causes the total air to drop along the opposite wall and flow slowly for some distance across the floor. For heating, under the same airflow capacity, the total air does not descend along the opposite wall as far.

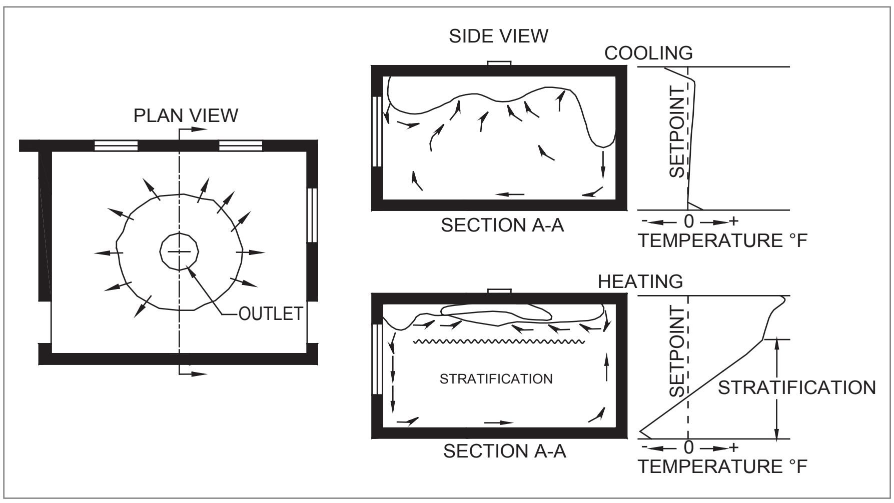
Through careful selection, by limiting air temperature differential between supply and room air and by maintaining airflow rates high enough to ensure mixing through induction, adequate room air motion can be obtained while minimizing stratification in the occupied zone.

The figure “Air Motion Characteristics of Ceiling, Group A Diffusers” shows a ceiling outlet application in which the total air movement is counteracted by the rising natural convection currents along a heated perimeter wall, compared to the descent along a cooler interior wall on the opposite side. These outlets discharge horizontally near the ceiling with significant spread, allowing the warmest air in the space to mix with the cool primary air far above the occupied zone. Therefore, these outlets are capable of handling relatively large quantities of air at large temperature differentials while providing acceptable levels of room air motion and temperature uniformity in the occupied zone.

AIR MOTION CHARACTERISTICS OF HIGH SIDEWALL, GROUP A GRILLES



AIR MOTION CHARACTERISTICS OF CEILING, GROUP A DIFFUSERS



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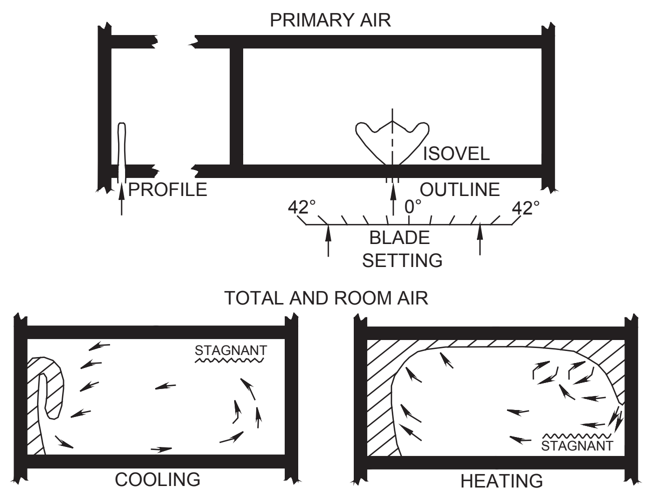
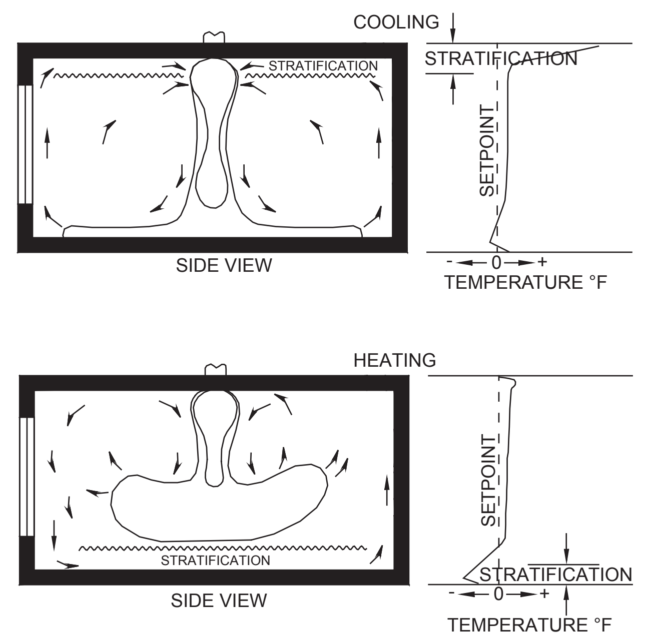
Group C: **Vertical Discharge from or Near Floor** - This group includes floor diffusers, low sidewall units, and linear grilles in the floor or windowsill. The figure “Air Motion Characteristics of Floor Mounted Outlets with Vertical Discharge, Group C Grilles” shows the similar type of vertical discharge from the floor region, however includes a wide spreading jet. This diffusing action of the outlet makes it difficult to project the cool air, but it also provides a greater area for induction of room air. Additionally, it shows that a stagnant zone occurs outside the total air zone and above the occupied zone. Below, the temperature uniformity and room air motion is acceptable. This may not apply to larger zones; however, a distance of approximately 15’ - 20’ between the drop region and the exposed wall is a conservative design value. In heating applications, the stagnant region is small due to the air being entrained in the immediate vicinity of the outlet, very close to the stagnant area. This is beneficial during heating because the induced air comes from the lower regions of the room. The result is greater temperature uniformity in the space when compared to other types of supply outlets during heating.

AIR MOTION CHARACTERISTICS OF FLOOR MOUNTED OUTLETS WITH VERTICAL DISCHARGE: GROUP C GRILLES



OUTLET IN OR NEAR FLOOR, SPREADING VERTICAL JET

AIR MOTION CHARACTERISTICS OF CEILING OUTLETS WITH VERTICAL DISCHARGE: GROUP E DIFFUSERS



Group E: **Vertical Discharge from or Near Ceiling** - This includes ceiling diffusers as well as linear diffusers and grilles designed for vertical downward air projection. The figure “Air Motion Characteristics of Ceiling Outlets with Vertical Discharge, Group E Diffusers” shows that in cooling, total air is projected vertically to the floor and continues along throughout the floor, leaving the stagnant area near the ceiling. During heating, the total airflow reaches the floor and folds back toward the ceiling. If the projected air does not reach the floor, a stagnant zone results.

The “Laminar Flow CFD Model” on the next page shows vertical discharge from the ceiling of the non-turbulent laminar flow type diffusers. Laminar flow occurs when the discharge velocities leaving the diffuser are uniformly moving in the same direction at relatively low speeds. A large mass volume of low velocity air projects vertically to the floor without much change in overall air speed. The result is not a mixing type diffuser, but a laminar device that experiences very little entrainment from the space. This is especially important for clean air applications in such areas as hospital operating rooms, clean rooms, and critical work areas. See the Krueger Model 5000 Series diffuser.

The “TAD CFD Model” on the next page shows air discharge from the ceiling in a uniform, low velocity radial isovel. This is not considered to be a laminar device in that the air velocities are not moving in the same direction. Hence, this is a non-entraining radial displacement diffuser. The concept is to displace particulates away in the space in all directions without mixing, which is important in critical environments such as pharmaceutical laboratories, chemistry laboratories, animal holding rooms, isolation wards, clean rooms, etc. - where particulate control and air change effectiveness are high priority. See the Krueger Model TAD and RadiaFlo radial displacement diffusers.

Pressure Measurements

Engineers require pressure data for air outlets to properly size the air delivery system within a building. To size and balance air delivery systems, one must have an understanding of three types of pressure measurement.

- **Static Pressure (Sp)** - The outward force of air within a duct, measured in inches of water column.
- **Velocity Pressure (Vp)** - The forward moving force of air within a duct, measured in inches of water column.

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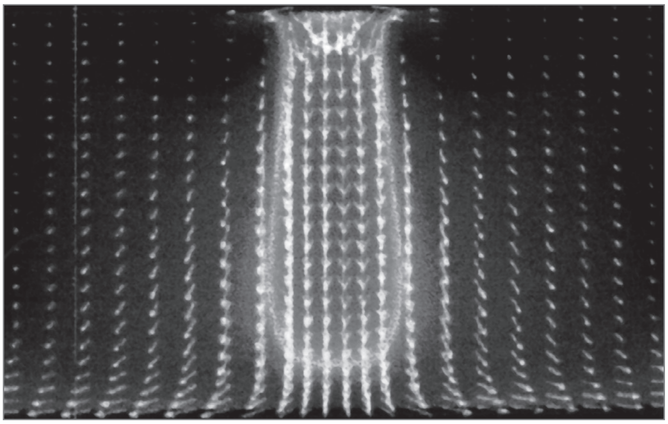
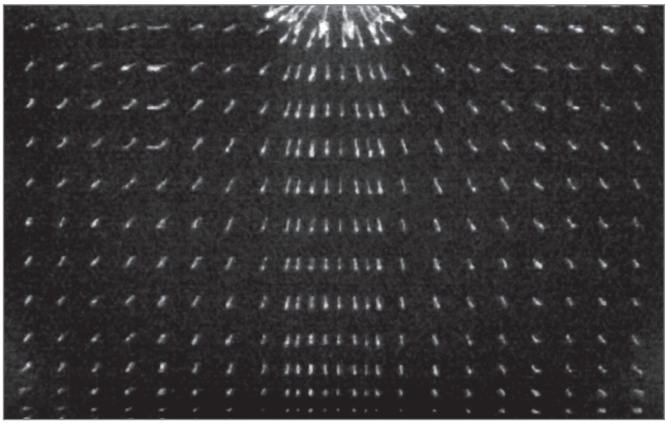
- **Total Pressure (Tp)** - The sum of the velocity and static pressures, measured in inches of water column. Tp = Vp + Sp.

LAMINAR FLOW CFD MODEL



TAD CFD MODEL



Outlet total pressure, referenced in the catalog performance tables, is the amount of total pressure required by the air source to push a given amount of flow (cfm) through a desired supply outlet. Total and static pressures are measured in the supply duct at a minimum of 1.5 duct diameters upstream of the outlet as per ASHRAE Standard 70.

The procedure requires a traverse to assure that static regain is not affecting the data. In many cases, data must be taken as far as 10’ upstream from large inlet diffusers that may tend to ‘extract’ air. Alternatively, the ASHRAE procedure allows for measuring total pressure in a large plenum (with a short aerodynamic duct attached to the diffuser), and calculating pressure drop. This method is preferred for many types of diffusers. Return grille and duct pressures must be conducted with the air inlet located on a large flat surface, if that is the intended application. This can have a significant effect on reported pressure drop. Also, it is not proper to report the difference between a duct with and without the exhaust grille as the return pressure. This results in very low reported pressure drops.

Continuous Duct Applications

The ASHRAE handbook does not give advice on the application of a series of diffusers/grilles along a section of duct. Experience has shown, however, that a successful design requires a constant diameter duct, rather than a constant velocity/stepped size duct, for the most uniform air delivery. As velocity pressure decreases, or as air is lost through diffusers/grilles, the static pressure drop will decrease as well, due to duct friction, and actual static will increase slightly. In a good design, minimum static pressure will be reached at a point 2/3 of the way down the duct.

Several rules are recommended for constant delivery along the entire duct run.

- The larger the supply duct, the better. Minimum diameter should be a function of desired length, not flow rate. Duct velocities should never exceed 1000 fpm.
- A constant duct diameter is required to achieve the required static regain. Stepping the duct to maintain duct velocity is not only counter-productive, but also results in less than optimal supply distribution.
- Required inlet pressures are very low. In most cases, less than 0.2 in. Sp is required (when properly ducted). A major source of noise has been found from pressure reducing/balancing dampers required at the supply inlet to the large duct.
- Friction in the supply duct is beneficial. Interior fiberglass lining’s rough surface allows smaller duct diameters than a smooth sheet metal duct.
- The air has to be turned to get the discharge perpendicular to the duct run. If not turned properly, air tends to pile up in the room at the end opposite the supply inlet.
- As there is some temperature rise on long runs, it may be beneficial to size the duct slightly smaller than optimal, which will result in more airflow at the far end than at the supply, which will allow a continuous btuh/ft, rather than cfm/ft supply flow.

For duct mounted spiral duct grille applications, this has several implications. When properly sized (ie: large enough), a duct mounted grille’s scoop damper may not be needed. As a result of static regain, undampered airflow may be essentially constant, but discharge throw will be at a slight angle. The vertical blades in the grille can be adjusted after installation to account for this, but there will be some increase in noise level from this adjustment. We have estimated this increase to be no more than 5 NC, and we estimate a slight (10%) increase in total pressure. (As the duct velocities should be low, velocity pressure will be low, and most of the increase will be in terms of static pressure.) If duct velocities are higher, or duct pressures vary (often the result of stepping down the duct size), some dampering will likely be required, further increasing the sound produced. The Krueger Model DMG Series performance data is therefore based on the Krueger Model 880 Grille Series performance data, with the NC increased by 5 and the total pressure increased by 10%. This will be a conservative estimate for most applications.

Occupant Comfort

Many applications for the spiral duct mounted grille will be near the ceiling. If within one diameter of a horizontal surface, the air pattern will likely attach itself to that surface, and the grille will exhibit a characteristic grille throw pattern (which is shown in the Krueger Model DMG Series of Duct Mounted Grilles performance data). When the duct is further than one duct diameter from an adjacent surface, the grille will have a 'free jet' pattern. Testing shows that this will result in a 30% decrease in throw.

When duct mounted linear diffusers are used in a length of continuous duct, similar rules apply. Ducts should not be stepped down, and if sufficiently large, air discharged will be at a relatively continuous rate. There will be some sideways deflection in the direction of airflow, except at the end of the run.

In conclusion, maintaining a constant diameter (as large as practical) will result in the optimum application of grilles and diffusers along the run. Slight increases in grille sound levels and static pressure will result from optimizing discharge angles. The data provided for the Krueger Model DMG Series of Duct Mounted Grilles will probably be conservative, if the duct is large enough, and doesn't step down. (In many cases, the cost of the step-down transition is more than the cost of 20 feet of straight duct. Typically, a constant diameter run of spiral is the same cost as one that is stepped.)

For duct-mounted linear diffusers, again, a constant diameter duct is an optimal design. There will be some directionality to the jets that cannot be avoided. (Effective turning vanes have been engineered in the past, but never employed because of cost). When properly sized, scoops and balancing dampers may not be required.

OCCUPANT COMFORT

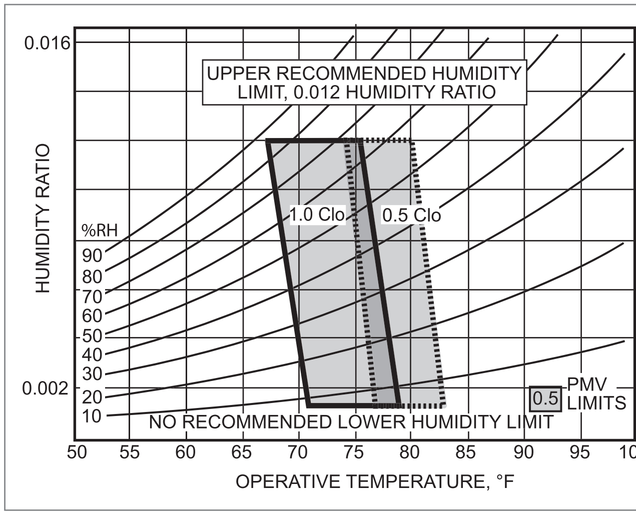
Room air distribution plays a very important role in providing comfort to the occupants of a building. Before understanding air distribution, one must understand what comfort means to occupants in a space.

In typical commercial office environments, occupants are generally considered thermally comfortable when body heat loss equals the body's metabolic heat production without conscious involvement of the body's thermo-regulatory mechanisms, such as sweating or shivering. Body heat loss to the environment can be attributed to radiation, conduction, convection, or evaporation. In an attempt to provide acceptable comfort levels for occupants in a space, the designer or engineer is challenged by the variables that influence the heat transfer of the human body:

- Dry Bulb and Mean Radiant Temperatures
- Relative Humidity
- Air Movement
- Activity Level of the Occupants (Met Units)
- Insulating Value of Occupant Clothing

A thermal comfort program based on the ASHRAE Thermal Comfort Standard (Standard 55-2004) is available on Krueger's website, at www.krueger-hvac.com. The standard identifies

STANDARD 55-2004 COMFORT ENVELOPE



thermal comfort based on clothing levels, a revised upper humidity level, and no minimum humidity level, compared to previous versions of the standard.

Proper selection and location of air distribution devices (grilles, registers, diffusers) is key in the control of occupant comfort levels. Although other entities of the HVAC system are important, it is the space air distribution that directly impacts the average temperature and air movement within the occupied zone. The ability to control the air distribution will result in overall acceptable thermal comfort levels for all occupants within the space.

Air Diffusion Performance Index (ADPI)

The relationship of room air motion and temperature as it relates to occupant comfort has been studied for years and continues today. Generally, occupant comfort can be achieved when the following conditions are maintained:

- Air Temperature Maintained Between 73°F - 77°F
- Relative Humidity Maintained Between 25% - 60%
- Maximum Air Motion within Occupied Zone (6" to 6' Vertical, within 12" of Walls) < 50 fpm
- Maximum Vertical Temperature Variation No Greater than 5°F from 6" to 6' Level

Researchers evaluate occupant comfort primarily on the effects of local air motion (velocity), local temperature, and ambient temperature relative to the feeling of warmth or coolness. Criteria for the "feeling of comfort at the neck" vs. varying room temperatures vs. air velocities are shown in the graph on the next page. This graph represents what is known as "effective draft temperature," which is expressed in the following equation:

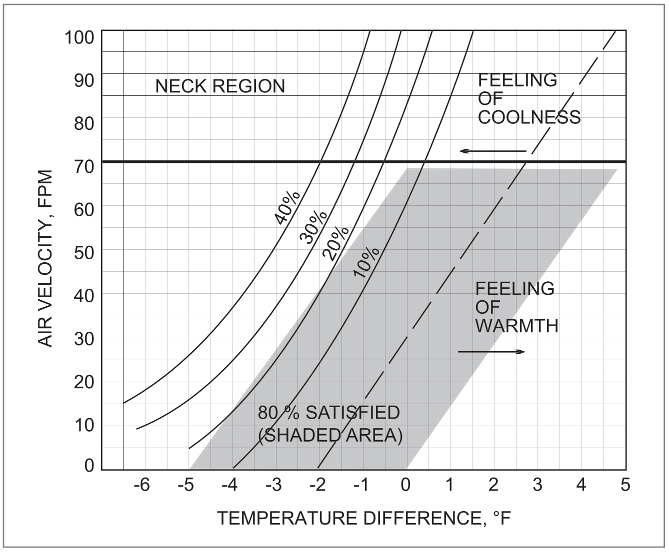
$$ø = (t_x - t_c) - 0.07 (V_x - 30)$$

Where

ø = Effective Draft Temperature
t_x = Local Air Temperatures, °F
t_c = Ambient Temperature,
(Average Room Temperature or Control Temperature, °F)
V_x = Local Air Velocity, fpm

Selection & Design Considerations

PERCENTAGE OF OCCUPANTS OBJECTING TO DRAFTS IN AIR CONDITIONED ROOM



Occupants objecting to the temperature and velocity conditions are represented by the percentage curves in the above graph. The dashed line indicates the difference between the feeling of heating and cooling. Research indicates that 80% of the occupants in a typical office environment would feel comfortable at -3 < ø < +2 with a velocity less than 70 fpm. This 80% comfort criteria is represented by the shaded parallelogram.

Based on the "effective draft temperature" equation, temperatures and velocities can be uniformly measured in an occupied space to determine the "Air Diffusion Performance Index." ASHRAE Standard 113 provides a method for testing to determine these values, typically in a laboratory setting. The ADPI expresses the number of points at which the draft temperature satisfies the comfort criteria as a percentage of the total number of points. Thus, an occupied space with an ADPI rating of 80 translates to 80% of the occupants will be comfortable when located anywhere within the occupied space (floor to 6' level). The higher the ADPI rating, the better. An ADPI greater than 80% is indicative of a well mixed air distribution system.

In order to obtain an acceptable ADPI rating, the designer or engineer must study a variety of variables.

- Temperature Differentials of Supply and Room Air
- Outlet Type and Design
- Supply Air Volumetric Flow Rates
- Space and Diffuser Layout
- Design Heating and Cooling Loads

ADPI is for cooling mode conditions only. Heating conditions can be evaluated using ASHRAE Standard 55 guidelines. The ADPI technique uses isothermal throw data, determined under ASHRAE Standard 70, to predict what will happen under cooling conditions.

Once space parameters are understood, proper air distribution devices can be selected to assure acceptable ADPI ratings.

GRD SELECTION & DESIGN CONSIDERATIONS

For the best thermal comfort conditions and the highest ventilation effectiveness in an occupied space, the entire system performance of air diffusers should be considered. This is particularly true for open spaces, where airstreams from diffusers may interact with each other and for perimeter spaces, where airstreams from diffusers interact with hot or cold perimeter walls. While throw data for individual diffusers are used in system design, an air diffuser system should maintain a high quality of air diffusion in the occupied space with low temperature variation, good air mixing and no objectionable drafts in the occupied space. Studies show that ADPI can be improved by moving diffusers closer together by specifying more diffusers for a given space and air quantity and by limiting the differential temperatures of supply air to room air.

For VAV (Variable Air Volume) systems, the diffuser spacing selection should not be based on maximum or design air volumes, but rather on the air volume range in which the system is expected to operate most of the time.

The **throw** of a jet is defined as the distance from the outlet device to a point in the air stream where the maximum velocity occurring in the stream cross section has been reduced to a selected terminal velocity (vt). Catalog performance tables list terminal velocities at 150, 100 & 50 fpm under isothermal conditions (except when noted otherwise). The **throw distance** of a jet is denoted by the symbol Tv, where the subscript number indicates the terminal velocity for which the throw is given.

The **characteristic room length L** is generally defined as the distance from the outlet device to the nearest boundary wall in the principle horizontal direction of the airflow. However, where air injected into the room does not impinge on a wall surface, but mixes with air from a neighboring outlet, the characteristic length L is defined as one-half the distance between outlets plus the distance the mixed jets must travel downward to reach the occupied zone. The table to the left summarizes definitions of characteristic length for various air outlet devices.

CHARACTERISTIC ROOM LENGTH FOR DIFFUSER TYPES

Diffuser Type	Characteristic Length, L
High Sidewall Grille	Distance to Wall Perpendicular to Jet
Circular Ceiling Diffuser	Distance to Closest Wall or Intersecting Air Jet
Sill Grille	Length of Room in the Direction of the Jet Flow
Ceiling Slot Diffuser	Distance to Wall or Midplane between Outlets
Perforated, Louvered, Ceiling Diffusers	Distance to Wall or Midplane between Outlets

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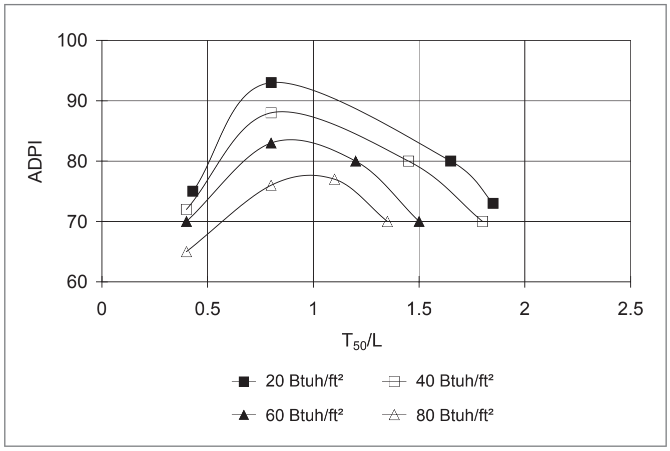
ADPI Selection Guidelines

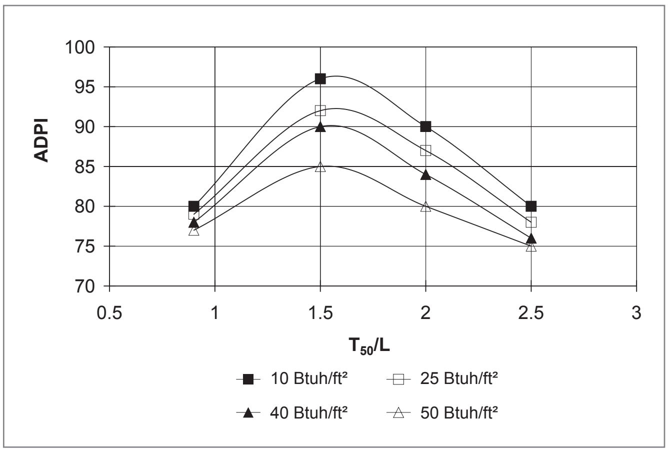
Optimal comfort levels in a space can be estimated using the various outlet types and different room loads. The recommendations in the following nine types of ADPI graphs cover cooling loads of up to 80 Btuh/ft² of floor surface. The loading is distributed uniformly over the floor up to about 7 Btuh/ft², lighting contributes about 10 Btuh/ft² and a concentrated load against one wall that simulates a business machine or a large sun-loaded window supplies the remainder. The maximum ADPI condition is lower for the highest loads; however, the optimum design condition changes only slightly with the load.

For a given diffuser type, the ADPI performance as a function of room load and T_{50}/L has been studied in ASHRAE research, and is available in the ASHRAE Handbook of Fundamentals. Refer to the example graph to the right of a round pattern diffuser, such as Krueger's Model 1400 Louvered Diffuser, which has a Type 3 Circular Ceiling Pattern indicating the ADPI performance.

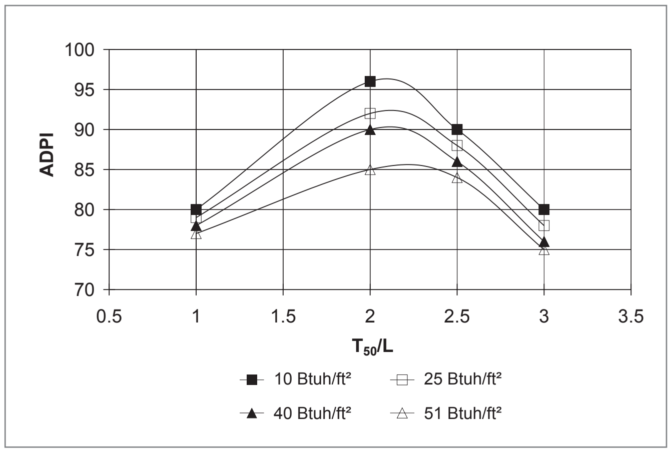
EXAMPLE: GRAPHICAL ADPI SELECTION GUIDE



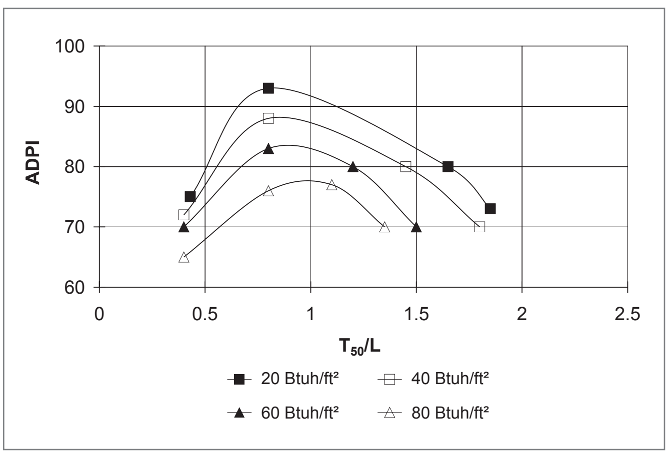
TYPE 1: SEMI-CIRCULAR PATTERN CEILING DIFFUSER



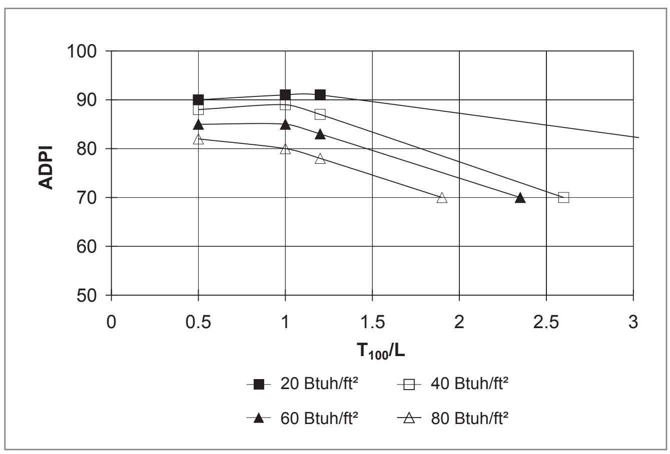
TYPE 2: CROSS FLOW PATTERN CEILING DIFFUSER



TYPE 3: CIRCULAR CEILING PATTERN



TYPE 4: CEILING SLOT DIFFUSER



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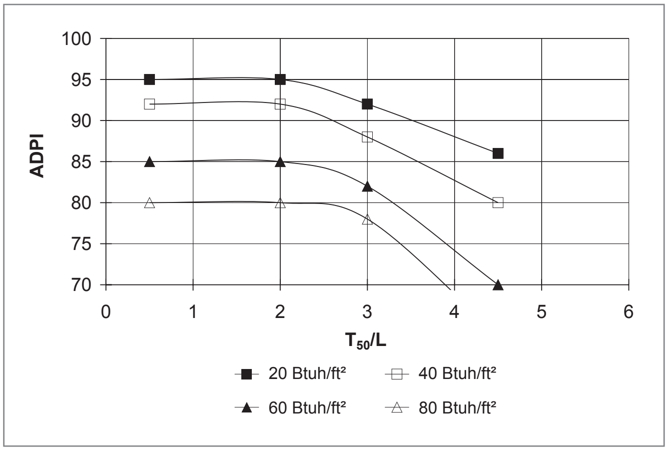
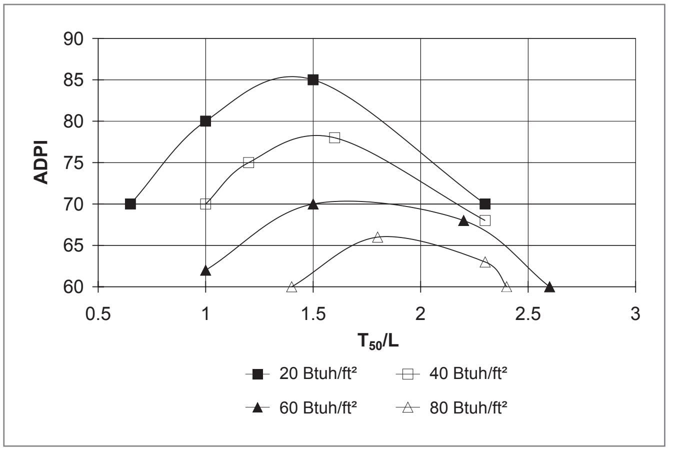
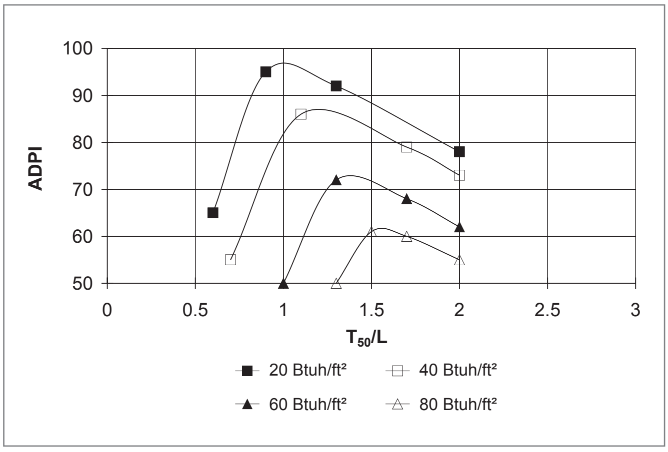
TYPE 5: LIGHT TROFFER



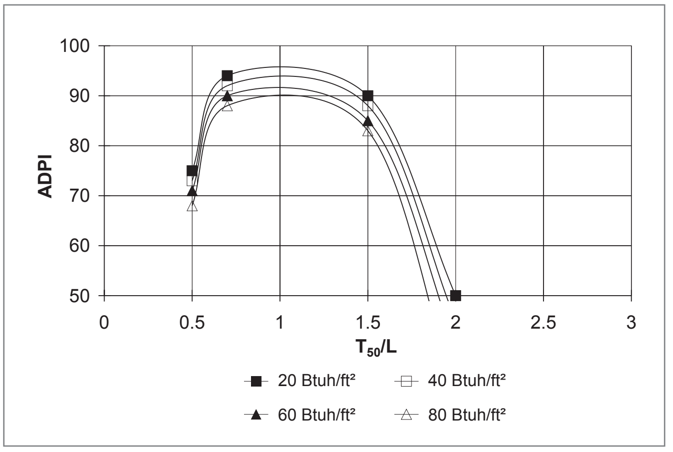
TYPE 6: HIGH SIDEWALL



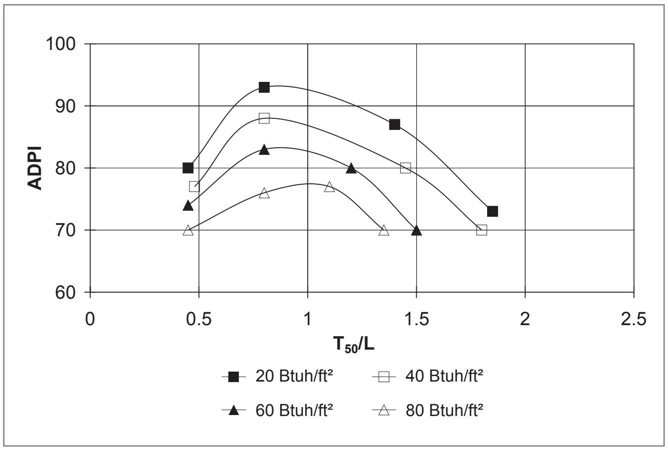
TYPE 7: SILL GRILLE STRAIGHT VANES



TYPE 8: SILL GRILLE SPREAD



TYPE 9: SQUARE CEILING PATTERN



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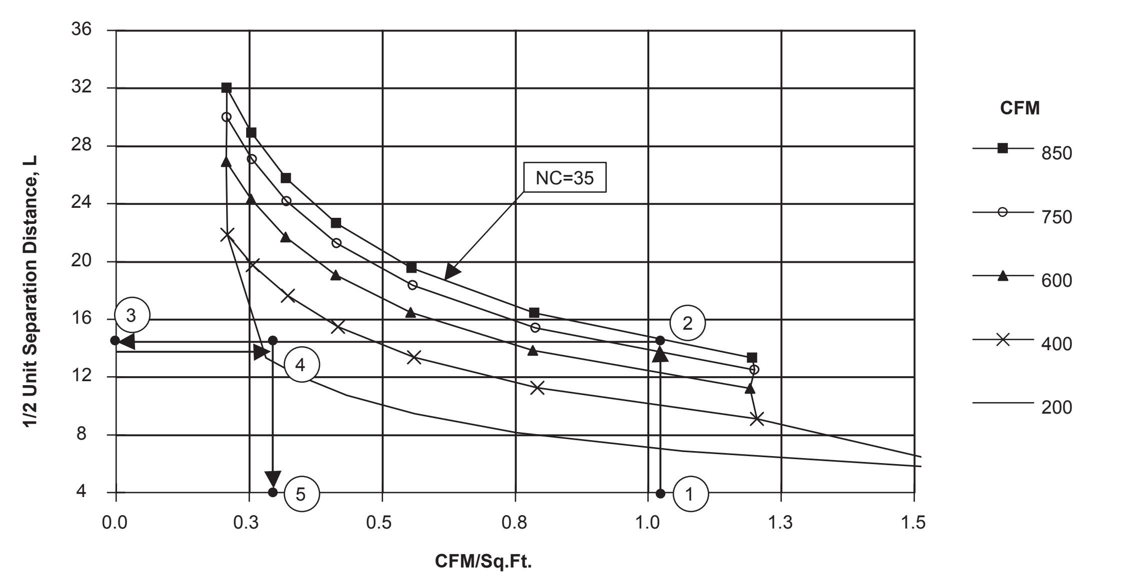
EXAMPLE: 1400 24"x24" PANEL, 14" INLET, 4-WAY (SPACING FOR 80% ADPI) (NO DAMPER)



NOTE: Chart based on 20 Btuh/ft² loads.

ADPI Charts

This data can be used to prepare an ADPI performance envelope graph for any type of ceiling diffuser based on neck size and resultant throw, as a function of room air loading, or cfm/ft². For most interior spaces, heat load is less than 20 Btuh/ft². An example of such a graph, for a Krueger Model 1400 Louvered Diffuser with a 14" inlet, is provided above.

The effective operating airflow rate range for a space with the above diffuser can be estimated by following this procedure.

1. Determine the maximum air volumetric flow requirements based on load and room size. For VAV systems, evaluation should include the range of flow rates from minimum occupied to design load.
2. Enter the above graph at the maximum (design) airflow rate per unit area (cfm/ft², x-axis).
3. Proceed up to the airflow resulting in the maximum allowed sound level.
4. Proceed across to the value of L for that airflow rate (y-axis). This is the maximum half diffuser spacing for acceptable ADPI. (ADPI of 80%)
5. Determine the minimum flow rate for that diffuser by starting at the separation distance, proceeding to the right to the left-most edge of the acceptable envelope, and proceeding down to the minimum CFM/ft².

Example:
Room Size: 29' x 29' with 9' Ceiling
Load: 21.6 Btuh/ft² or 25000 Btu/h
Volumetric Flow: 1 cfm/ft²
Diffuser: Krueger Model 1400 Louvered Diffuser with 14" Neck
Maximum Allowable NC: 35

Determine:
L (Characteristic Length) and Minimum Volumetric Flow
for Acceptable 80% ADPI.

Solution:
1. Refer to the ADPI performance envelope graph for the 1400 series with 14" neck and 24"x24" panel with 4-way throw. This chart is based off 20 Btuh/ft².
2. Draw a vertical line from the x-axis value of 1 cfm/ft² [point 1] to line with NC = 35 (850 cfm line) [point 2].
3. Draw a horizontal line from [point 2] to the y-axis [point 3]. Point 3 is the value of L. In this example, L = 15'.
4. To find minimum volumetric flow, locate [point 4] created by the line drawn from [point 2] to [point 3]. From [point 4] draw a vertical line down to the x-axis to [point 5]. Point 5 is the minimum volumetric flow for acceptable flow. In this example, the value is .33 cfm/ft².

NOTE: ADPI performance envelope graphs are provided for most ceiling diffusers in their performance sections throughout this catalog.

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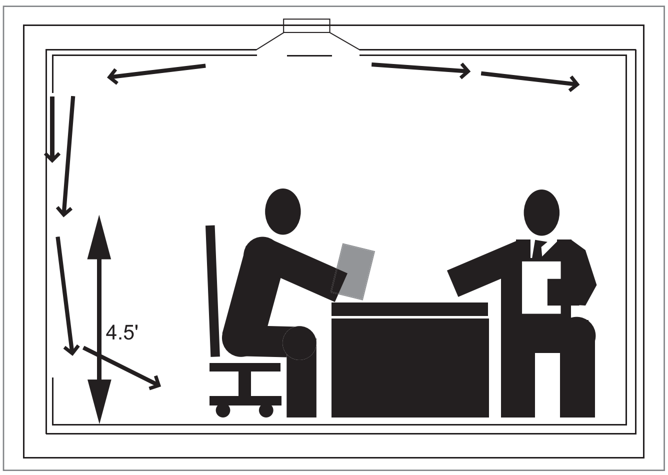
Overhead Perimeter Heating
A critical application of horizontal pattern ceiling diffusers involves heating a perimeter zone in the winter from a ceiling located diffuser.

In order to provide the best overall performance in both heating and cooling, providing the diffuser pattern will not be changed as the seasons change, tests have shown that several rules need to be followed.

- The discharge temperature cannot be more than 15°F above room temperature (heating mode) or excessive stratification will result.
- The discharge pattern should be both towards and away from the glass for both summer and winter acceptability.
- The 150 fpm terminal velocity (measured across the ceiling and then down the glass) must come to within 4.5' of the floor, or the jet will go back up to the ceiling and not mix with the cold convective current falling off the glass.

OVERHEAD HEATING



Failure to meet any of these requirements will usually result in vertical temperature stratification that does not meet ASHRAE comfort limitations and has poor system temperature response and ventilation short-circuiting. (ASHRAE Standard 62.1, 2004 Addenda N mandates that these conditions be met, or an increase in ventilation rates is required.) To achieve the required heating throw, most systems will need to supply at least 0.8 cfm/ft^2 in heating mode. In most northern climates, it will only require 10°F ΔT (above room) to meet design heating load requirements.

When energy codes do not allow for high enough air supply rates with VAV-reheat units for the diffuser to be effective when heating, either fan powered terminals or a separate heating system may be required. If a separate system is used, a vertical pattern diffuser or jet, at the glass, may provide an effective means of heating a perimeter zone without the supply temperature limitations previously noted.

Return & Exhaust Openings
The selection of return and exhaust openings depend on the following.

- Velocity in the Occupied Zone Near the Openings
- Permissible Pressure Drop Through the Openings
- Noise

Room air movement is typically not influenced by the location of the return and exhaust outlets beyond a distance of one characteristic length of the exhaust or return grille. Air approaches the opening from all directions, and its velocity decreases rapidly as the distance from the opening increases. Therefore, drafty conditions rarely occur near return openings. The table below shows recommended return opening face velocities.

RETURN FAN VELOCITY

Recommended Return Inlet Face Velocities	
Inlet Location	**Velocity Across Gross Area, FPM**
Above Occupied Zone	> 800
Within Occupied Zone, Not Near Seats	600 - 800
Within Occupied Zone, Near Seats	400 - 600
Door or Wall Louvers	200 - 300
Through Undercut Area of Doors	200 - 300

The location of return and exhaust openings is far less critical than that of supply outlets. However, the return should be located in an area to minimize short circuiting of supply air. If a return is located in a stagnant zone, it will return the warmer air in the space during cooling. In rooms with vertical temperature stratification, such as computer rooms, theaters, bars, kitchens, dining rooms, etc., exhaust openings should be located near the ceiling to collect warm air, odors, and fumes.

Acoustical Considerations
The most acceptable frequency spectrum for HVAC noise is a balanced spectrum. This means that it sounds neutral and is not too “hissy” (high-frequency content) or too “rumbly” (low-frequency content). The room criterion (RC) curves approximate the balanced spectrum deemed acceptable by most people. They are straight lines sloped at -5 dB per octave band on a sound level versus frequency graph. Although still widely in use, the noise criterion (NC) curves are beginning to be replaced by the RC curves because an RC rating provides more information about sound quality.

The top graph on the following page shows an indoor sound spectrum and the manner in which fan noise and diffuser noise contribute to that spectrum at various frequencies. The fan noise has been attenuated such that it approaches the sound criterion (in this case, an RC-35 contour) only in the lower octave bands. If this were the only noise present in the space, it would be considered rumbly by most listeners. However, the diffusers have been selected to balance the spectrum by filling in the higher frequencies so that the quality of the sound is more pleasant. Unfortunately, achieving a balanced sound spectrum is not usually this easy, as there may be numerous

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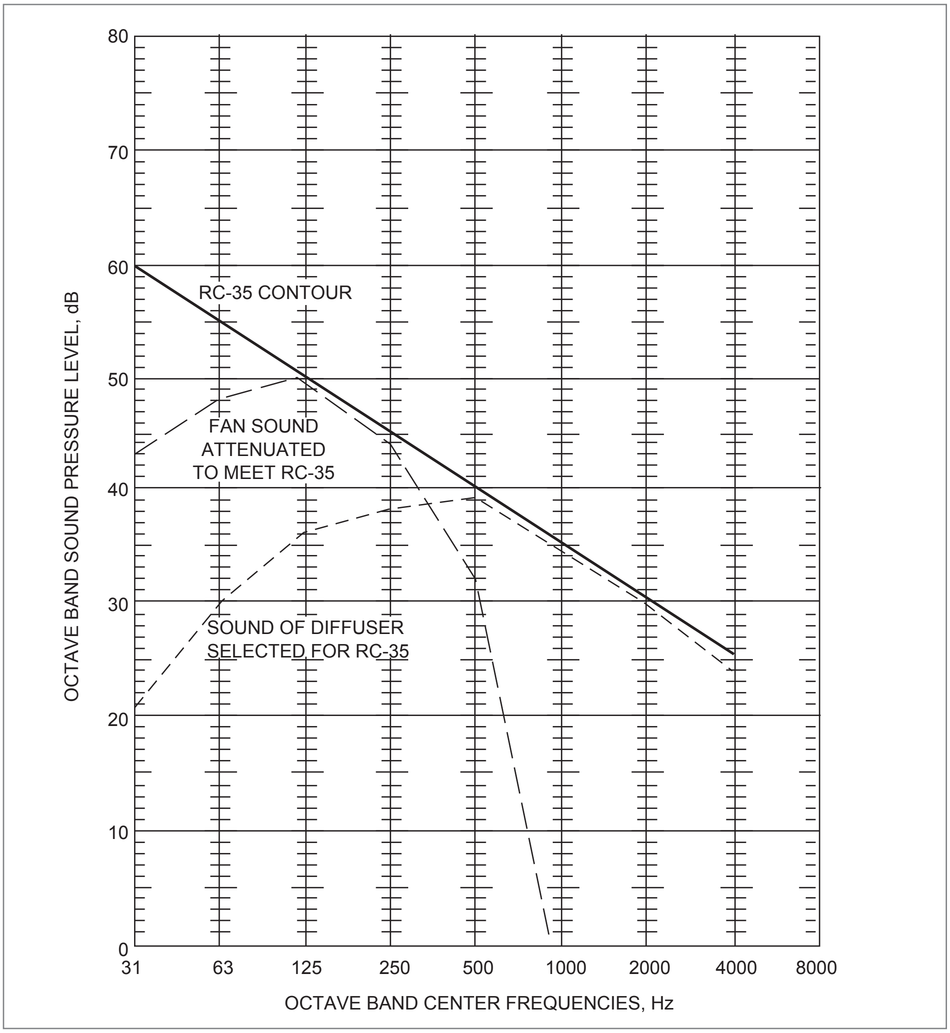
other sound sources to consider. AHRI 885 provides a method of estimating space sound levels from known sound power levels. A spreadsheet (885-calc.xls) is available on the Krueger website at www.krueger-hvac.com to analyze the effect of diffuser noise along with air terminal radiated and discharge sound.

Sound levels reported for diffusers are conducted in accordance with ASHRAE Standard 70, in a reverberant room. This procedure requires ten equivalent diameters of straight duct leading to the unit being tested. In practice, however, there is seldom even one equivalent diameter of straight run at the diffuser inlet. Our tests indicate that while the effect on sound generation differs from unit type to unit type, an average effect of +3 NC can be used to estimate these inlet effects.

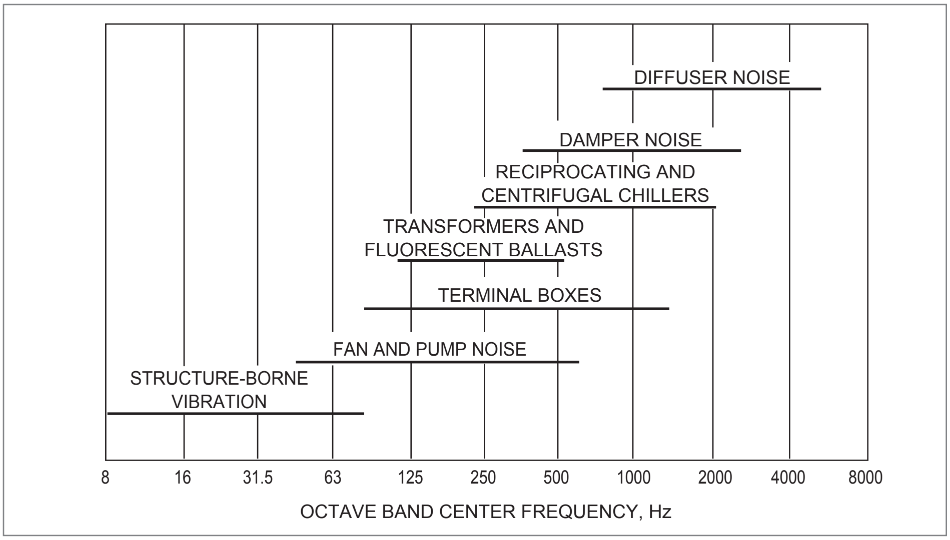
Additionally, room NC levels published for diffusers use a 10 dB reduction for room effect. Most rooms, if the observer is standing under the diffuser, will have an effective reduction less than 10, more like 8 dB, which results in an additional correction of +2 NC.

The sum of these two realities is that catalog NC values should probably be increased by five to estimate sound levels in the space. On the other hand, the typical NC=35 requirement for most open plan offices is probably too low to provide the desired speech privacy levels, and an NC=40 is seldom objectionable in that situation. As a result, selecting at a catalog NC=35, which results in an RC=40N, will usually provide occupant acceptance on open plan offices.

BALANCED SOUND SPECTRUM RESULTING FROM PROPER SELECTION OF AIR OUTLETS AND ADEQUATE FAN NOISE ATTENUATION



FREQUENCIES AT WHICH VARIOUS TYPES OF MECHANICAL AND ELECTRICAL EQUIPMENT GENERALLY CONTROL SOUND SPECTRA



AIR DISTRIBUTION ENGINEERING

References

a1-4: Centerline Velocities of Axial & Radial Jets. 2001 ASHRAE Fundamentals Handbook; Chapter 32, Figure 2.

a1-6: Air Motion Characteristics of Outlets In or Near Floor with Non-Spreading Vertical Jet. a1-7: Air Motion Characteristics of High Sidewall; Group A Grilles. a1-7: Air Motion Characteristics of Ceiling; Group A Diffusers. a1-8: Air Motion Characteristics of Outlets Mounted In or Near Floor that Discharge Air Vertically in a Spreading Jet. a1-8: Air Motion Characteristics of Outlets Mounted In or Near the Ceiling that Project Primary Air Vertically. Straub, H.E., S.F Gilman, and S. Konzo. 1956. Distribution of Air Within a Room for Year-Round Air Conditioning - Part I. University of Illinois Engineering Experiment Station, Bulletin No. 435.

a1-10: Standard 55-93R Comfort Envelope. ASHRAE Standard 55-93. ISO 7730.

a1-11: Percentage of Occupants Objecting to Drafts in Air Conditioned Room. 2001 ASHRAE Fundamentals Handbook; Chapter 32, Figure 12. Parallelogram defined in ASHRAE Standard 113-90.

a1-16: Balanced Sound Spectrum Resulting from Proper Selection of Air Outlets and Adequate Fan Noise Attenuation (top). Frequencies at which Various Types of Mechanical and Electrical Equipment Generally Control Sound Spectra (bottom). 1997 ASHRAE Fundamentals Handbook; Chapter 7, Figure 1.

a1-23: NC Chart. 2001 ASHRAE Fundamentals Handbook; Chapter 7, Figure 6.

a1-11: Characteristic Room Length for Several Diffuser Types. 2001 ASHRAE Fundamentals Handbook; Chapter 32, Table 3.

a1-15: Recommended Return Inlet Face Velocities. 2001 ASHRAE Fundamentals Handbook; Chapter 32, Table 5.

a1-24: NC Radiated Reductions. Data in Accordance with AHRI Standard 885-08.

a1-24: NC Discharge Reductions. Data in Accordance with AHRI Standard 885-08, Appendix E.

a1-24: Ratio of Branch Flow to Total Flow. Data in Accordance with AHRI Standard 885-08.

a1-25: Recommended Indoor Design Goals (NC Ranges). 2001 ASHRAE Fundamentals Handbook; Chapter 7, Table II.