Cold Air Distribution Design Manual

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Prepared by
Titus, Division of Tomkins Industries
990 Security Row
Richardson, TX 75081

Dan Int-Hout

Reinhard Ratz

Prepared for
Electric Power Research Institute
3412 Hillview Avenue
Palo Alto, California 94304

EPRI Project Managers

M. Khattar

R, Wendland

Commercial Customer Systems Group

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Abstract

In the selection of diffusers for a building, the engineer has few tools to assist him in making an informed selection. Often the choice is driven by architectural concerns and past experience. With Cold Air Distribution (CAD) Systems, however, the choices are more critical due to concerns over ventilation and comfort at reduced air flows. The EPRI Design Guide for Cold Air Distribution, recently re-released in 1995, provides a great deal of information on Cold Air System design. It does not, however, fully describe the process of diffuser selection when cold air is delivered directly to the space. To assist the engineer in making these selections, EPRI contracted for the creation of a design manual for cold air diffuser selection. This manual is in final edit and review at this time.

Selecting and sizing diffusers to be used with cold air systems is an important issue to today's building industry where optimal design is desired. Concerns over Indoor Air Quality (IAQ) and occupant comfort, combined with the reduced loads present in many spaces, leads to designs with very low air supply rates. The economic advantages of using colder supply air (smaller ducts, fans, etc.) are weighed against the problem of providing proper air circulation with the even lower air flows that result.

The Cold Air Distribution Manual shows that CAD is not very different from conventional air delivery systems. The colder air requires a more careful selection of diffusers, as unwanted drafts can be more objectionable with colder air. The same acceptable design practices which work for conventional supply systems still apply with air colder than 50^{o}. The process will show that some diffuser types are unsuited for very low air flow rates, regardless of the supply air temperature.

1
INTRODUCTION

Background

Selecting and sizing diffusers to be used with cold air systems is an important issue to today's building industry where optimal design is desired. Concerns over Indoor Air Quality (IAQ) and occupant comfort, combined with the reduced loads present in many spaces, lead to designs with very low air supply rates. EPRI (1995) has prepared a "Cold Air Distribution Design Guide" covering all aspects of the basis for and design of a CAD system. In this manual, the selection of a diffuser for use with a cold air system is discussed in detail, along with some of the other issues covered in the EPRI CAD design Guide.

Purpose

The purpose of this manual is to assist engineers in selecting and applying diffusers for applications involving the distribution of cold air. Cold air is defined, for the purposes of this manual, as air at temperatures below 52^{o}F being delivered directly to the occupied space.

Scope

This manual is limited to the proper application and selection of diffusers. It is not intended to be a justification of cold air, nor a design guide for selecting cold air over other systems. This manual will indicate the issues surrounding how to determine a diffuser's acceptability for cold air, and should be used with the EPRI **"Cold Air Distribution Design Guide"** for a full analysis of Cold Air Distribution (CAD) designs. It will be an extension of the ASHRAE Fundamentals Chapter 31 diffuser selection process, extended to include cold air/very high induction diffusers.

The manual has been created to solve two problems:

- ASHRAE's current guidelines on diffuser selection and Air Diffusion Performance Index (ADPI) selection are based on diffusers available at the time (early 1970s). The special very high induction diffusers often employed now with cold air systems were not available and not included in that analysis.

- Engineers need to be certain that diffusers selected for use with Cold Air Distribution (CAD) will be able to maintain both adequate and acceptable ventilation within the space over the range of expected air flow.

This manual discusses types of outlets, diffuser types, selection methods, comparison of diffuser types, Psychrometrics, heating applications, and economics. Often cold air diffusers will also be used for heating. With these varying conditions, occupant comfort and ventilation requirements must be considered.

Design parameters include throw, sound pressure drop and zone comfort. Several methods are detailed outlining the selection and application of diffusers for cold air applications. Manufacturer catalog data at isothermal conditions can be used with known temperature conditions to calculate expected performance. At least one manufacturer has throw and drop already available in catalog tables for cold air diffusers given colder outlet air.

The use of cold air delivery systems has increased over the past few years. Before the development and acceptance of direct delivery of cold air with qualified diffusers or with special cold air diffusers, mixing systems were used to deliver air to spaces at the traditional 52-55^{o}F. These included both series and parallel fan-powered terminals and air-to-air induction devices. The increased energy required by the fan motors or higher system static required for induction partially offset energy savings from CAD design. Direct injection of cold air offers the maximum energy savings resulting from the reduced air quantities.

A number of studies (See Appendix C) have been conducted into the physical aspects of cold air emerging from a horizontal diffuser and then mixing with room air. This manual emphasizes the essential considerations that will allow the consulting engineer to design and specify an optimum cold air distribution system.

A discussion of the basics of air distribution is required in order to lay a foundation for discussions of cold air applications.

2

THERMAL COMFORT

An understanding of the principles of room air distribution helps in the selection, design, control, and operation of air systems. The real evaluation of air distribution in a space, however, must answer the question; Are the occupants comfortable?

In general, one is thermally comfortable when one's body heat loss equals one's heat production without one being conscious of any changes in one's temperature regulating mechanisms. This condition is often referred to as thermal neutrality. However, there is considerable evidence that thermal neutrality is not often the preferred condition. Many studies have indicated that a "slightly cool" feeling produces the highest level of occupant satisfaction. Human body heat loss to the environment can occur through the following:

- Radiation
- Conduction
- Convection
- Evaporation

The comfort of an occupant is also determined by both occupant variables and the environmental conditions of the space. Occupant variables include activity level and metabolic rate (reported in Met units) , as well as clothing levels (reported in Clo units). The environmental factors that influence space comfort conditions include:

- Dry bulb and radiant temperatures. Radiant temperatures seldom differ significantly from dry bulb air temperatures, except in direct sunlight or within a couple of feet of exterior glass walls. Air temperature measurements, however, should be shielded from potential radiant effects.
- Relative humidity (RH). A single measurement is typically sufficient to characterize the room, or even the building RH, as it seldom varies significantly throughout the space.
- Air speed. Air speed is a nondirectional value, best measured with omni-directional anemometers. Air velocity, on the other hand, implies a known direction. The direction is seldom known in interior, occupied spaces, and air speed is the proper descriptive term. The average measured air speed is commonly referred to as the average air motion in the space.

The design of the air distribution system should address the above factors so that the occupant's heat loss is maintained at a comfortable rate. We will discuss three different approaches to determining an individual's comfort level.

Comfort Limits Set by ASHRAE Handbook, ASHRAE Standard 55-92, and ISO Standard 7730-94

For many years, it has been shown that individual comfort is maintained through changes in season when the following conditions are maintained in the occupied zone of a space:

- Dry bulb maintained between 73 and 77° F
- Relative humidity maintained between 25 and 60%
- Maximum air motion in the occupied zone :
  - 50 fpm cooling
  - 30 fpm heating
- Floor to 6 ft. level, 5-6°F maximum temperature gradient

The ASHRAE comfort standard states that no minimum air movement is necessary to maintain thermal comfort, provided the temperature is acceptable. To maximize energy conservation, therefore, an air system design should attempt to maintain proper temperatures at the lowest possible air speed.

Analyses of recent ASHRAE studies on occupant preferences indicate that it is easy to misdiagnose an occupant's response to his/her environment. The problem comes with the definition of thermal neutrality. Many occupants have stated they felt that they were thermally neutral, but desired higher air speeds, because the room felt stuffy. In a significant number of cases, these locations were actually slightly above the predicted comfort zone. It has been observed that dropping the temperature slightly produces a sensation of slightly cool, but fresher. Dropping the relative humidity has the same effect.

The above conditions assume occupants are sedentary or slightly active and appropriately dressed. Variations in clothing can have a strong effect on desired temperature levels, with traditional differences between men's and women's clothing often creating circumstances where a single setpoint will fully satisfy neither.

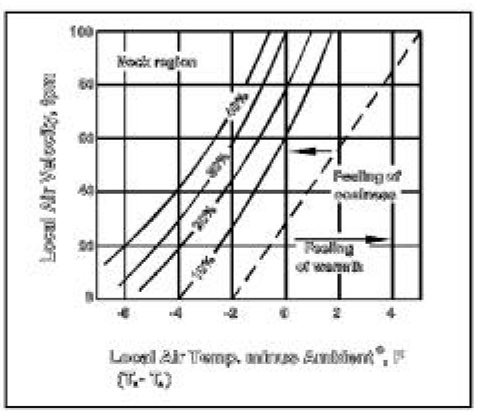
In meeting the above criteria for comfort, the temperature of the space and the relative humidity are largely controlled by the mechanical equipment, including chillers or package units, air handlers, room thermostats, and air terminal units. The air motion in the occupied zone is a function of the discharge velocity, discharge temperature (and room load), and the pattern of the air diffusion device into the space. At today's relatively low (< 1 cfm/sq. ft) air delivery rates, and with properly selected diffusers, room load is often the strongest variable in setting room air motion. Thus, if the objective of a design is to increase air motion, it probably cannot be achieved, except by increasing the space load!

It should be noted that ASHRAE 55-1993 includes allowances for very high relative humidities, provided the temperature is cool enough. This provision was included to avoid prohibiting economizer operation in climates where these conditions may result. It is noted that these high humidity conditions can result in significant problems with the building structure due to condensation problems, and that other considerations may limit building humidities.

Air Diffusion Performance Index (ADPI): Comfort as a Function of Room Air Velocity and Temperature Difference

Some interesting relationships exist between room air motion and the feeling of occupant comfort. The available comfort data show that the feeling of comfort is a function of the local room velocity, local temperature, and ambient temperature.

Local temperature (T_x**)** is the temperature at a given point in a space. Ambient temperature (T_c) is the desired room or control temperature and can be considered the thermostat setpoint. The basic criteria for room air distribution can be obtained from the curves shown in **Figure 1**. The chart shows the equivalent feeling of comfort for varying room temperatures and velocities at a person's neck. The percentage curves indicate the number of people who would object to the temperature and velocity conditions. The same comfort perceptions are shown in **Figure 2** for the ankle region. As women are less likely to wear insulating socks, they tend to be more sensitive to floor temperatures than men.



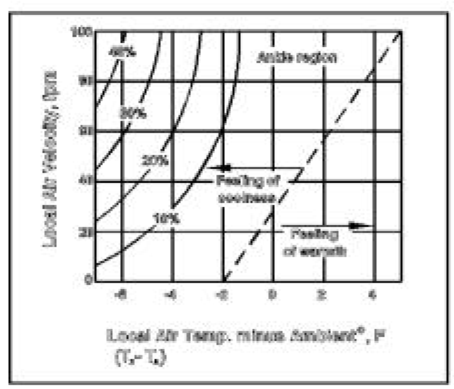


Figure 1,

Neck Comfort Region

Figure 2,

Ankle Comfort Region

Local Velocity FPM	Temperature Deg. F	(Tx-Ta)	Percentage Objecting
80	75	0	20
60	74	-1	20
40	73	-2	20
15	71	-4	20
60	75	0	10
40	75	0	5
30	75	0	Neutral
15	75	0	Warm

If 20% objections or 80% acceptance at the same velocities are allowed between Figures 1 and 2, the temperature deviation allowed between a person's ankle and neck levels would be about 4°F (less than ASHRAE values of 5-6°F).

Table 1 shows the relationship between local velocities and temperatures on occupant comfort. As an example, at a local velocity of 80 fpm, the local temperature can be maintained at 75°F to reach an 80% comfort level in the space. Table 1,

Velocities and Temperatures

The same 80% comfort level can be maintained with local velocity of 15 fpm and a local temperature reduced to 71°F.

The lower portion of Table 1 shows the effect on comfort of room air velocity with local temperature remaining constant at 75°F. For example, with a local velocity of 30 fpm and a local temperature at 75°F, the comfort reaction is neutral. Increasing the velocity to 60 fpm results in the number of those objecting increasing to above 10%. This phenomenon of "feeling cooler" can be illustrated by using a ceiling fan. A person can be "cooled" without decreasing the actual temperature by turning on a ceiling fan. The fan, in effect, increases the local air velocity and increases the feeling of coolness. It is shown that a velocity change of 15 fpm produces approximately the same effect on comfort as a 1°F temperature change. The dotted lines in Figures 1, 2, and 3 show the division between the feeling or perception of warmth and coolness.

Generally, the acceptable level of comfort for a space is considered to be at the point where 20% or less of the room occupants object to the room conditions. This would indicate that the given condition is acceptable to 80% of the occupants. Most people have perceived these subjective responses to drafts (temperature difference and air velocity). In 1938, Houghten et al. developed the curves shown in Figures 1 and 2**.** Utilizing these data, the equation for effective draft temperature was generated:

$$Ø = (t_X - t_C) - 0.07(V_X - 30) \qquad \text{(eq.1)}$$

where: $Ø$ = effective draft temperature

t_X = local air temperatures, ° F

t_C = ambient (control) temperature (avg. room temperature), ° F

V_X = local air velocity, fpm

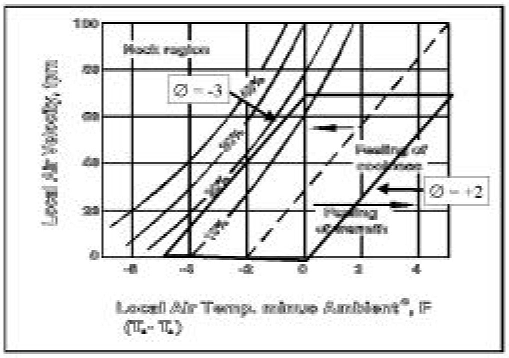


Figure 3,

ADPI Comfort Region

The ADPI was derived by Nevins and Ward (1968). The percentage of all local points in an occupied space where -3 ≤ Ø ≤ +2, with the velocity less than 70 fpm results in a single comfort index for the space.

In the shaded parallelogram shown in Figure 3, the left line is Ø = -3, the right line is Ø = +2 and the top line is 70 fpm. All the points within the shaded area are within the desired overall 80% acceptance.

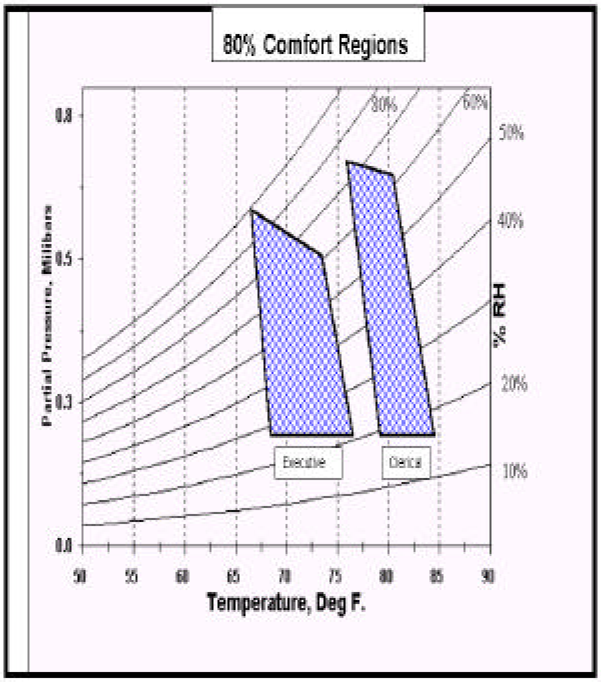
This method of obtaining a single point comfort rating can be used to rate a system in the field and provides a tool in the selection of a diffuser.

It is also expected that there is a strong correlation between ADPI and ventilation air mixing, or Air Change Effectiveness, and this issue is expected to be included in the upcoming revision to ASHRAE IAQ Standard 62-1989.

Fanger's PMV Comfort Index

Another comfort index which is used frequently to compare research results is Fanger's procedure which is the basis for ISO Standard 7730 (Appendix B). Fanger's method determines the Predicted Mean Vote (PMV) and the Predicted Percentage of Dissatisfied (PPD) at the occupants' locations in the occupied zone.

The Fanger equation includes thermal parameters not considered in the ADPI (relative humidity, mean radiant temperature, clothing insulation, and activity levels). The PMV approach has the advantage of providing a single number rating combining all comfort elements. The ASHRAE 55 and ISO 7730 standards yield essentially the same space conditions for acceptability. It is expected that the ASHRAE standard will incorporate PMV in its next revision. From both ASHRAE and ISO standards, the estimated comfort of 80% of the individuals in a space can be plotted. From a program developed as part of the ASHRAE 55 review process, a consensus computer program was developed and published. This program has been used to plot data on a psychometric chart for two sets of typically occurring conditions (as mentioned earlier, the high humidity region may not be consistent with good building design and IAQ Standard requirements).



Condition 1(Executive)

Met Rate = 1.1 (typical for office)
Clothing Rate = 1.0 (shirt, tie, long pants, socks)
Air Speed = 20 FPM (typical)

Condition 2 (Clerical) :
Met Rate = 0.9 (sedentary)
Clothing Rate = 0.5 (skirt, blouse, no socks)
Air Speed = 20 FPM (typical)

Figure 4,

PMV Plot.

It can be seen from Figure 4 that a single setpoint, such as 75^{o}F, 50% RH is not likely to satisfy even 80% of all individuals in a space. This figure also indicates the effect of relative humidity on comfort, an important consideration with cold air delivery systems, which usually lower the space RH by 10 to 20% over conventional 55^{o}F supply systems.

General Comfort Guidelines

Based on published guidelines and research, the following conditions should be maintained to achieve acceptable comfort levels. These conditions are adjusted for seasonal clothing levels:

- **Heating.** Generally, during heating, local air velocities are low, often below 20 fpm. If the 80% comfort factor is to be met, the maximum temperature gradient from ankle to the neck should be no more than 4°F.

- **Cooling.** During cooling, which is the predominant mode in most occupied spaces, local air temperature differentials generally should not

- be more than 1° to 2°F from ankle to neck region. To maintain the 80% comfort level, the air distribution system should limit the local air velocities to not exceed 50 fpm.
- **General.** There is no minimum air speed for comfort, provided acceptable air temperatures are maintained.

3

OUTLET TYPES

Outlets have traditionally been classified into several types to better describe their performance. Per Chapter 31, ASHRAE *Handbook of Fundamentals (1993),* diffusers are classified into five groups :

- **Group A.** Outlets mounted in or near ceiling that discharge air horizontally.
- **Group B.** Outlets mounted in or near the floor that discharge air vertically in a non-spreading jet.
- **Group C.** Outlets mounted in or near the floor that discharge air in a vertical spreading jet.
- **Group D.** Outlets mounted in or near the floor that discharge air horizontally.
- **Group E.** Outlets mounted in or near the ceiling that project air vertically.

The primary area of interest for cold air systems, and this manual, is Group A. The common application is an office with a 9- or 10-ft. ceiling. Air is discharged horizontally from a ceiling-mounted diffuser and is intended to mix with room air sufficiently so that the occupied zone remains comfortable while heat exchange and ventilation occur.

A side category of Group A is wall-mounted grilles close enough to the ceiling so that air jets experience the surface Coanda effect of clinging to the ceiling. Studies have been conducted on wall-mounted grilles with and without an adjacent ceiling.

The charts for these wall-mounted grilles show the high end air flow problem of air distribution. Generally, when cold air comes out at a very low flow rate from a horizontal grille or diffuser it will drop downward instead of having any type of ceiling surface effect. As the air flow is increased, the jet clings to the ceiling until it is mixed with room air. It is possible to put out a large enough quantity of air from a horizontal outlet for the jet to thicken into a billowing cloud that impinges into the occupied zone. This high air flow level is usually accompanied by high air noise level. A further problem with extremely high horizontal air flow rates is that ceiling tiles can be pushed upward by the jet. The solution is to have sufficient supply outlets spread over an area to maximize occupant comfort.

Engineering tables are also available for Group E outlets that are ceiling mounted and project air primarily vertically with displacement.

Floor-mounted outlets from Groups B, C, and D are not addressed in this manual. Manufacturers' catalogs and applications personnel can best address the use of specific outlets in cold air applications. Floor air distribution offers certain advantages that have led segments of the building industry to adopt this approach, but not yet with <55^{o}F air.

4

THROW AND DROP FROM OUTLETS

Figures 5 and 6 show the effect of buoyancy on drop and throw for cooling jets from a high sidewall double deflection grille. The relationship of cfm, velocity, drop, and throw is based on deflection angle and ceiling effect. Figures 5 and 6 represent 50 fpm terminal velocities and can be used to estimate throw and drop for sidewall grilles, with and without an adjacent ceiling. The circled dots on the figures represent a single outlet using varying configurations. While these are for specific conditions indicated, charts are available from manufacturers for other outlet conditions.

See Fig 5 (Attached)

Figure 5:
Buoyancy With Ceiling, Throw and Drop for Outlet 2 to 4 ft Below the Ceiling, 20^{o} Vertical Deflection, 0^{o} Spread

See Fig 6 (Attached)

Figure 6:
Buoyancy Without Ceiling, Throw and Drop for Outlet Mounted Without Adjacent Ceiling, 20^{o} Vertical Deflection, 0^{o} Spread

Notes:
1. Throw and drop values are based on 50 fpm terminal velocity.
2. Data are based on tests with 20°F cooling temperature differential in space with no boundary wall.
3. Data are based on Titus Models 271 and 272.
4. The small circle in the white area of each chart shows comparative performances of one size grille at 300 cfm and 600 fpm outlet velocity.
5. Shaded areas to right of each chart indicate noise level above 30 NC (Noise Criteria).

Figures 5 and 6 should be compared to ceiling height and the distance to walls and opposing air jets. If the drop extends into the occupied zone, or if the throw interacts with opposing diffuser jets, discomfort due to drafts may be experienced in the space.

Jets impinging on opposite walls are less of a problem, except when a table or desk is against that wall. Manufacturers' catalogs often provide specific guidance on throw and drop, often discussed as "mapping."

Other Diffusers With Cooling

In Figures 5 and 6, the drop (the distance below the ceiling, or sidewall grille location, to a specified terminal velocity, 50 fpm in the examples here) is primarily a function of the device flow rate. Therefore, to obtain a small drop, a small air quantity is necessary. For all outlets with horizontal air flow along the ceiling, the drop can be related to the air quantity and the ceiling height, as shown in Table 2. In these cases, air flow greater than maximum shown may result in objectionable drop into the occupied zone. These data are applicable to systems providing air no colder than 55°F to the space. If air colder than 55°F is supplied, as with cold air systems, the drop may be significantly greater than shown.

Table 2
Maximum cfm for Diffusers Based on Drop

Outlet	Ceiling Height (ft.)					
Type	8	9	10	12	14	16
1	550	1300	2200	4000	6200	9300
2	270	1500	1300	2100	3300	5500
3	1100	1500	2000			
4	650	1000	1500			
5	250	400	650	900	1400	1600
6	160	250	400	600	800	1000
7	320	500	800	1200	1600	2000

Type Description:
1. Round neck and face, adjustable, set for thin pattern
2. Round neck and face, non-adjustable, thick pattern
3. Square face, round (360°) pattern.

4. Perforated, square pattern
5. Perforated, four-jet pattern
6. Linear slot, cfm/side
7. Linear ceiling grille

5

TYPICAL AIR DISTRIBUTION CHARACTERISTICS

The following are important terms to remember when selecting outlets and applications.

Room Primary Air

Room Primary Air is the starting point when laying out or investigating the space room air motion. (This is not to be confused with the air entering a mixing air terminal, often referred to as Primary Air, which is combined with Induced Air to result in a Mixed Air Quantity.) Room Primary Air is defined as 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 isovels taken through two perpendicular planes. These show the number and angles of the jets in the primary airstream.

Maximum velocities in the primary air can be measured, and plotted, using procedures outlined in ASHRAE 70-91 *"Method of Testing for Rating the Performance of Air Outlets and Inlets*". 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.

Total Air

Total Air is defined as the mixture of Room Primary Air and Room Air which is under the influence of the outlet conditions. Normally the Total Air has a relatively high velocity (>100 fpm) but it has no sharply defined lower limit. Even though the Total Air follows the general pattern indicated by the Room Primary Air, its spread and travel may not be in proportion to that of the Room Primary Air. Other factors such as ceiling height, obstructions, internal and external loads, etc. disturb the orderly course of the airstream.

The temperature difference between the Total and Room Air (see following) produces a buoyancy effect which causes cool Total Air to drop at some point in the room and warm Total Air to rise. The most complete mixing of Total and Room Air occurs during isothermal conditions. Consequently, the location and type of outlet can reduce the buoyancy effects and increase the travel of the Total Air during heating when cool Room Air is induced and mixed rapidly with Room Primary Air, and during cooling when warm Room Air is induced and mixed rapidly with Room Primary Air. In addition to the outlet type and location, the action due to buoyancy effects is greatly dependent on the temperature differential between the supply air and the room air.

Air has a tendency to "scrub" surfaces. A near perfect situation can be envisioned where the Total Air covers all the walls and ceiling in a thin film. The occupied space would then be completely enclosed within an envelope of conditioned air. (No matter how desirable it might be, from the smudging standpoint, to keep the Total Air off ceiling surfaces, good air distribution usually requires that all surfaces be used.)

Since the Total Air within a confined space is affected by factors other than outlet conditions, it is not subject to complete analytical treatment. However, air characteristics for cooling and heating within a free space can be estimated. 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 can be created.

Natural Convection Currents

Natural Convection Currents are created by a buoyancy effect caused by the difference in temperature between the room air and the air in contact with a warm or cold surface. The air in contact with a warm surface will rise and the air in contact with a cold surface will fall. Convection 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 room air motion, but also play a major role in the comfort of a space. At today’s lower supply air rates, natural convection may be the predominant variable in determining actual room air motion levels in the occupied zone.

Results of tests of natural convection have shown ankle level temperatures during non-air supply heating may be 5°F , or more, below room temperature and that velocities ranged from 15 to 30 fpm. Figure 1 indicated that about 10% of the occupants would object to these conditions.

Stratification Layers and Zones

Stratification Layers exist in many situations. They often occur in practice as identified by a region where a layer of smoke will "hang" for some time. Whether a true stratification layer actually exists is not important, but the concept of a region of relatively low (<20 fpm) air speeds zone leads to a better understanding of air distribution. At the center of any circular air pattern, for example, there will be a stratified region. The presence of a stratification layer or zone is not indicative of poor air distribution, and in fact can almost always be found at some location in a space. Only if the temperatures in a stratified zone are significantly different (> 4°F) than the room average, is there any concern over these conditions. (The ASHRAE Comfort Standard *(55-1992 “Thermal Environmental Conditions For Human Occupancy”)* states that “there is no minimum air motion for comfort”, provided that the temperatures are satisfactory).

Natural convection currents form a *mixing zone* between the stratification layer and the ceiling during cooling and between the stratification layer and the floor during heating.

Return Intake

The Return Intake (air inlet) affects only the air motion within its immediate vicinity. Even natural convection currents possess enough energy to overcome the draw of an intake. This does not mean that the return location is not important, but only that it has little effect on the room air motion. Other considerations may affect the placement of the return air inlet.

Room Air

The room air motion picture is completed when the remaining Room Air drifts towards the Primary and Total Air. The highest air motion in the space is in and near the Room Primary and Total Air. The most uniform air motion is between the Total Air and Stratification Layer. The lowest air motion is in the Stratification Zone.

Temperature Gradient

The factors discussed above are interrelated and affect the resultant space temperature distribution. Where the air motion is uniform (between the Total Air and the Stratification Layer) the temperatures are approximately equal and uniform. As the Stratification Layer is crossed, the temperatures in the Stratification Zone may vary considerably, as a result of natural convection. Gradients in the Stratification Zones usually show that the air is stratified in layers of increasing temperature with an increase in space height.

Since the Stratification Zone depends primarily on natural currents, it depends on the magnitude of the perimeter heating or cooling load, space construction and volume, the area of exposure, or the interior load for uniform temperature mixing. It has been observed that the gradient changes with indoor-outdoor temperature difference. This will vary from zone to zone. Consequently, the magnitudes of the temperature variations between levels will be smaller:

1. In mild climates than in severe climates,
2. In space having exposed walls with greater resistance to heat flow,
3. With minimum internal loads.

6

DIFFUSER SELECTION METHODS

A number of different techniques can be used for selecting diffusers. These different methods discussed below, along with the advantages and disadvantages.

Method 1. Selection by Noise Criteria (NC) or alternatively, Room Criteria (RC)

The type of outlet selected for a space can be based either on the performance of that type of device, or on the architect's requirements based on appearance. The most frequently used procedure to select the size of the selected outlet type is by using the estimated room NC (Noise Criteria) level as presented in a manufacturer's catalog. Manufacturers determine sound power (or sound energy generated by a diffuser) in a reverberant chamber, in several octave bands or frequencies, under idealized inlet conditions as required by the testing standards, and for repeatability. A single number rating is reported in catalogs, which is based on a "typical" room attenuation of 10 dB (decibel) in all octave bands.

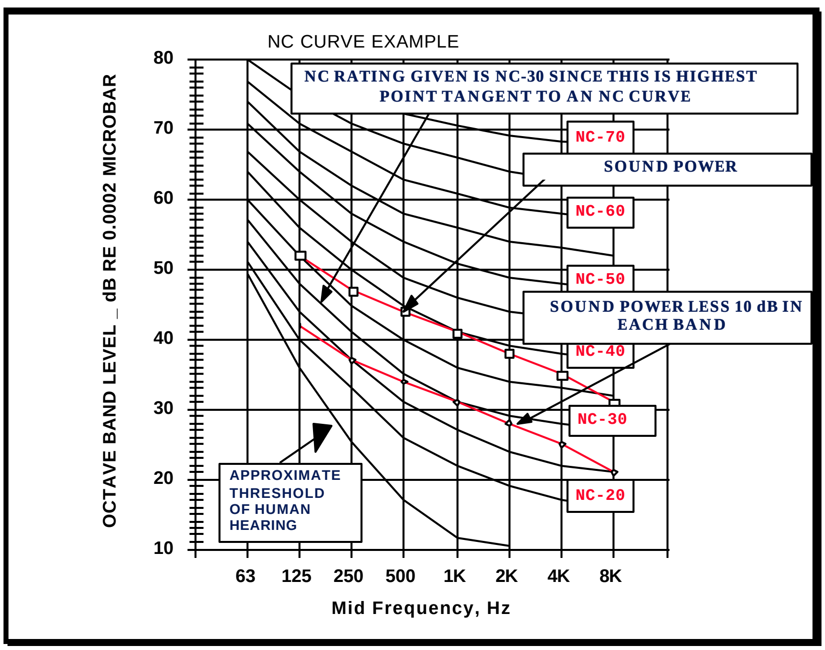


Figure 7
NC Chart with Sound Power - Sound Pressure Reduction

The 10 dB reduction results in an estimated sound pressure, as would be measured by a sound level meter in a "typical" room. This can be used to determine an NC rating, as shown in Figure 7. The NC rating is based on the highest NC line which is tangent to the observed sound spectra, in this example, an NC=30.

ASHRAE is recommending a change from NC to an RC (Room Criteria) rating. It will take some time, however, before RC data are available in catalogs. In most cases, a diffuser's RC and NC ratings will be the same, as diffusers tend to have their predominant sound levels in the frequencies (500-2000 Hz) where NC and RC are equivalent. Linear diffusers, which are often predominant in lower frequencies, may have lower RC ratings than NC ratings.

To take into account the number of outlets, distance, room, etc., manufacturers often provide guidance. Additional guidance can be found in ASHRAE handbooks, ARI 885-90, and other publications on sound applications.

Multiple outlets in a space at the same catalogued NC rating will result in an increase in the actual sound levels heard. However, a second outlet within 10 ft. will add no more than 3 dB to the measured sound pressure level. Thus the single NC rating as obtained in Figure 7 is a good reference and allows for additions and subtractions to the rating.

The use of a 10 dB "room effect," while in common practice and accepted for many years, is not as accurate a prediction as is possible using newer techniques. The ASHRAE Handbook and ARI Standard 885-90 present an equation for determining the "space effect" based on both room volume and the distance from the observer to a point sound source:

$$\text{Space Effect} = 25 - 10 \text{ Log (ft)} - 5 \text{ Log (cu. ft)} - 3 \text{ Log (Hz)} \qquad \text{(eq. 2)}$$

Where ft = Distance from observer to source
cu. ft = Room volume
Hz = Octave band center frequency

Table 3 - Space Effect

Space Effect per ARI Standard 885 & ASHRAE Handbook									
Room	Band	1	2	3	4	5	6	7	8
Volume	Hz	63	125	250	500	1000	2000	4000	8000
	@ 5 Ft	-4	-5	-6	-7	-7	-8	-9	-10
2000 cuft	@ 10 Ft	-7	-8	-9	-10	-11	-11	-12	-13
	@ 15 Ft	-9	-10	-10	-11	-12	-13	-14	-15
	@ 5 Ft	-4	-5	-6	-7	-8	-9	-10	-11
2500 CuFt	@ 10 Ft	-7	-8	-9	-10	-11	-12	-13	-14
	@ 15 Ft	-9	-10	-11	-12	-13	-14	-15	-15
	@ 5 Ft	-5	-6	-7	-7	-8	-9	-10	-11
3000 CuFt	@ 10 Ft	-8	-9	-10	-10	-11	-12	-13	-14
	@ 15 Ft	-10	-10	-11	-12	-13	-14	-15	-16
	@ 5 Ft	-6	-7	-8	-9	-9	-10	-11	-12
5000 CuFt	@ 10 Ft	-9	-10	-11	-12	-12	-13	-14	-15
	@ 15 Ft	-11	-12	-12	-13	-14	-15	-16	-17

Eq. 2 yields a range of deductions which differ in each octave band, as shown in Table 3.

The 10 dB room effect which has been traditionally used for diffuser sound ratings, which typically peaks in the 5th band, can be considered to be equivalent to a room about 2500 cu.ft in size, with the observer located about 7 ft from the source.

With VAV terminals, which peak in lower bands, the “10 dB” room size is larger, or the distance is greater.

NC ratings have been common in specifications for a number of years, with an NC-35 being the most common requirement. While NC is a great improvement over previous single number ratings, including “Sones,” “Bels” and dBA requirements, it gives little indication of the “quality” of the sound. A more comprehensive method, RC, has been proposed, and will replace the NC ratings in future editions of ASHRAE Handbooks. In most cases, however, an RC(N) value will result with the same numerical rating as the NC value.

Most manufacturers’ catalog data are based on ASHRAE Standard 70-91, or the ADC 1062 series of standards, both of which require several diameters of straight duct at the approach to the diffuser. Field installations seldom, if ever, allow these ideal conditions.

Direct Duct Connections

When a diffuser is connected directly to a duct, the duct velocity can create unwanted effects in the diffuser. Regions with localized high velocities can create higher than predicted noise levels, as much as 12 NC higher than catalog levels.

Flexible Duct Connections

A number of tests have been conducted to evaluate the effects of flexible duct inlets. The effects of flexible ducts at the inlet on diffusers include increased pressure drop, increased sound levels, and non-uniform air distribution from the diffuser. The test results showed that the effect of different inlet configurations varied for different types of diffusers. The effects are greatest for diffusers with low pressure drop and/or low self-generated noise levels. In general, however:

- The minimum add for an ideal (gentle 90^o) flexible duct connection = 1 NC.
- The worst case add, with a “Kinked” inlet connection = 7-9 NC.
- Air distribution pattern can be greatly affected
- A 90^o Bend and flex adds 0.07 - 0.1 in. Ps., @700 fpm inlet velocity
- Results are not the same for all diffuser types.

Multiple Diffusers

When a number of similar diffusers are located in a single space, an observer may hear more than one diffuser, and the apparent noise level will increase. As a rule, doubling the number of equal noise sources adds 3 dB in each octave band, for an add of 3 NC. For example:

- Two diffusers adds 3 NC
- Four diffusers, equidistant from the observer, add 6 NC
- Eight diffusers, equidistant (unlikely), add 9 NC

Balancing Dampers

A wide open damper can add 3 NC, and 10% pressure drop, depending on the type of damper and diffuser. When adjusted to less than full open, the NC and pressure increase will typically depend on the ratio of the duct total pressure (the sum of the diffuser supply duct static pressure and the connecting duct velocity pressure) to the catalog total pressure for the diffuser. Manufacturers can provide guidance for these effects. This increase can be significant, greater than 10 NC for a duct total pressure

which is three times the diffuser total pressure. High pressure drop diffusers have a lower add than low pressure drop units, at the same flow rate and duct pressure.

Combining Effects

Combining all the above effects, one manufacturer’s catalog reports that:

- Typical 90^o flex inlet will probably add 5 NC to a normally quiet diffuser. Note that plenum slots, which have a side inlet, do not usually experience this condition.
- For an observer in the middle of four diffusers with equal flow rates, add 6 NC.
- If the total pressure in the supply duct is 150% of the diffuser’s required total pressure, add 4.5 NC for the balancing damper generated noise.
- This results in a sound level 16 NC greater than the catalog rating for a single low pressure, low sound diffuser.

Note: These effects are for a “standard” diffuser. Special low-temperature diffusers with high- velocity nozzles are less sensitive to inlet effects, and the add for inlet configurations for these will be less than for conventional low-pressure diffusers.

Procedures

- Use equalizing grids on direct diffuser connections.
- Locate balancing dampers at branch takeoff.
- Keep flexible duct bends as gentle as possible. (Remember, however, flexible duct is an effective attenuator of upstream noise sources, and may be required to quiet them.)
- Keep duct velocities as low as possible, but oversizing can result in higher thermal loss.
- With VAV systems, highest flow only happens at full load. Occupants may need to hear diffusers at full load to be assured system is operating.
- Noisy diffusers work better at mixing air than quiet ones.
- Oversized diffusers may have objectionable drop into the room at low flows (often referred to as "dumping").

Method II. Selection by Supply Jets Mapping

This selection procedure uses the throw values to terminal velocities of 150, 100, and 50 fpm from manufacturers' performance tables. Temperature differences at these terminal velocities are added to the map by using the following equation:

$$\Delta t_x = 0.8\,\Delta t_o\,(V_x/V_o) \qquad \text{(eq. 3)}$$

where:

Δt_x = $t_x - t_c$

Δt_o = $t_o - t_c$

t_x = local air temperature, ° F

t_c = ambient temperature (average room temperature or control temperature, ° F)

V_x = local air speed, fpm

V_o = outlet air velocity, fpm

t_o = outlet air temperature, ° F

Selection by supply jet mapping identifies the portion of the space most likely to be uncomfortable. Portions of a space away from the supply jet will have velocities and temperatures that are nearly equal to the space ambient conditions.

Not all applications result in "overblow", or air that reaches an opposite wall. In some cases the throw terminates with the airstream dropping into the occupied space. This is due to the buoyancy effect between the airstream and space air and/or external forces. Drop must then be considered as shown in Figures 5 and 6 for sidewall outlets and Table 2 for ceiling diffusers.

Mapping Procedures

1. Select type of diffuser. (This may be based on a number of factors, including appearance, performance characteristics, or experience.)
2. Check Table 2 to ensure that the air quantity is less than the maximum.
3. Plot isothermal T_{150}, T_{100}, T_{50} from performance for a selected size and cfm at the throw distances.
4. When selecting a sidewall grille, check configuration in Figures 5 and 6, or similar figures provided by the diffuser manufacturer, for drop during cooling.
5. If the outlet (i.e. slot) provides a horizontal pattern below the ceiling, make sure the drop occurs near the 100 fpm terminal velocity.
6. Repeat processes until all parameters are within recommended limits. Too long a throw may travel down an opposed wall, or collide with opposing jets. Solutions may require relocating diffusers, down-sizing diffusers, or both.

Method III. Selection by Calculating Separation Distance

As the temperature difference between room air and diffuser inlet cold air increases, the throw often decreases and the jet may separate from the ceiling, entering the occupied zone as an unwanted and objectionable draft.

Several methods have been developed to evaluate this effect.

Kirkpatrick and Hassani (1994) have developed a separation distance calculation. As long as the manufacturer catalog throw at T_{50} (throw to a terminal velocity, or measured total air-stream air speed of 50 fpm) is less than the calculated separation distance, the diffuser can be used at the specified flow rate and temperature difference. The Kirkpatrick method includes an assumed variable, C, which is dependent on the location of the heat source relative to the diffuser in the room. Because the location of the heat load is seldom known at the design stage, this method may be unworkable in practice.

Ratz (1996) building on the work of Straub and Cooper (1991), has developed a method to calculate the reduced throw given a colder air temperature difference. A calculation can be made of the drop from ceiling level at the reduced throw. This allows a fuller useful range for the diffuser. The jet can have separated from the ceiling but not have entered the occupied area of a room at T_{50}.

Ratz Method

This method builds on work done by Straub and Cooper (1991). The Froude number (inverse of Archimedes) is calculated from given conditions. From performance data for a specific diffuser, a K factor (third zone throw coefficient at isothermal conditions, derived from the non-dimensional throw plot described in ASHRAE Standard 70-91) is obtained from manufacturer catalog data. The following calculations are used to obtain corrected T50 throw and drop at given cold air conditions:

Given:

- Select diffuser type, neck area, options and specific requirements for isothermal throw and pressure drop.
- Flow rate, Q
- Temperature differential, ΔT
- Ambient absolute room temperature in ^oR, T_a
- Outlet velocity, $V_o = Q/A_o$ or $V_o = 4005*(\Delta P_v)^{.5}$
- Outlet area factor, A_o (directly from mfg. or $A_o = Q/V_o$)
- K' factor (from ASHRAE *Fundamentals* 1993, Chap. 31 p. 9: K' corrected from manfuacturer given or $K' = T*V_T* A^{.5}/Q.$):

Froude's # (Buoyancy number) $N_f = (T_a * U_o^{\,2})/ ((11.58*10^4)*A^{.5}*(\Delta T))$

$T'50 = (K'*N_f)^{.435}* (A^{.5})$

$Drop = (T'_{50} * 22)/(K'*N_f)$

If the drop exceeds the distance between the comfort zone and the ceiling, there may be objectionable drafts.

Central to both the Kirkpatrick and the Ratz method is the Archimedes number (or the inverse, the Froude number). This dimensionless number factors in the room temperature, gravity, temperature difference, outlet area factor, and outlet velocity. Calculations given assume a horizontal ceiling adjacent to the horizontal diffuser.

Tests have shown that room load positions have an effect on throw and drop. A heat load directly under the cold air outlet tends to assist the jet to cling further to the ceiling. A heat load at the far end of the room tends to oppose the jet and cause it to separate earlier. A manufacturer has tested a number of diffusers with a cold air temperature difference and calculated throw zone coefficients at 30^{o}F cooling conditions. The throw and drop at the 30^{o}F condition are then given in catalog tabular form. This saves the user from having to make the detailed calculations needed in the Kirkpatrick or Ratz methods.

Another approach is to simply contact the diffuser manufacturer with a planned cold air application and work with them to get performance data. If needed, a mockup of the conditions can be set up to verify calculations.

Manufacturer catalog data may be used to determine an area factor, outlet velocity (when available), and throw coefficient. In many cases, the outlet velocity may not be easily derived from catalog data, and the manufacturer may have to be contacted.

Method IV. Selection by Comfort Criteria (ADPI), Effective Draft Temperature

ADPI (Air Diffusion Performance Index) statistically relates the space conditions of local or traverse temperatures and velocities to occupants' thermal comfort. This is similar to the way NC relate local conditions of sound to occupants' noise level comfort. High ADPI values are desirable as they represent a high comfort level. Acceptable ADPI

conditions are shown in Figure 3 for velocities less than 70 fpm and velocity-temperature combinations that will provide better than the 80% occupant acceptance.

The temperature and velocity at each measured point are used to calculate an effective draft temperature (Equation 1). A draft temperature of 0 is essentially thermally neutral. Negative draft temperature equates to a sensation of cool, whereas positive values represent a predicted feeling of warmth. The percent of points, measured in the occupied zone, having a calculated draft temperature between -3 and +2, where the air speed is < 70 fpm, results in the ADPI.

The ADPI curves in the ASHRAE *Handbook of Fundamentals* (Chapter 31) summarize some of the tests that established ADPI and the relationships from which this selection procedure originates. The curves show relative comfort for:

- Four different outlet types
- Catalog throw and space characteristics
- Loading (1 cfm/sq.ft with a 20°F differential is a load of about 20 Btuh/sq.ft)
- Flow rate (variable volume)

L is the space characteristic length in feet. This is usually the distance from the outlet to the wall or mid-plane between outlets. This can also be considered the module line when outlets serve equal modules through a space, and all consideration can then be based on the module parameters. (See table 4)

T_{50} is a catalog throw value to a terminal velocity of 50 fpm. A throw value can be selected using a catalog performance table by multiplying the desired throw ratio (T_{50}/L) by the characteristic length (L). The throw ratio is based on a 9 ft. ceiling height. The throw can be increased or decreased by the same amount that the ceiling height exceeds or is less than 9 ft.

To obtain optimum comfort in the space, ADPI tests indicate selecting the outlet from the throw ratios in Table 4**.** These data are reported in the ASHRAE *Fundamentals Handbook*, Chapter 31, Table 2. They are based on a relationship between isothermal throw and ADPI under load. These data represent a typical load of about 20 Btuh/sq.ft, and represent about 20^{o} ΔT at 1 cfm/sq.ft for most applications.

Table 4: ADPI Range

Device Type	Optimum T_{50}/L	Best ADPI	Range of T_{50}/L	For ADPI >:
Grille	1.5	85%	1.0 - 1.9	80%
Round	0.8	93%	0.7 - 1.3	90%
Slots (T_{100}) (T_{50})	0.3 0.5	92% 92%	0.3 - 1.5 0.5 - 3.5	80% 80%
Troffers	1.0	95%	<4.5	90%
4-Way	1.0	95%	1.0 - 3.4	80%

ADPI may be predicted from Table 4, or may be tested using the procedures outlined in ASHRAE Standard 113-90 (See Appendix B, Applicable Standards). This test standard was developed to verify performance for some GSA projects in the late 1970s, and has been used to document the results of many air distribution systems. The validity of the relationship between T_{50} and room length has been consistently verified.

Using this test standard, and modern computer graphics capabilities, room draft temperature profiles have been generated which illustrate well the movement of air and temperature gradients in a room. Figure 8 is just one example of a temperature velocity profile generated using the ASHRAE 113 procedure and a PC-based data acquisition system.

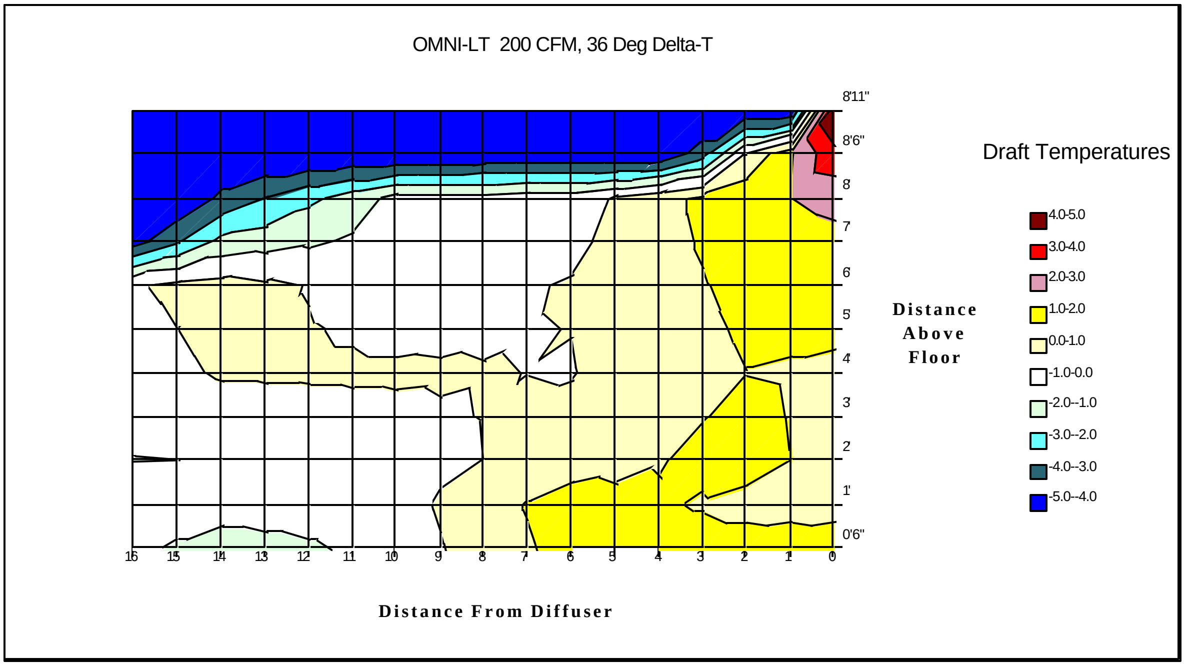


Figure 8
Draft Temperature Profile

The profile in Figure 8 is for a Cold Air Diffuser with very cold air (40^{o}F), and low flow rate (0.2 cfm/sf). The ADPI is 96%.

ADPI vs. Air Change Effectiveness (ACE)

ASHRAE has recently defined a term for describing the mixing of supply and room air, replacing the somewhat ambiguous terms Ventilation Effectiveness and Ventilation Efficiency with a new term, Air Change Effectiveness, or ACE. This term is used in both the upcoming ASHRAE Standard 129P, the Method of Test for Air Change Effectiveness, and in the proposed revisions to ASHRAE Indoor Air Quality Standard 62-89.

The relationship between ADPI and ACE has not been fully evaluated at this time, but the data available at present indicate that if a high ADPI is attained, the ACE will also be high. Furthermore, there have been no reported tests where the ACE was significantly below 100% when cooling from the ceiling. It has been demonstrated, however, that in heating mode the ACE may decrease significantly.

The ASHRAE IAQ Standard, 62-1989, assumes a ventilation mixing, or ACE, of 100% in setting minimum ventilation rates. If it can be shown that the ACE is less than 100%, then the amount of outside air must be increased above the required minimums. With presently available information, when a high ADPI is measured, the ACE is high as well. For this reason, the upcoming draft of the ASHRAE IAQ standard will require diffuser selection based on ADPI in order to ensure acceptable ACE. It is also possible, however, to have a high ACE and a low ADPI, especially if the HVAC system air is supplied directly into the occupied zone. Uniform comfort will not be likely in this case, however.

Cooling Selection Based on ADPI

ADPI is a measure of performance in cooling mode. When heating is employed, the ADPI criteria become overly sensitive to temperature differences due to the very low air speeds present in the space, and as a result ADPI is not recommended for heating evaluations. Interior spaces, however, are predominantly in a cooling mode of operation so this limitation is seldom a problem in interior zone evaluations.

ADPI Selection Procedure, Classical

A typical procedure for using ADPI to select a diffuser size, is as follows.

1. Select type of diffuser.
2. Check Table 2 to determine if the air quantity is less than the maximum.
3. Select the characteristic length from the plans. This is the distance from a diffuser to a wall, or to the centerline between two diffusers, etc.

4. Determine the desired diffuser throw using the characteristic room length and the throw ratio from Table 4.
5. Select a size with a T50 within range from performance table for diffuser at required cfm.
6. Check sound levels for NC compatibility and excessive pressure drop.
7. Ideally, with VAV systems one should now recheck the analysis at the expected minimum occupied flow rate.

This selection will result in maximum comfort for the application. In the event that this selection cannot be made as outlined, supply jet mapping can be used to determine the discomfort areas in the space.

Selecting for ADPI in each room can be tedious, and may not be possible at the time of diffuser selection as the room layout may not be known (moveable partitions and speculative offices lead to uncertainty about final zoning). In addition, when VAV systems are employed, the diffuser must be selected which operates at both design flow rates (typically at designed maximum loads) and at reduced load and flow rate. As an alternative to determining the T_{50}/L for each room and load, it is possible to determine some characteristic curves for different diffuser types based on the airflow rate/unit area. This allows an "operational envelope" to be predicted for different diffuser types.

The process requires that a typical diffuser supply area be defined, and this area be combined against a flow rate/unit area and the diffuser's throw performance, and the result plotted on a graph. When this is done, the envelope of acceptable operation of a diffuser, based on cfm/sq. ft is presented. Detailed examples have been included in Appendix A on 80% ADPI Range Graphs.

An example is shown in Figure 9. In this graph, the x axis is flow rate/unit area, and the y axis is half the separation distance, or L, the characteristic room length. The horizontal curved lines are performance at a given flow rate, while the vertical boundaries are computed from the ASHRAE maximum and minimum T_{50}/L ratios. Performance within the area bounded by the lines should achieve an ADPI of 80% or greater.

While this example was developed with throw at 30^o ΔT, as opposed to the Isothermal throws used in the ASHRAE tables, for this diffuser the difference between isothermal and 30^o Δ is slight, and could be ignored. This is not the case, however, for all cold air diffuser designs, as shown in Appendix A. The graph depicts a calculation of a simple calculation of flowrate/unit area as a function of diffuser spacing (horizontal data) limited by ASHRAE T_{50}/L criteria (Vertical Data), from Table 4, for a given diffuser type.

An example of a design selection is shown with the arrows. Starting at 0.7 cfm/sq.ft, the engineer will proceed upward to the design flow rate (200 CFM) yielding an NC = 35. Then the engineer will proceed to the left to determine the recommended separation distance (arrows). The intersection is between 8 and 9 ft, thus the diffusers should be located 16 to 18 ft apart.

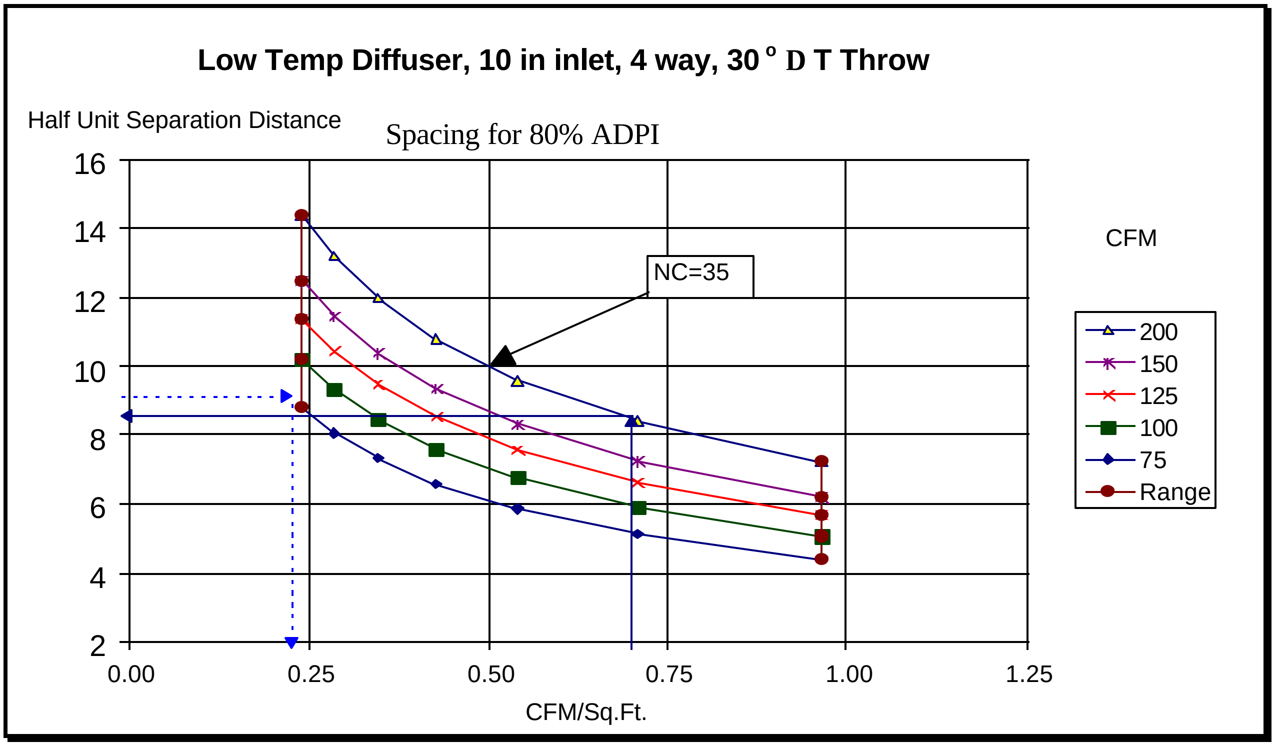


Figure 9
Low-Temperature Diffuser cfm Range for Acceptable ADPI

The engineer can determine the allowable turndown with this design. Tracing the horizontal path to the leftmost edge of the defined 80 ADPI zone, and proceeding down to the X-axis, the minimum cfm/sq.ft can be seen to be about 0.24 with this diffuser (dashed lines). Further explanation of this concept is provided in Appendix A.

ADPI Selection Procedure, cfm/ sq.ft Based

1. Determine the cfm/sq.ft range for the space based on loads and desired supply temperatures. (Minimum flow rate should be based on occupied minimums, not ventilation minimums.)
2. From the 80% ADPI range charts for a given diffuser (some are available in Appendix A), select a diffuser type that works over the range of expected air flow rates, checking at both maximum and minimum occupied rates.
3. Check Table 2 to determine if the maximum air quantity is less than the maximum recommended for the diffuser type selected.

4. Check product performance tables sound levels for NC compatibility, and check for acceptable pressure drop.
5. Determine diffuser spacing from selected chart.

This selection will result in acceptable comfort and air change effectiveness for the application.

Method V. Integrated Analysis

Given time and resources, selection of a diffuser should take into account all the above methods. An integrated, step-by-step selection process should look something like Figure 10.

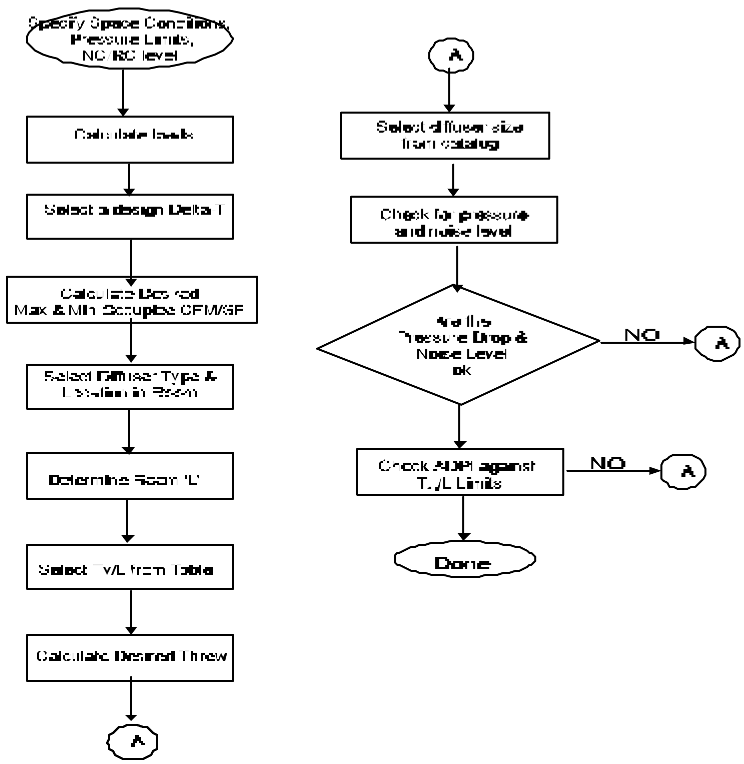


Figure 10
Integrated Step-By-Step Selection Process

Figure 10
Integrated Step-by-Step Process

7

CANDIDATES FOR COLD AIR DISTRIBUTION

A number of diffuser types are considered for delivery of very cold air. These include both conventional and specially constructed diffusers.

Conventional Diffusers

- Ceiling Linear Diffusers. Most linear diffusers have excellent induction, and most should work with cold air. Issues include the potential for condensation on the center tee with opposed (two way) discharge patterns. The Kirkpatric Separation equation predicts early separation with multiple slots discharging in a single direction with many linear diffusers. Therefore single slot, one-way discharge linear diffusers should be used with cold air applications.
- Square Face, Circular Pattern. A radial pattern has been shown to avoid "dumping" at low flows due to its complete 360^{o} jet. At low flows, the jet tends to curl back to the face. A shortcoming is the relatively short throws of these units, which limit coverage at reduced flows. The center cone, or plate, may be prone to condensation near building entrances or during building start-up, where higher humidities may be experienced. Backpans should be insulated if ceiling plenum spaces can have higher absolute humidity levels than the occupied space.
- Perforated Face. The perforated face diffuser typically has short throws. It is also more likely to have "cross" pattern, or four-jet discharge, rather than 360^{o} patterns. Cross patterns are more prone to excessive drop at low flows. The perforated face diffuser, however, will permit higher flows without overthrow, in areas of high load. Perforated diffusers with deflectors in the neck often provide better mixing than those with deflectors at the face. Condensation potential with these should be tested before use in critical areas, as above. Again, backpans should be insulated if ceiling plenum spaces can have higher absolute humidity levels than the occupied space.
- Variable Geometry. Diffusers with variable geometry offer a potential for excellent performance with cold air discharge. Simply increasing the jet velocity inversely to the flow rate, however, is not always sufficient to ensure that the diffuser will maintain high ADPI. Often, optimum performance can be achieved by changing the shape as well as the velocity of the jet.

Cold Air Diffusers

- Linear Multiple Jet Diffuser. This design has excellent proven performance with cold air, and almost no condensation potential. The throw on at least one model, however, has been observed to vary greatly with ΔT, reducing in half from isothermal to full cooling. The selection, therefore, should be made at design ΔT throws, rather than with isothermal air. This performance results from the fact that the jet is aimed slightly below the ceiling, (reducing soiling at the outlet, a problem with other linear diffuser designs) maximizing the jet surface area available for induction of room air. The protrusion below the ceiling is considered objectionable by some, a feature by others.

- Perforated Face Multiple Jet Core. This diffuser has the appearance of a standard perforated face diffuser, but has been proven down to 35^{o}F, with no separation from the ceiling. Throw with these designs is less affected by supply temperature than the linear model, and isothermal throw can be used in the selection.
- Perforated Face High Capacity Multiple Jet Core. Supplied with twice the number of holes, this unit has high capacity, but less turndown, than the standard unit. Again, throw with these designs is less affected by supply temperature, and isothermal throw can be used in the selection.
- Flush Face/Recessed Multiple Jet Core. This is the best performing of the cold air diffusers, in that it has the greatest coverage of all the units, with the best turndown performance. Throw with these designs is only slightly affected by supply temperature, and isothermal throw can be used in the selection.
- Sidewall Upward-Aimed Jet. This device is located in a sidewall, aimed up at the ceiling. In theory this is a good candidate, with limited installations to date. It is limited to locations where walls are available for diffuser mounting.

8

PSYCHROMETRICS (CONDENSATION ISSUES) & IAQ

Using supply air less than 50^{o} F will both drop the humidity of a space, and lower the surface temperature of HVAC system components more than with conventional systems. These two facts must be considered in any CAD system analysis.

Condensation

One of the greatest perceived problems with the use of cold air delivery systems is the increased potential for condensation resulting from colder surfaces in the HVAC system.

For diffusers, the areas of potential consideration are:

- The diffuser back pan which adjoins the ceiling plenum area
- The diffuser head area through which the cold air travels
- Diffuser air deflection plates and vanes

The conditioned, occupied space will almost always be at a lower relative humidity than with conventional air supply systems, because of the reduced moisture carrying capacity of CAD systems. With conventional 55^{o}F supply systems, the lowest possible dewpoint in the supply airstream is 55^{o}F. With CAD systems, the dewpoint drops proportionally with the supply air temperature. The space dewpoint approaches the discharge dewpoint, over time. The time to equilibrium is a function of many variables, including air change rate, air change effectiveness, latent load including latent moisture in the building structure and furnishings, etc. In practice, however, the building relative humidity and HVAC system duct temperatures result in about the same condensation potential for CAD systems as for conventional systems, in most areas of the building.

The use of the ceiling plenum for air return is recommended with CAD systems, to minimize the condensation potential in the plenum. If the plenum is not maintained at the humidity levels of the space (as happens with plenum returns) condensation potentials increase. This, of course, is true with conventional system design as well. The ceiling plenum must be common with the space, on the basis of absolute humidity levels, carefully sealed against air infiltration from outdoors. Negative plenums can draw moisture in from outside, creating moisture problems in walls and ceilings. This could be a very serious problem, and the engineer should consult with the architect whenever a return plenum is used.

In locations near outside doors and entranceways, condensation potentials are greatest. In these areas, it may be necessary to add some reheat to the supply air on very humid days to reduce the condensation potential on the supply air outlets.

However, it should be noted that, based on a survey of a large number of projects using cold air delivery, there are very few, if any, reported condensation problems that would not have been an equal or greater problem with conventional 55 ^{o}F air distribution systems.

Comfort - Satisfaction

Studies have been conducted to evaluate individual comfort on the basis of temperature and humidity given various levels of dress and activity of people; these are discussed in Section 2 of this manual. In addition to the comfort equations predicting that individuals will feel the same with higher temperatures and reduced humidity, there is another variable not covered by the equations. In many cases, a lower humidity will be described as seeming "fresher" than equal thermal conditions at a higher humidity. This results in the apparently better IAQ observed in many CAD facilities. Since IAQ is as often a matter of perception, reducing the humidity almost always increases the acceptance of a space.

Indoor Air Quality

The maintenance of a minimum ventilation rate is recommended by ASHRAE and required by codes in many areas. The strategies to maintain minimum outside air quantities at minimum energy cost are many and constantly developing. The required amount of outside air with a CAD system is, of course, the same as for a conventional system. The percentages, however, are higher for the CAD system. In most designs, this poses no problems, and CAD systems will still use less HVAC energy than conventional systems.

With both CAD and conventional systems, in spaces with low loads, or with high ratios of equipment to latent loads, it may be necessary to raise the discharge temperature to avoid over cooling at part load conditions. This can be accomplished with a number of strategies, including using fan boxes, limited re-heat, supply air reset, and other means. Raising discharge temperatures at an air handler may, however, reduce the dehumidification capacity of the system. The cost of local reheat must be weighed against the loss of dehumidification that results from raising air handler temperatures. Heat transfer at the air handler, including heat wheels, run-around coils, and heat pipes can reduce the limitations on air handler air temperature reset strategies.

On the other hand, the diffusers designed to use cold air always have optimum mixing characteristics, guaranteeing excellent air change effectiveness. If the ACE is expected to be less than optimum, outside air quantities will have to be increased. This should never be a problem with CAD system diffusers.

9

HEATING ISSUES

The ASHRAE *Fundamentals Handbook (1993)* Chapter 31 provides guidance for overhead heating. In short, it recommends that diffusers be located to wash cold walls with warm air, but to limit heating ΔT to no more than $15^{o}F$ to avoid stratification and unwanted drafts at the floor. This limits discharge temperatures to $90^{o}F$ in $75^{o}F$ spaces. It also recommends that the diffusers be located between 2 and 6 ft from the glass and have a two-way pattern, to provide both acceptable heating and cooling performance. These recommendations are based on conventional linear diffusers.

Limited testing with the very high induction cold air diffusers indicates that the rapid mixing of supply and room air results in air washing the glass that is close to room temperature rather than warm air. It is recommended, therefore, that very high induction cold air diffusers be located closer to the glass than conventional diffusers, if the design calls for washing of cold glass with the ceiling supply air.. The performance suggests that higher discharge temperatures can be tolerated with diffusers designed for cold air than for conventional diffusers. This may be a benefit for many gas-fired rooftop units which cannot produce discharge air at less than $100^{o}F$.

A real concern with heating at the ceiling relates to the mixing of supply and room air, especially where ceiling returns are employed (as is most common in offices), and the resultant decrease in air change effectiveness. All the data available on ventilation mixing in a room indicate that the ACE is lower than 1, sometimes significantly so, when heated air is supplied at the ceiling. The upcoming ASHRAE Standard 62 will likely require increased outside air in heating mode to accommodate this fact. CAD diffusers, however, with their rapid mixing, have the potential to provide excellent ACE at all supply temperatures, although there is no published data at this time. This needs to be further verified in tests under the new ASHRAE test method 129-1996 *"Measuring Air Change Effectiveness*", which uses tracer gas in a laboratory environment to determine the ACE of a diffuser under load.

The use of conventional, common heating/cooling systems typically results in cooling air flow rates twice that of heating in all but very cold climates. With CAD systems, however, the cooling and heating air flow rates are nearly identical for many climates, resulting in heating flows at the high end of a diffuser's range, rather than in the middle, increasing the ACE in the space.

10

DIFFUSER QUALIFICATIONS

In addition to many conventional diffusers, a number of special diffusers have been advertised and qualified through tests for application in CAD systems. In determining if a diffuser will be acceptable for a CAD application, there are several things to evaluate:

- Condensation potential. It is difficult to specify or validate the condensation performance of diffusers. There is little hard data on the psychometric conditions in either conventional or CAD buildings. Several areas to check and solutions to consider include:
  - ⇒ Back Pan - Especially if a non-plenum return is a part, of the design, it is recommended that insulation be specified for the diffuser's back pan to avoid condensation in the ceiling plenum.
  - ⇒ Center diffuser blades - The diffuser blades not covered by a jet of supply air are a possible source of condensation. These can be fabricated out of plastic, which has a lower thermal conductivity, or made of aluminum to prevent rust if some condensation does occur under limited conditions.
- **Throw vs. DT.** Many diffusers have a standard reduction in throw of about 1% ^{o}F ΔT between room and discharge, resulting in a typical 20% reduction in 50 fpm throw for 55^{o}F air. This is built into the ASHRAE ADPI T_{50}/L tables. Some diffusers, however, have considerably more throw reduction. This should be verified by testing by the manufacturer, and those which have a high reduction should state the reduction in throw used for design purposes.
- **ADPI verification.** Tests using ASHRAE 113-90 should be conducted at the designed supply temperatures and flow rates to verify that a diffuser fits the ASHRAE ADPI model. If it does not, then the T50/L data should be supplied at the design ΔT.
- **Variable pattern diffusers.** Diffusers which have a variable geometry do not necessarily fall under the ASHRAE ADPI table. Again, ASHRAE 113-90 tests should be conducted to verify whole room air distribution performance under design conditions and temperatures.

What is known about most diffusers, includes the following:

- Typically, insufficient data on ΔT response are available from manufacturers. Most catalog data are based on isothermal performance. In fact, isothermal throw is the basis for the ASHRAE ADPI table. Tests at greater than 20^{o}F ΔT are often difficult to perform with existing equipment.
- Condensation potential is difficult to verify, but can be avoided through cautious design and system control, including soft startups.

- Heating performance should be tested. This is true for all critical heating applications. When the perimeter load exceeds 300 Btuh/ft of wall, it is a good idea to have a documented performance test, regardless of the diffuser type employed.

10

SUMMARY

In summary, cold air distribution is not very different from conventional air delivery. The colder air requires a more careful selection of diffusers, as unwanted drafts can be more objectionable with colder air. The same acceptable design practices that work for conventional supply systems still apply with air colder than 50^{o}. The process developed in the guide will show that some diffuser types are unsuited for VAV Systems and very low air flow rates, regardless of the supply air temperature (even 55 ^{o}F air).

The concept of using throw and room length to predict ADPI is seen as an excellent design tool, made easier by plotting the acceptable operation envelope of a diffuser as a function of air supply rate and diffuser spacing. Other accepted selection processes will work as well. Using the ADPI analysis one can estimate performance at reduced flows, as provided by VAV systems at part load, and selections can be adjusted accordingly.

It has been shown that CAD systems will drop space relative humidities. Occupants typically respond to lower humidities as seeming “fresher.” The concerns over reduced air flows resulting in sensations of “stuffiness,” as a result of lower air motion, have been disproven. At today’s loads, room air speeds are essentially independent of supply air quantity or method, with the exception of task air delivery systems.

Condensation on the diffuser is an element of concern, and especially in critical locations, such as near building entranceways. Special care should be taken, and design considerations may include special systems for these areas. The problem, however, is probably no worse than for conventional 55^{o}F systems. When in doubt about a diffuser’s acceptability, the engineer should perform tests to verify the performance under actual conditions.

Indoor air quality issues include both the reduced total air quantity, which increases the percentage of total room supply air, but not the quantity of outside air, and the reduced RH which both reduces the threat of biological growth and raises the occupants, acceptance of the indoor environment.

Cold air distribution, by reducing the quantity of air to be moved, conditioned, filtered and delivered, promises to reduce energy and increase occupant acceptance. Proper selection will ensure that the promise is kept. For further information on the justification and design of Cold Air Distribution systems, see the EPRI “Cold Air Distribution Design Guide” (EPRI, 1995).

APPENDIX A - 80% ADPI PERFORMANCE GRAPHS

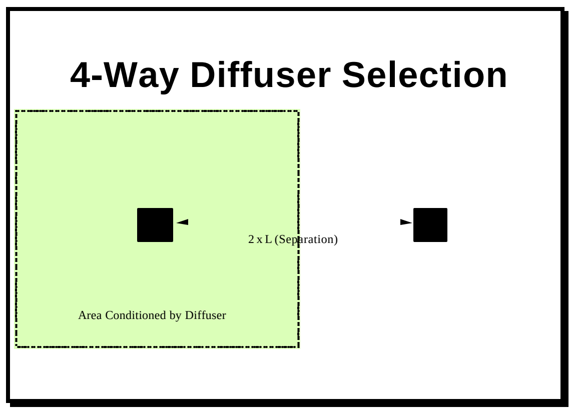
Tables of ADPI Vs T_{50}/L have been derived from the ASHRAE *Handbook of Fundamentals*, Chapter 31, Table 2, and from tests conducted at TITUS's laboratories over the past 20 years.

By analyzing the known performance data for a diffuser, it is possible to calculate the flow rate/unit area vs. the diffusers' throw to 50 fpm. These data can be plotted against the area served, and a series of graphs developed. The minimum and maximum values from the above Table 2 must be adjusted for load.

2X2 Diffusers

For rectangular diffusers with a 4 way discharge pattern, the variable L is 1/2 the distance to the next diffuser, or the distance to a wall. (See Figure A-1).

Figure A-1: 4 Way Diffuser

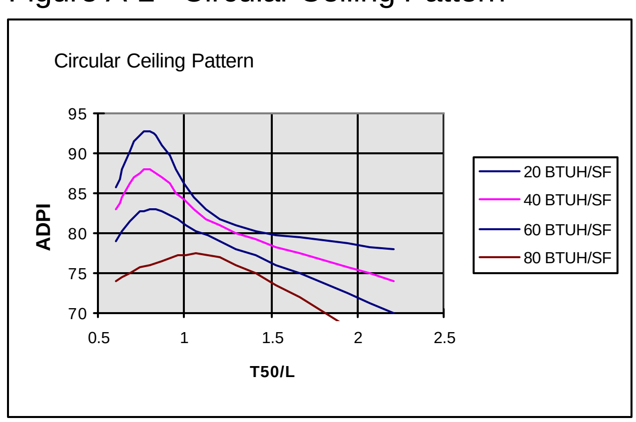


Conventional Diffusers

The effect of discharge temperature on throw is included in the ASHRAE ADPI data, which is based on isothermal throw and ADPI for different room loads (implying non-isothermal supply air). Data are plotted for several TITUS diffusers. Data for diffusers of a similar type by other manufacturers should be similar.

1. The OMNI is a square faced, non-louver diffuser, with a 360^{o} air pattern. The predicted ADPI vs. Throw graph in Figure A-2 shows the response to room load. Using these curves, and catalog (isothermal) throw data, a performance envelope can be generated. Figures A-3 and 4 assume no greater than 20^{o} F Δ T. Colder air temperatures will correlate to higher loads at a given flow,

Figure A-2 - Circular Ceiling Pattern



increasing the minimum allowable flow rate per unit area. The graph depicts a calculation of a simple calculation of flowrate/unit area as a function of diffuser spacing (horizontal data) limited by ASHRAE T_{50}/L criteria (Vertical Data), from Table 4, for a given diffuser type. The indicated NC is for the highest flow rate shown (highest horizontal line) in each graph.

Figure A-3: Omni Diffuser, 8 in neck

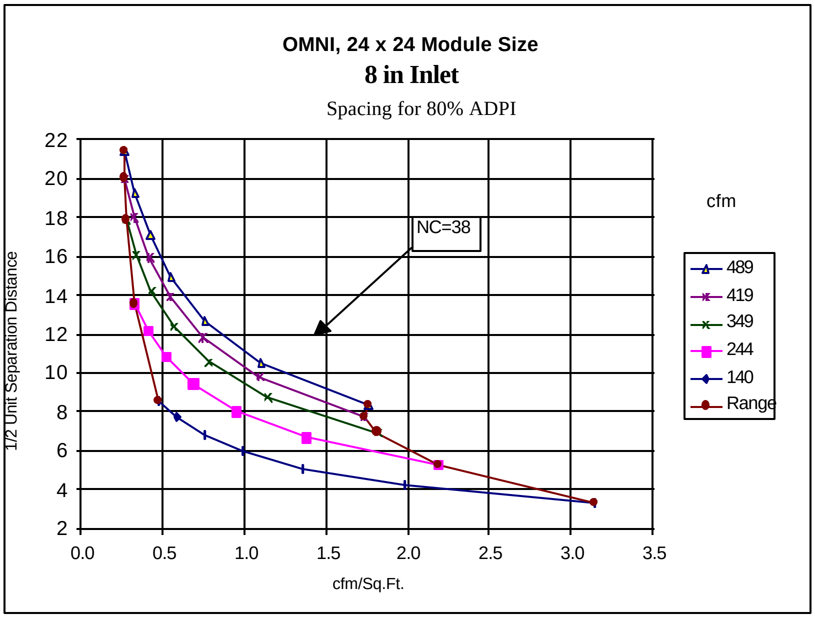
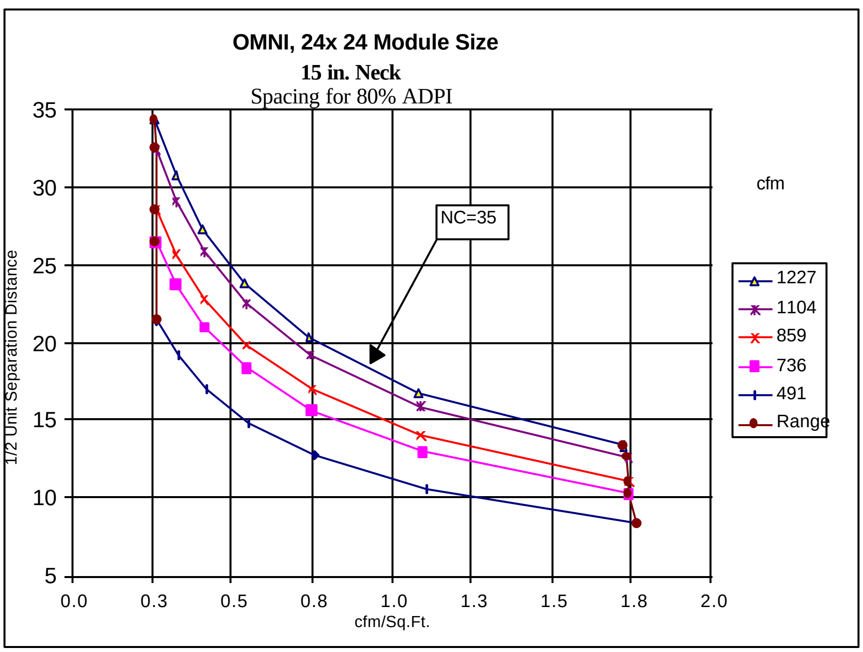


Figure A-4 : Omni 15 In Neck



Plotting data for two neck sizes predicts that the ADPI at flows below 0.3 cfm/sq.ft. will result in unsatisfactory performance, as there is no selection of diffuser spacing and/or flowrate which results in data within the max and min range shown.

2.) The TITUS TMS diffuser is a square multivane, 360^o pattern diffuser with the same ADPI performance curves as the OMNI (Circular Pattern) with a similar performance envelope. (See Figures A-5 and A-6).

Figure A-5: TMS, 10 in. Inlet

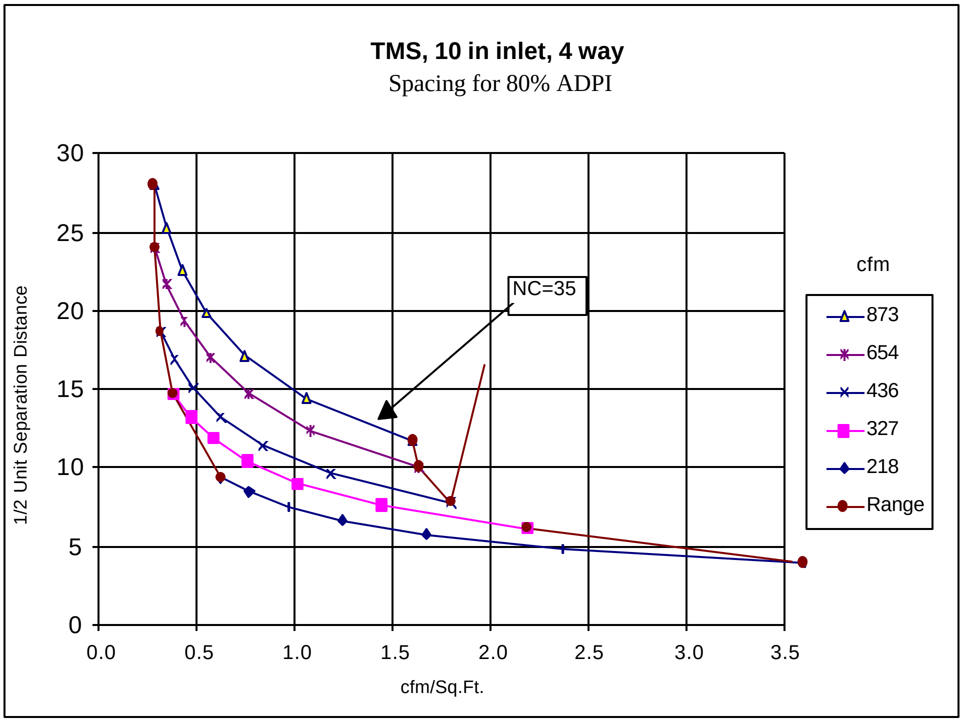


Figure A-6: TMS 14 in Inlet

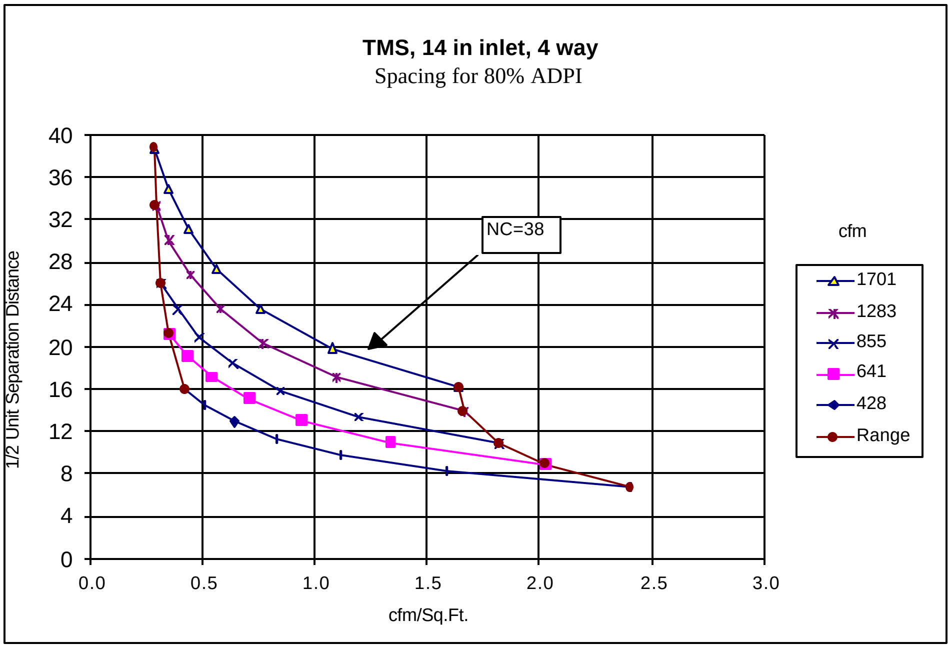
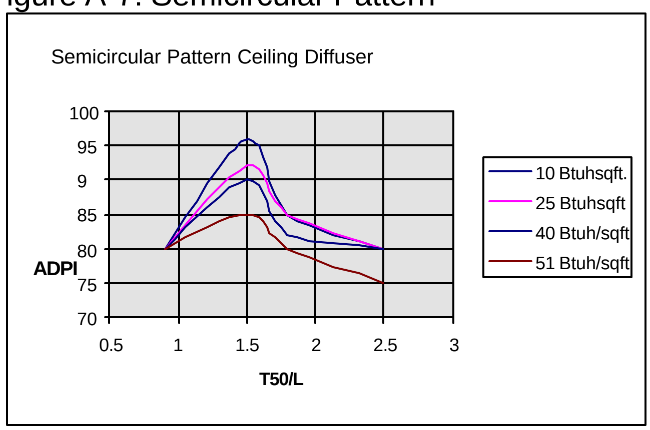


Figure A-7: Semicircular Pattern

3. This TITUS perforated face diffuser is a unit with square deflection elements located at the face of the diffuser. It exhibits a semi-circular pattern, (a mix of circular and cross flow patterns) and has a different set of ADPI curves (see Figure A-7).



The type of diffuser in Figure A-8 has a narrower acceptable ADPI performance range than the ones in Figures A-6 and A-7.

Figure A-8: PAS, 10” Inlet

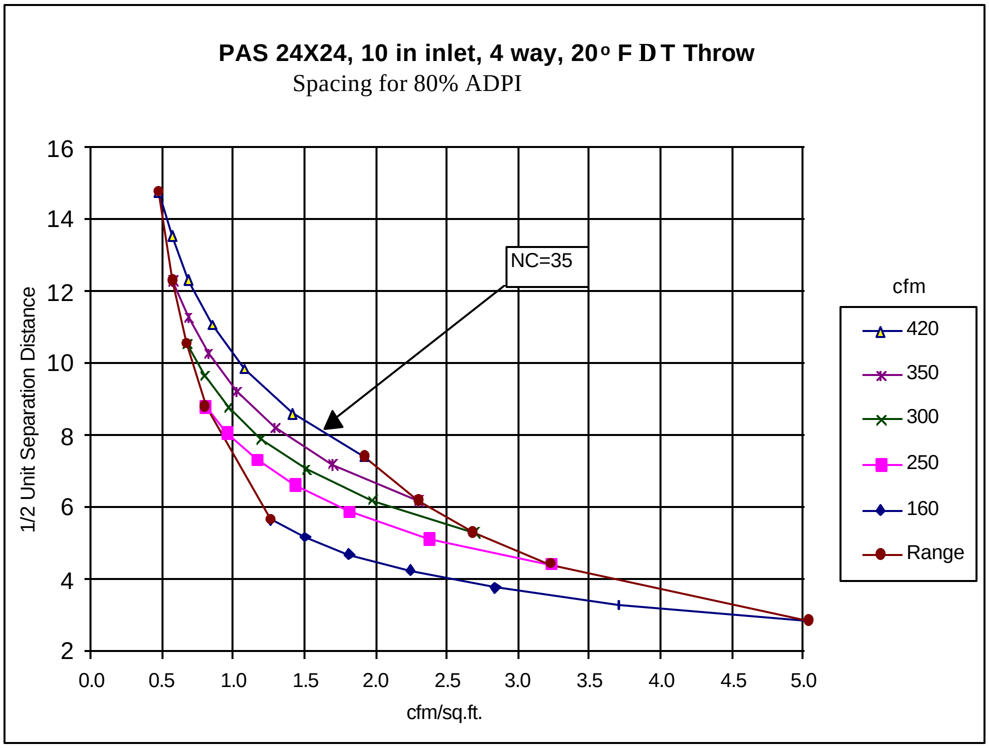


Figure A-9: Cross-Flow pattern ADPI 4.

The Titus TDC is a narrow louver-face diffuser, with a true cross flow pattern, having four jets (see Figures A-9 and A-10).

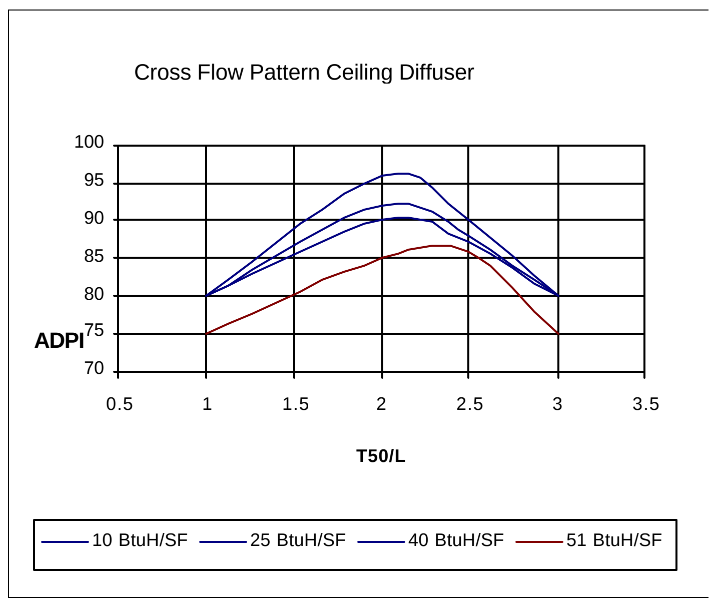
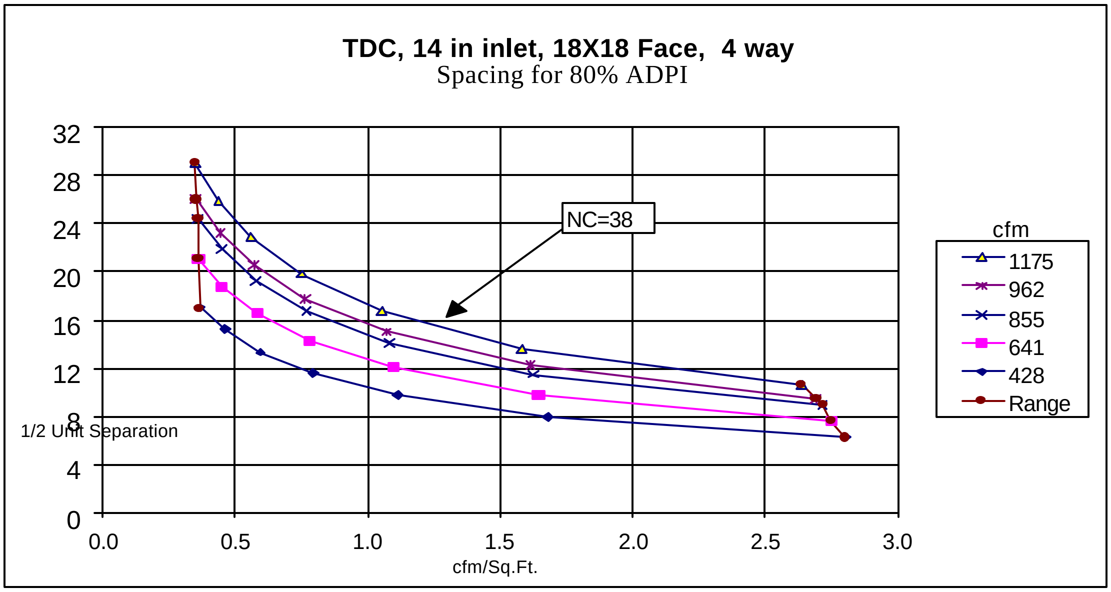
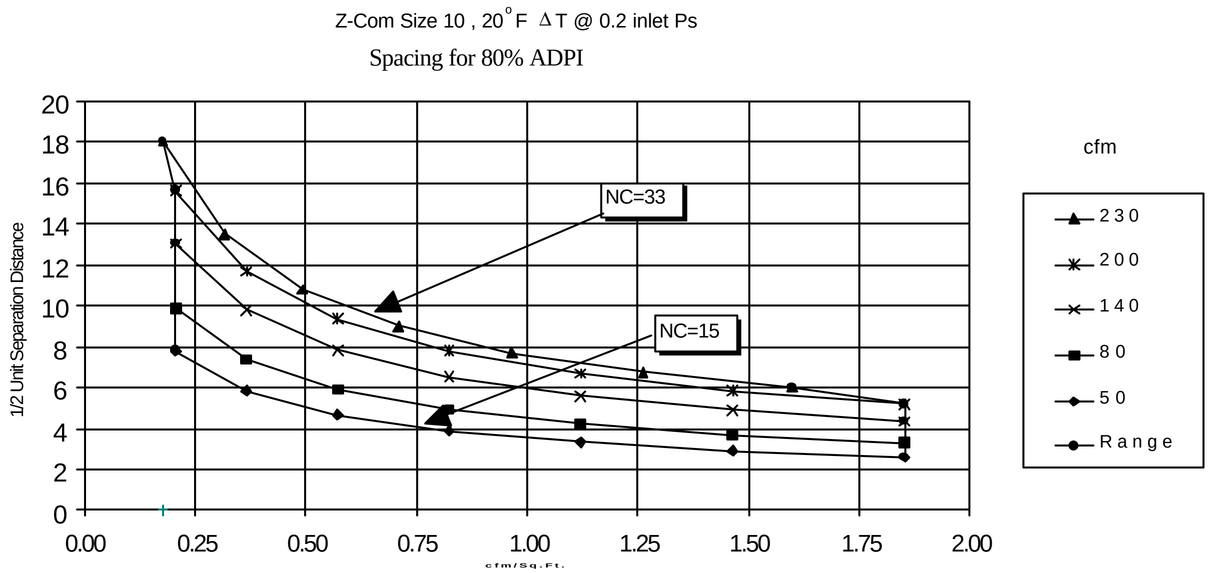


Figure A-10, TDC 14 in Inlet (4 way)



5. The Titus Z-Com is an example of a variable geometry diffuser (see Figure A-11). The jet changes from a radial pattern at high flows to a semicircular jet at low flows. This results in nearly constant throw length over a broad range of flows, and high ADPI's and assumed ventilation mixing - ACE . This diffuser's performance varies with duct pressure. For the example here, a controller duct pressure of 0.2 in is used.

Figure A-11: Z-Com Diffuser



Cold Air Diffusers

High-induction diffusers specially designed for supplying air at a temperature below 50°F provide satisfactory performance down to a much lower flow rate than the standard units. These units usually come with insulated back-pans to reduce the possibility of condensation in the ceiling plenum. Tests have shown that there is very little difference in throw with changes in delta-t for these diffusers. (The Ratz method, verified by tests, is used to determine the effect of discharge temperature differential on throw.)

1. The TITUS OMNI-LT is similar in appearance to the OMNI, but has a high induction (high velocity discharge) thermal core installed in the center of the unit which results in a cross flow ADPI pattern (Figure A-9). Data is shown for both 10° and 30° ΔT. In conducting this comparison, both changes in throw and load are combined. (At reduced flows, duct temperature losses increase, and design discharge temperatures are unlikely with most systems).

With the diffuser in Figures A-12 and A-13, there is a slight response to temperature difference.

Figure A-12: OMNI LT @ 30^o ΔT

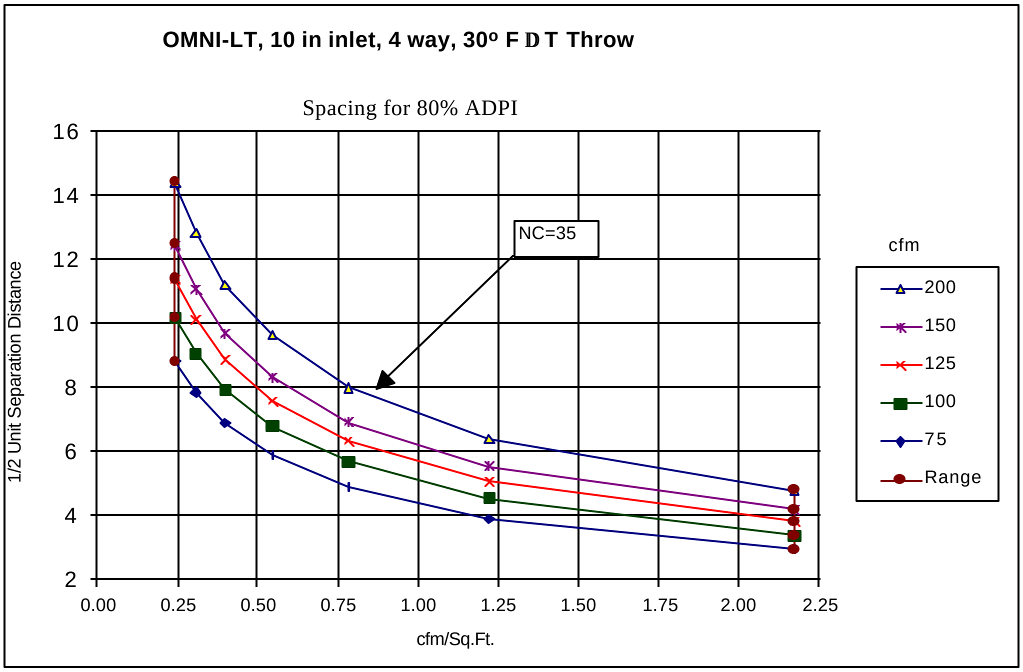
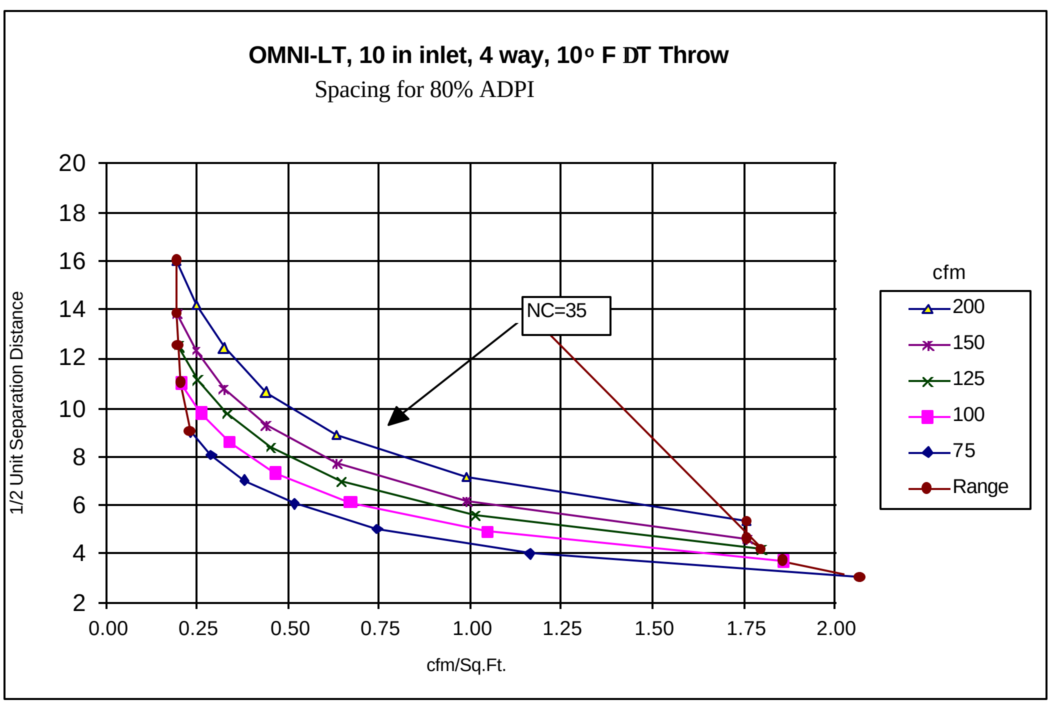


Figure A-13: OMNI-LT @ 10^o ΔT



2. The PSS-LT is a perforated face diffuser with the same core as the OMNI-LT in Figure A-12, and the same cross-flow ADPI response. Again, the unit exhibits a much lower minimum effective flow rate, although not as low as the one in Figure A-12. The unit, however, is more sensitive to ΔT, as seen in Figures A-14 and A-15.

Figure A-14: PSS-LT - 30° ΔT

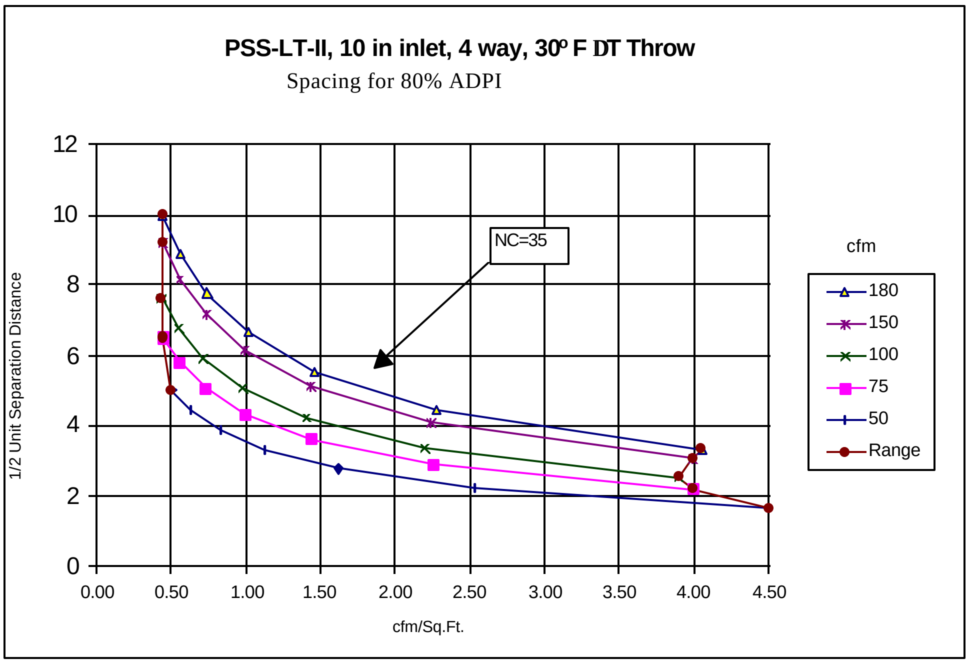
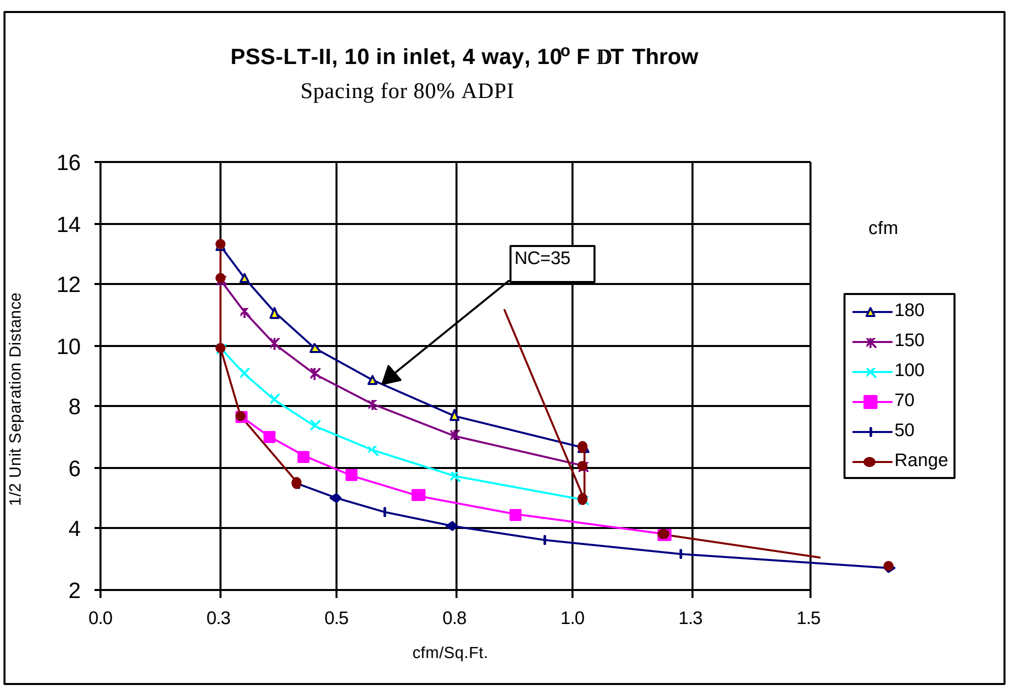
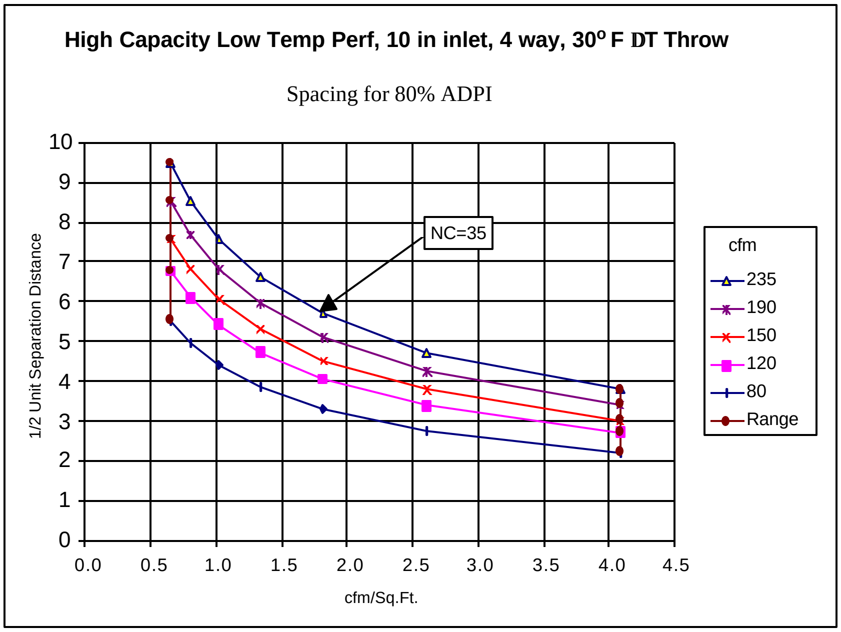


Figure A-15: PSS-LT, 10° ΔT



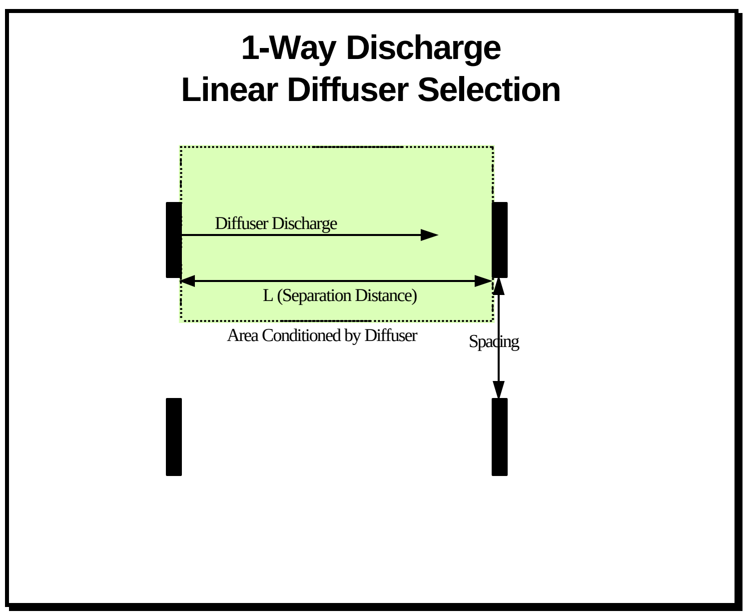
3. A high-capacity low-temperature perforated face unit is available from TITUS (PSS-LTHC) for high load areas served with cold air systems. This diffuser (Figure A-16) has a significantly higher minimum flow point than the standard capacity low temperature perforated diffuser, and should be used in areas of high heat loads where flows below 0.6 cfm/sf are unlikely.

Figure A-16 PSS-LTHC



Linear Diffusers

Figure A-17: Linear Diffuser Areas.

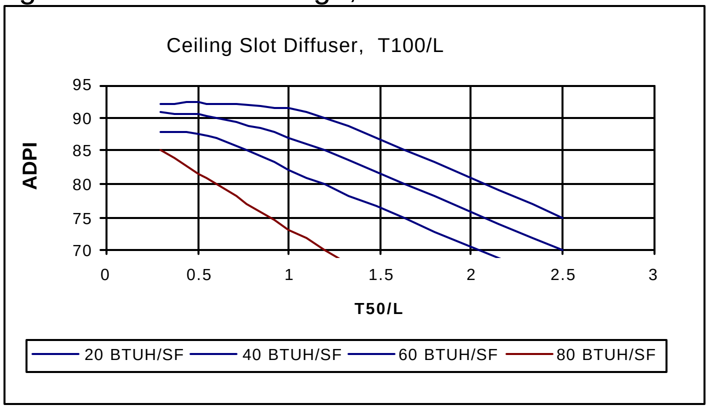


The computation for the 80% ADPI range of a linear diffuser is more complicated due to the rectangular area served by the diffuser. Both the distance between diffusers (or to a wall) and the spacing, end-to-end, must be considered. The example shown in Figure A-17 is for a one-way discharge diffuser (a safe design for cold air, avoiding center 't' condensation potentials). A similar analysis may be conducted for two-way discharge diffusers.

When graphs are prepared, they are done with multiple X-axis data, depending on the assumed diffuser spacing.

The ADPI Range for a Linear diffuser is based on another set of ADPI charts. Linear Diffuser ADPI values are based on T_{100}/L values, rather than T_{50}/L (Figure A-18):

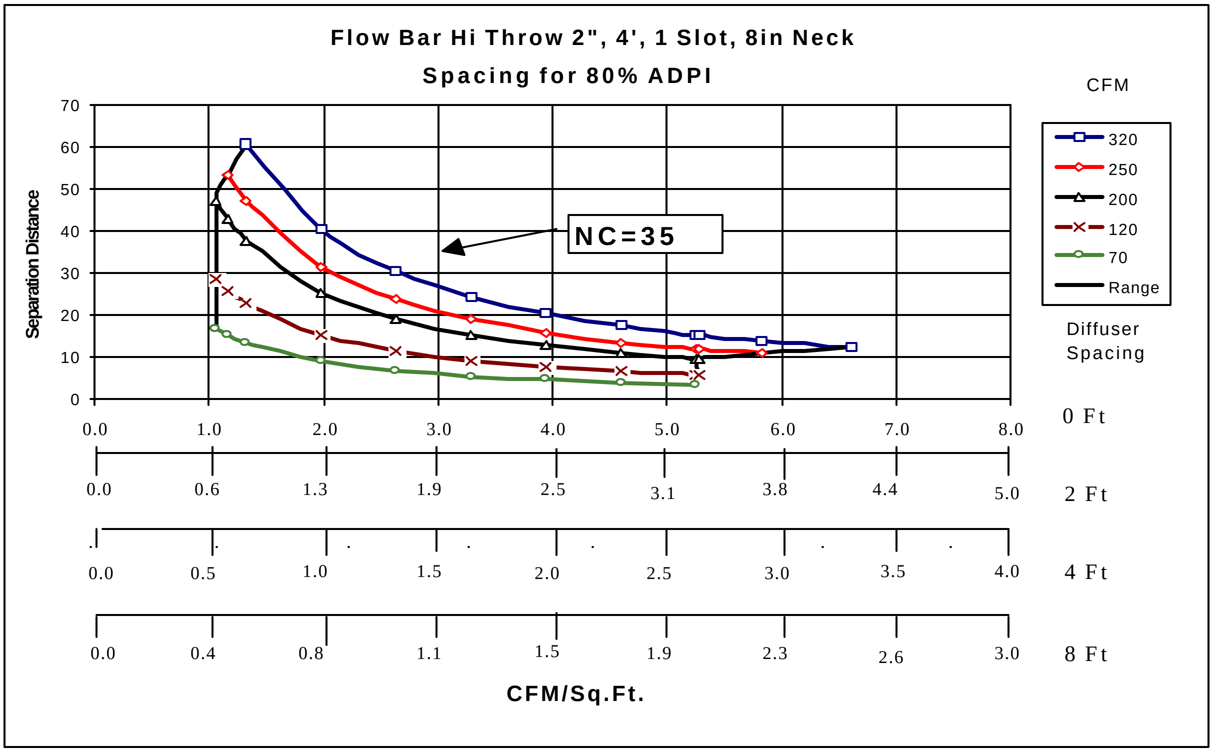
Figure A-18: ADPI Range, Linears



Conventional Diffuser

Data is shown for a conventional linear diffuser, with a two in. slot width, 1 way pattern, horizontal throw.

Figure A-19: 2 in linear diffuser



Cold Air Diffuser

When data are plotted for the TITUS cold air linear diffuser, it can be seen that the 80% ADPI flow limit is much lower than for a standard diffuser. This diffuser also exhibits a significant response to delta-t. Data are shown for both a 30^{o} F and a 10^{o} F Δ T circumstance, again providing data for a high turn down condition, where design delta-t is unlikely.

Figure A-20: Cold Air Linear, 30^{o} F Δ T

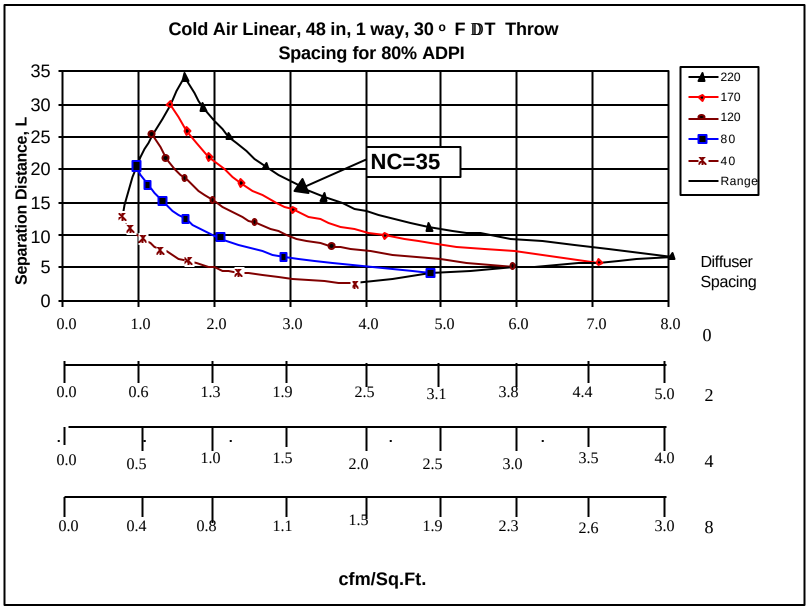
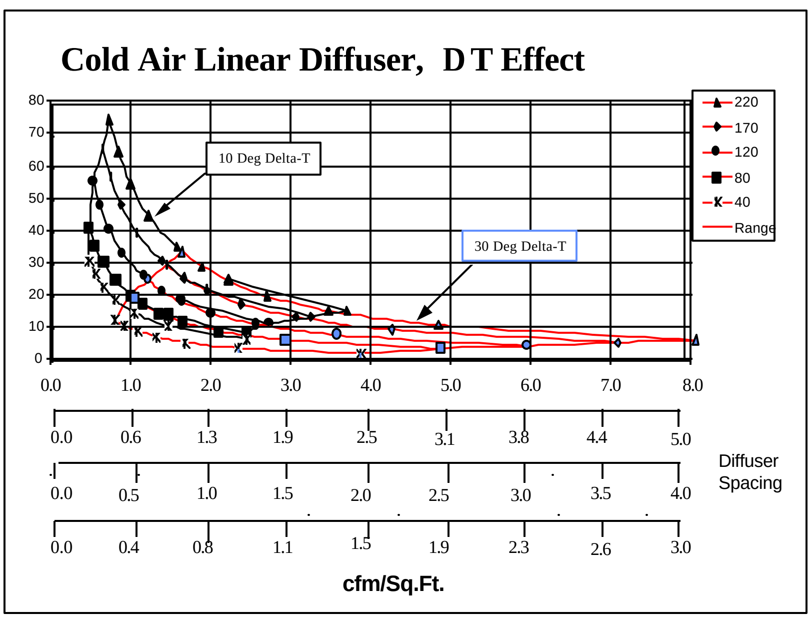


Figure A-21 : Δ T Effect, Cold Air Linear Diffusers



Use, limitations, and notes on the graphs

1. As mentioned earlier, ADPI is primarily a measure of cooling performance.
2. A graph can be used to demonstrate the acceptable range of a particular style of diffuser. In a VAV situation, a diffuser should be selected whose range of operation is within the envelope shown.
3. The graph can be used to determine diffuser spacing in an open office situation, given a known cfm/sq.ft.
4. The graph can be used to determine the lowest flow which will result in acceptable ADPI for a given spacing, with the understanding of the notes.
5. The minimum flow rate is based on the ADPI range from the corresponding table. In practice, a VAV system operating at low loads, and resulting low flows, is not likely to deliver air at design delta-t. This is because of the significant temperature rise that is observed in a low velocity duct. Naturally, the effect is most pronounced with cold air systems. As the delta-t falls (or as discharge temperatures rise), throws tend to increase and ADPI tends to increase as well. (This is especially obvious with the cold air linear diffuser example above.) The minimum cfm/sq.ft. values shown are based on design delta-t, and therefore, should be considered to be conservative in a turndown analysis (but not in a design load analysis). Alternatively, reduced flow points in a VAV application should be judged against a reduced delta-t chart to get a more realistic estimation of performance.

APPENDIX B - APPLICABLE STANDARDS

The following listed standards are used for measuring the performance of air distribution devices. The discussion will include some history, applicability, and relevance to use with cold air distribution devices.

Standards for Grilles, Registers and Diffusers

ANSI/ASHRAE 70-1991

"*Method of Testing for Rating the Performance of Air Outlets and Inlets*" This standard covers all testing for performance, both isothermal as typically cataloged and with hot or cold air. This standard is similar to the now obsolete ADC1062:GRD 1984, which is not currently supported. This standard references ANSI S12.31 for specific acoustical test methods, but modifies the multiple microphone requirement, allowing a single qualified microphone position. The Standard outlines the method of mounting diffusers in the reverberant room. This single position was pioneered by the ADC test code, and verified for diffusers in round-robin tests.

Standard 70-1991 replaces the previous version, 70-1972. The earlier standard did not include acoustical tests. ASHRAE 36B-72 was the acoustical standard, which was dropped by ASHRAE in 1973, in favor of the ASNI Standard. From 1973 till 1991 there was no ASHRAE method of test for acoustical measurements of grilles and diffusers.

<u>CAD Impacts</u>: The standard covers cold air diffusers well, although some cold air diffuser designs exhibit quite a difference between isothermal and full cooling performance. The presentation of data is left up to the manufacturer, and some guidance should probably be given to the recommended presentation method for these types of diffusers.

ANSI S12.31-1980

"*Precision Methods for the Determination of Sound Power Levels of Broad-band Noise Sources in Reverberant Rooms*." This Standard tells how to make the acoustic measurements required in ASHRAE 70-1991. The standard calls for multiple microphone positions in the reverberant room. This standard is also an ISO standard (3741), and is nearly identical to the now obsolete ADC 1062 requirements, except for the microphone issue.

ISO 3741

"*Acoustics-Determination of Sound Power Levels of Noise Sources-Precision Methods for Broad-band Sources in Reverberation Rooms*". This explains actual measurements for the configurations given in ISO 5135 and other ISO standards. This standard is currently equivalent to ANSI S1.31, but a recently approved revision to the ISO standard, requiring a minimum of 6 microphones and a microphone location qualification for every test, may make referencing the ISO standard impractical.

ISO 5219

"*Air Distribution and Air Diffusion-Laboratory Aerodynamic Testing and Rating of Air Terminal Devices*." This is the ISO standard for GRDís, and was the basis for ASHRAE Standard 70-1991. In fact, an attempt was made to adopt ISO 5219 as an ASHRAE Standard, but failed due to some fundamental differences in basic definitions, specifically ISO's definition of Standard Air which differs from ASHRAE's. It references ISO 3741 for reverberant room sound measurements.

<u>CAD Impacts</u> : The ISO standard should not be referenced; rather, the ASHRAE Standard should be utilized, in the United States. The standard will yield the same answers, except for the "standard air" requirement, which could result in a 3 to 4% difference in measured data. As the new ISO 3741 becomes available, however, data will still be applicable, but less practical to obtain.

ARI 890-1993

"*Rating of Air Diffusers and Air Diffuser Assemblies*" The Air Conditioning and Refrigeration Institute has recently developed a Certification Standard for Diffusers. It references the ASHRAE 70-1991 standard for the test method, and specifies rating points and procedures. No diffusers are currently certified under this standard.

<u>CAD Impacts:</u> This standard will cover Cold Air Diffusers, and will likely be a specifiable requirement for CAD projects.

ASHRAE 113-1991

"*Method of Test for Measurement of Room Air Distribution*". This test standard was derived from a test method developed to meet a GSA PBS (Public Buildings Specification) room air motion requirement (air speed greater than 20 fpm at all points) from the mid 1970's. During the tests conducted in these projects it was found that the minimum air motion requirement was impossible to meet; at some point, a 0 air speed was likely to be present, with any air distribution system. Accurate anemometers, including an omnidirectional low-speed anemometer, were developed during these evaluations.

The 113 standard includes all that was learned from these GSA tests, and defines a proven repeatable test method for determining room air speeds and temperature differences. The appendix describes how to use data to develop ADPI and other ratings. Required instrument accuracy prohibits thermocouples, which are typically no better than +/- 0.4 F; the standard requires 0.1C accuracy, radiant shielded. Anemometers must be either omnidirectional or carefully oriented at each test point to the predominant air movement direction.

The standard includes a method of adjusting room temperature swing, a phenomenon difficult to control in practice. The standard may be employed in both laboratories and actual buildings, but is most practical in laboratories due to the need to maintain stable conditions for the time required to gather data, typically a minimum of 30 min. with multiple probes.

<u>CAD Impacts:</u> Many tests have been conducted under this standard, and very repeatable results are to be expected. This procedure provides the best way to evaluate the performance of Cold Air Diffusers in typical operation, and should be required for any evaluation of potential dumping or other comfort and draft evaluations.

ASHRAE 129-1996

"*Standard Method of Measuring Air Change Effectiveness*" . This recently approved standard will utilize tracer gas to determine the mixing of system supplied air and room air in a space using the age of air methodology. This standard was recently approved for publication.

An ASHRAE research project was initiated in January 1995 to conduct the proposed test procedures a number of times in a typical office space to determine the expected variability of the test results, as well to evaluate the practicality of the methods. These tests are complete and indicate that the method is repeatable.

<u>CAD Impacts:</u> With the present emphasis on IAQ, it is expected that this standard will be required to be used to evaluate a number of air distribution designs, including CAD systems, and the data used to apply for credits in reduction of outdoor air quantities. The performance of diffusers under heating raises the most interest, as high ACE test results will avoid mandatory increases in outside air quantities.

ADC 1062

"*Test Code for Grilles Registers and Diffusers*"; Now obsolete, this was one of the original standards dealing with testing and certifying registers, diffusers, dampers, and terminals. Originally developed in 1962 and encompassing boxes and grilles at one time (but never fan powered boxes), several revisions were released through a standard labeled GRD-1984. Tests included pressures, throw, velocity, and sound under isothermal conditions. Also included was a cooling test to correlate isothermal throw with room conditions as a room velocity. It told how to set up, take readings, and determine the number of readings and sizes to test, analyze, interpolate and report.

The Standard required the use of the Anemotherm Air Meter for measurement of room air speeds and throw values. This meter has been reported to be very slow to respond, highly directional, and subject to high error at low air speeds. Throws measured with the instrument required by this standard may be shorter than those reported using faster response, more accurate omnidirectional air speed meters.

The ADC 1062 Test Code has been supplanted by ASHRAE, ARI, ANSI and ISO procedures. At present, there are no ADC certified products or certified laboratories in the United States, and this standard should not be referenced in specifications. The ADC, however, is still active in certifying flexible duct.

CAD Impacts: This Standard should not be referenced in specifications for CAD devices, although it often is. It never covered fan powered VAV terminals, yields different results from the current box testing standards, and may yield short throws for air diffusers.

Standards for VAV Terminals

ARI 880-94

"*Industry Standard for Air Terminals*" (VAV boxes). The purpose of this standard is to establish definitions and classifications; requirements for testing and rating; specifications, literature, and advertising requirements; and conformance for air terminals. It includes an appendix which is the method of test for the rating portion of the standard When ASHRAE Standard 130P is approved, ARI will delete most of the appendix in favor of the ASHRAE standard. This standard was recently revised, and is in print at this time. Manufacturers are working from annotated versions of 880-89 until the printed version is out.

Modifications to 880-89 include mounting details in the reverberant room, and some reduction in options to testing methods. These changes were made to get more consistent data in check tests.

CAD Impacts: The ARI procedure does not emphasize the temperature mixing aspects which are important to CAD systems. Even if the ASHRAE standard is implemented, there is no requirement to report mixing performance.

ARI OM-880

"*Certification Program Operational Manual*". This standard defines the air terminal certification program rules and regulations.

ASHRAE 130-1996

"*Method of Testing for Rating of Ducted Air Terminal Units*". This method of test has successfully completed its second public review, and is resolving comments from that review. A new procedure, it is based on Appendix A of the ARI 880-94 test procedure. It will not, however, state the operating points to be tested, only how data are to be obtained.

CAD Impacts: The ASHRAE standard does cover temperature mixing somewhat better than the ARI standard from which it was developed, and should be specified (when available) for temperature mixing requirements.

ARI 885

"*Procedure for Estimating Occupied Space Sound Levels in the Application of Air Terminals and Air Outlets* ". This standard gives sound transmission losses through ceilings, effects of plenum, loss through ducts, and all consideration of the sound paths to the space. With many items that affect the final sound, it gives a procedure to estimate what the space sound levels will be. Most of the tables are available elsewhere, but a few are unique. These include tables for flexible duct breakout and insertion loss, a ceiling / plenum transfer function and the use of an Environmental Effect, required to correct for an understood error in the acoustical testing rating.

CAD Impacts : The 885 procedure should be utilized for all acoustical specifications.

ISO 5220

"*Air Distribution and Air Diffusion-Aerodynamic Testing and Rating of Constant and Variable Dual or Single Duct Boxes and Single Duct Units*." This standard is based on obsolete ADC test requirements, and was last updated in 1984. It should not be used as a requirement.

ISO 5135

"*Acoustics-Determination of Sound Power Levels of Noise from Air Terminal Devices, High/Low Velocity/Pressure Assemblies, Dampers, and Valves by Measurement in a Reverberation Room*." All of the diagrams and methods shown in the ADC standards are shown in this standard. It was recently updated and circulated for review. In its present form, it requires that an end reflection correction be applied to discharge sound measurements. This will result in data which are as much as 10 dB higher in the second band than data conducted under the ASHRAE 130/ARI 880 procedures. Until this difference is resolved, it should not be referenced.

ISO 7244

"*Air Distribution and Air Diffusion-Aerodynamic Testing of Dampers and Valves*." Another currently obsolete standard.

Related Standards

ASHRAE 62-1989

"*Ventilation for Acceptable Indoor Air Quality*" The purpose of this standard is "To specify minimum ventilation rates and indoor air quality that will be acceptable to human occupants and are intended to avoid adverse health effects." This standard is being modified by a review committee. There is much debate over the new standard's direction, and it remains to be seen what will result.

The current draft, however, requires that diffusers be selected to result in acceptable ADPI in accord with the Fundamentals Handbook requirements for throw vs. room length (Chapter 31, Table 2).

The standard is being written in code language, and the review committee is a standing committee, which stays in place even after a revision is issued, and personnel roll on and off the committee.

<u>CAD Impacts</u>; A major concern for the use of Cold Air Diffusers and designs is the reduction in air flows and subsequent increase in the percentage of outside air. The final requirements for outside air may have a major impact on the use of cold air systems. If outside air requirements increase even slightly above current requirements, cold air supply temperatures may have to be adjusted upward to avoid overcooling spaces at minimum ventilation rates.

ASHRAE/IES Standard 90.1-1989

"*Energy Efficient Design of New Buildings Except Low Rise Residential Buildings*." This standard will be out for public review shortly. The new standard will include both new and renovated buildings, and will likely prove to be very controversial.

The current public review draft has a prohibition against Series Flow Fan Powered terminals, except to meet minimum ventilation standards, such as when ventilation air flow rates exceed cooling flow requirements. It also proposes to make some prohibitions on the size of fans in parallel fan boxes. This draft will be a long time in reaching approval, and will no doubt differ from the proposed draft in a number of ways. This standard is being written in code language for easier adoption by code bodies by a standing standards committee. Standing committees allow continual maintenance of Standards, which means they can be more responsive to changing needs.

<u>CAD Impacts;</u> The potential impacts of this standard on the use of cold air systems are severalfold. The proposed prohibition of series fan terminals, and limitations on parallel boxes, will require direct injection of supply air into spaces, which when tied

to Standard 62 proposals to increase minimum ventilation rates in some circumstances. This can effectively prohibit the use of cold air supply.

ANSI/ASHRAE 55-1991

"*Thermal Environmental Conditions for Human Occupancy*." This revision to the ASHRAE Comfort standard still does not reference the PMV/PPD concept covered in the ISO 7730 standard, but yields the same environmental conditions. The standard states there is no minimum air motion for comfort. The new revision committee, just formed, is a standing committee, and the next revision will be written in code language. As the new chairman was partly responsible for the PMV/PPD concept, it can be assumed that PMV will be included in the next revision.

CAD Impacts : The comfort equations, especially PMV/PPD equations, favor the use of cold air systems by allowing quantitative analysis of the comfort benefits of low humidity.

ISO 7730-94

"*Moderate Thermal Environments - Determination of the PMV and PPD Indices and Specification of the Conditions and Thermal Comfort*"; This standard defines thermal comfort using the PMV / PPD single number rating. It also includes a program listing, in FORTRAN, which allows computation of the PMV/PPD indices.

CAD Impacts: This standard can be used to effectively demonstrate the beneficial effects of reducing humidity on occupant comfort. The single number rating (PPD) allows a simple comparison between conditions, and is better than ASHRAE-55 in promoting the comfort benefits of cold air.

APPENDIX C ANNOTATED REFERENCES:

ASHRAE Handbook of Fundamentals, 1993, Chapter 31. This chapter was recently rewritten and is the basis for much of the ADPI selection process. See Appendix B

ASHRAE Std. 55-1992 "Thermal Environmental Conditions For Human Occupancy," ASHRAE. This Standard is the American thermal comfort standard. See Appendix B

ASHRAE Std. 70-1991 "Method of Testing for Rating the Performance of Air Outlets and Inlets" This standard covers all testing for performance of diffusers, both isothermal as typically cataloged,and with hot or cold air.

ASHRAE Std. 113-1990 "Method of Test of Room Air Distribution" ASHRAE. This is the basis of measurements for validation of ADPI. See Appendix B

ASHRAE Std. 129-1996 "Measuring Air Change Effectiveness" ASHRAE. This standard covers the use of tracer gas methods to determine the mix of supply air to the occupied zone of a room. It is primarily a laboratory standard, and will be difficult to perform in actual buildings.

Benton, C.C., et al. "A field measurement system for the study of thermal comfort," ASHRAE Transaction, V96, Pt1., AT-90-6-5, 1990. A description of a modern portable system for validation of comfort models in spaces.

Bauman, F., et al. "Air Movement, Ventilation, and Comfort in a Partitioned Office Space," ASHRAE Transactions, V.98, Pt1, AN-92-4-4, 1992. Award winning paper on comfort in partitioned spaces. Concluded that comfort was easily achieved in partitioned spaces with proper diffuser selection, those meeting ADPI guidelines of Chapter 31, above.

Berglund, L.G. "Comfort Benefits for Summer Air Conditioning with Ice Storage," ASHRAE Transactions, V97, Pt 1, NY-91-15-2. An excellent discussion of the benefits of reduced humidity and perceived occupant comfort.

EPRI, "Cold Air Distribution Design Guide",1995. This manual provides a definitive analysis of Cold Air Distribution System design. It covers all aspects, including some on air distribution, of the Why's and How's of a CAD system. This manual is companion to the EPRI document.

Gregerson, John" Setting Sounder Standards for Acoustical Performance," Building Design & Construction, April, 1992, PP 58-60. A discussion of speech privacy and desired sound levels in offices, leading to acoustic quality issues.

Hassani, Vahab, et al. "Indoor Thermal Environment of Cold Air Distribution Systems," ASHRAE Transactions, 1993. Details of cold air diffuser tests using several techniques.

Houghten, F.C., et al.,"Draft Temperatures and Velocities in Relation to Skin Temperatures and warmth,"ASHVE Transactions, 44:289, 1938. This is the theoretical basis for much of the ADPI draft tmperature calculations.

Int-Hout, D. and G. Hart, "The Performance of a Continuous Linear Diffuser in the Interior Zone of An Open Office Environment," ASHRAE Transactions, 1980, V86 Pt1. Early work on validation of ADPI selection guides with commercial diffusers. Indicated that room air motion was not as dependent on jet characteristics as on room load.

Int-Hout, D. and G.Hart, "Performance of a Continuous Linear Diffuser in the Perimeter Zone of An Open Office Environment," ASHRAE Transactions, 1980, V86, Pt2. One of three papers used as the basis for ASHRAE Fundamentals' Chapter 31 perimeter heating recommendations.

Int-Hout, D. "Measurement of Room Air Distribution In Actual Office Environments To Predict Occupant Thermal Comfort," ASHRAE Transactions, 1981, V87, Pt2. Description of the test procedure that eventually became ASHRAE Std 113.

Int-Hout, D. "A Measurement System For Determination of Air Diffusion Performance in Interior Spaces," ASHRAE Transactions, 1981, V87, Pt2. Description of an early ADPI measurement system. Now superseded by linearized anemometers.

Int-Hout, D. "Analysis of Three Perimeter Heating Systems by Air Diffusion Measurements," ASHRAE Transactions, 1982, V88, Pt1. Investigations of heating in perimeter environments, including radiant panels. Verified ASHRAE's delta-t recommendations in heating.

Hart, G. and D. Int-Hout, et.al. " Lighting, Acoustics, and Air Diffusion; Evaluating the Design of a Passively Daylighted Building Through Performance Testing In A Full Scale Mock-Up," ASHRAE Transactions, 1983, V89, Pt1. Full scale mock up tests validating ADPI predictions, and using the predecessor to ASHRAE 113 as a procedure. Also included IAQ and lighting measurements.

Int-Hout, D.,"Air Distribution for Effective Ventilation," Proceedings, Washington Energy Extension Service Seminar, 14-15 November, 1986. A discussion of air change effectiveness, then called ventilation effectiveness, and ADPI. Diffuser selection is discussed.

Int-Hout, D., "Thermal Comfort Calculations / A Computer Model," ASHRAE Transactions, V96, Pt 1. (1990). This paper presents a compromise method for predicting thermal comfort using both the Pierce 3-Node and Fanger PMV methods in a single model. It was developed for ASHRAE Standard 55 but never used.

Kirkpatrick, A. and V. Hassani, Cold Diffuser Performance **Research Project 3280-29, 1994.**

Lorch, F.A., and H.E. Straub "Performance of Overhead Slot Diffusers with Simulated Heating and Cooling Conditions," ASHRAE Transactions, 89:1 (AC83-04, #5). The second of three papers which were the basis for the heating recommendations in the ASHRAE Handbook.

Miller, P.L.; "Diffuser Selection for Cold Air Distribution," ASHRAE Journal, pp32-36, September, 1991, ASHRAE. A reanalysis of the 1960's ADPI studies at Kansas state, using cold air as an input. Miller validated the ASHRAE recommendations for CAD systems.

NBSIR 83-2746 , "Strategies for Energy Conservation for A Large Office Building," U.S. Dept. of Commerce, Wash. DC, 1983 An early discussion of, among other issues, the benefits of cold air in Federal buildings.

Nevins, R.G. and E.D.Ward,"Room Air Distribution with an Air Distributing Ceiling," ASHRAE Transactions, 74:V1.2.1, 1968. This is an excellent guide to the theory of jets in rooms.

Ratz, R. - "Calculating Throw and Drop with Cold Air",TITUS, 1996. Paper published 1996 detailing method to calculate adjusted throw and drop with cold air based on isothermal manufacturer catalog data.

Rousseau, W.H.; "Perimeter Air Diffusion Performance Index Tests for Heating with a Ceiling Slot Diffuser," ASHRAE Transactions 89:1 (AC83-04, #4). The third paper used as the basis for the heating recommendations of the ASHRAE Fundamentals Chapter 31 on Air Distribution.

Schiller, G.E., "A Comparison Of Measured and Predicted Comfort in Office Buildings," ASHRAE Transactions,V96 Pt 1, 1990. The first of three studies on comfort in buildings. Typical met rates were found to be about 1.1, and typical clothing levels less than 1.0. The study proved beyond a doubt that 68-78F guidelines were not conducive to productivity or comfort in offices. 73-75F is a preferred comfort range.

Straub, H.E & Cooper, J.G.; "Space Heating with Ceiling Diffusers," Heating Piping & Air Conditioning, May, 1991, PP49-55. This paper goes further in defining the problems with overhead heating. A must for heating design.

Weed, J. & D. Int-Hout "Throw, the Air Diffusion Quantifier," ASHRAE Transactions, V94 Pt 2 1988. A discussion of how throw is measured and converted to catalog data, and how instruments are used to measure throw.