
July 2007
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INSIDE: Air Jet Dynamics for Diffusers
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Air Jet Dynamics of Air Diffusers

BY DAN INT-HOUT III, P.E., *Chief Engineer, Krueger, Richardson, Texas*

One of the most difficult HVAC parameters to quantify is the air distribution within a space.

Heating and ventilation systems are designed to provide a degree of thermal comfort to the occupants of a building. The ability of a system to meet minimum comfort levels is determined by a number of factors, such as supply air quality, control systems. But the most difficult parameter to quantify is air distribution within a space. Even with carefully designed systems, air movement is affected by an inexhaustible number of factors: furniture placement, diffuser adjustments and actual system operation.

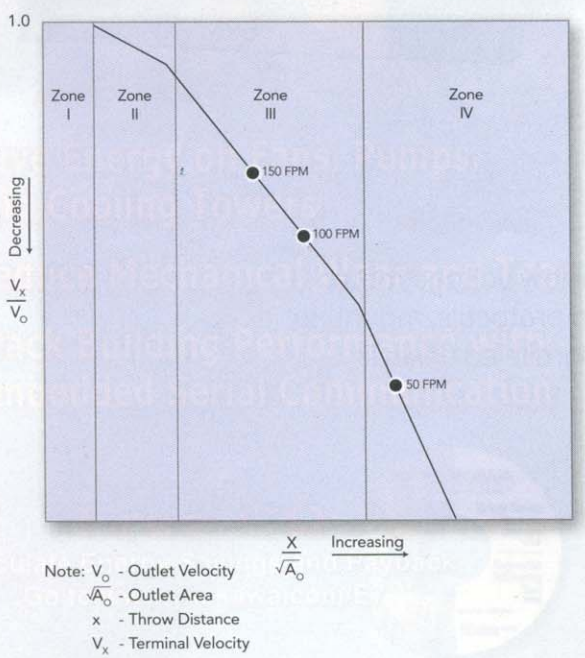

Figure 1 - Throw tests plotted on non-dimensional log-log chart

The most common type of air diffuser for office spaces employs a jet of air at the ceiling that induces room air and mixes the room and supply air above the occupied space. Under ideal conditions, the result is low velocity air movement within the occupied spaces and uniform temperatures throughout the zone. Most research shows, however, that the air speed in the occupied zone is more of a function of convection currents generated by room loads rather than by the air delivery system.

A number of research projects conducted during the past 30 years have shown that air patterns and temperature distributions within a space are distinctly related to discharge velocity, supply air volume, room-diffuser temperature difference, room dimensions, and type of diffuser employed. Essential to this analysis is a performance parameter of the supply air diffuser known as "throw," which is the distance from the diffuser at which the air jet has slowed to a stated terminal velocity.

The *ASHRAE Handbook - Fundamentals* (2005) provides a table that correlates the isothermal throw of a number of types of diffusers with air diffusion performance index (ADPI) or over a range of room loads. Using this table, a design engineer can determine the performance range of a selected diffuser, in terms of ADPI, which is related to predicted occupant comfort. In earlier releases (LEED-NC version 2.1) of the USGBC LEED certification process, it was suggested that use of the ADPI tables, and diffuser performance, could determine room air change effectiveness (ACE). Subsequently, however, research with ASHRAE Standard 129-2002 showed that with cold air coming from ceiling diffusers, the room ACE (or percent of ventilation air reaching room occupants) was always close to 100%. With subsequent releases of the LEED requirements (version 2.2, Jan. 2006), ACE is defined by ASHRAE Standard 62.1, Table 6.2, and ADPI analysis is no longer used in this manner. ADPI is a valuable tool, however, in predicting room temperature uniformity, and may be used to predict compliance to ASHRAE Standard 55 (Thermal Comfort), and assists the engineer in obtaining the LEED-NC EQ 7.1 credit for occupant comfort.

Nonstandard Applications

Often, one needs to determine jet performance for an unconventional discharge jet pattern, something that is required by an architect to hide the air outlet, or in retrofitting an existing building while preserving architectural details. The throw performance for a sidewall grille often can be used as a good starting point, with 0°F blade deflection. Mass often rules, so discharge dimensions are often not a primary driver as long as discharge velocities are low (<500 fpm). One has to be careful to ensure that if the jet is entrained to a surface, entrained jet performance is used. As mentioned, a free jet will have about 30% shorter throw.

Many times, it is necessary to mock up a situation, if it has not been encountered before. Many manufacturers have this capability, and will create a mockup of critical applications. When in doubt, mock it up.

Air jets and diffusers

Air jet performance is well documented, but many of these studies have little applicability to diffusers operating in indoor spaces, as they often deal with high velocity jets operating in open spaces. Air jet performance in interior spaces is perhaps best summarized by Ralph Nevins in his book *Air Diffusion Dynamics* (1976) and in the *ASHRAE Handbook.*

A jet of air discharged from a duct will expand and lose velocity in accordance with several physical principles. Conservation of momentum will dictate the total energy dissipation, as the mass times velocity (MV) available as the air is discharged is the available energy, and the mass of the airstream increases due to entrainment of adjacent room air.

Diffusers produce a specialized form of jet. Typically, a room air diffuser has the jet directed along a flat surface, such as the ceiling or wall. A diffuser often has either a radial or a four-jet pattern, and the jet is very thin and wide. The result is that classical jet theory must be modified for application to diffusers. The surface prevents induction along one side of the jet, and at the same time, causes the jet to stick to the surface. This attraction is often called the coanda effect, and helps keep cold supply air from falling into the room. The reduced induction results in greater throw than with a free jet.

Air discharged from a diffuser that is different from room temperature will be affected by buoyancy. Cold air will tend to fall off of the ceiling, reducing horizontal throw. It also will tend to increase downward vertical throw. Heated air will tend to travel farther along a ceiling, and project a shorter distance with downward vertical throw. A good rule of thumb for the extent of this effect is to apply a 1% per °F Delta-T factor to calculated throw data, in the direction indicated in Figure 1, for terminal velocities of 75 fpm. Throw velocities above 150 fpm, on the other hand, are not significantly affected by Delta-T.

Diffuser throw and air flow

Calculation of diffuser throw performance is a complicated procedure. One factor used is a calculated, or sometimes measured, discharge velocity (V_k). The ratio of supply air flow rate to discharge velocity is called the area factor (A_k). This value is used in plotting the throw data and in calculating the throw for catalog purposes. At one time, it was published by manufacturers, and then used to determine the diffuser's airflow rate for balancing purposes. The size and configuration of the sensing element is almost as critical as the accuracy of the velocity meter, because the discharge jet of most ceiling diffusers is very thin and typically is concentrated in a narrow envelope. The proliferation of electronic airflow sensing devices, and the myriad ways these devices respond to this narrow high velocity jet has made it impractical to publish an area factor that is valid for all types of devices.

In practice, a capture hood is used to determine airflow rate, and then one can determine the unit's on-side A_k, which can then be used for balancing systems where the main concern is repeatability.

For throw calculations by manufacturers, however, a "virtual" A_k is often employed. This requires only that the A_k value used to develop throw parameters be consistent for all calculations, as will be seen later. Most manufacturers today do not present A_k factors. When they do, it is valid only for a single velocity meter at specified measurement locations.

The Delta-T of the system during throw tests can have a significant effect on both types of measured performance. Air jets can be significantly affected by buoyancy. Cold air tends to settle toward the floor, and hot air rises. For this reason, most throw data is determined where the supply jet and average room air are the same. For testing purposes, this means that the difference between supply and room air temperatures cannot exceed 2°F. In addition, ADPI predictions using Table 2 in Chapter 33 of the *ASHRAE Handbook - Fundamentals* are based on isothermal throw.

When analyzing throw measurements, maximum room-air speeds and distances are plotted on a special non-dimensional log-log chart, which then is used to prepare a performance or catalog sheet. The results of throw tests are plotted in a prescribed fashion (see Figure 1). The graph has the following axes:

X Axis = $T/(A_k)^{-0.5}$ where:

T = Throw in ft.

$(A_k)^{-0.5}$ = Square root of the area factor in sq. ft. or $(\text{Volume/Discharge Velocity})^{-0.5}$

Y Axis = V_t/V_k; where:

V_t = Measured terminal velocity in fpm

V_k = Discharge velocity in fpm

The highest air speed measured at intervals out from the diffuser is plotted on the throw graph. The data will form a curve on the graph. Most standards then suggest that this curve be intersected with three

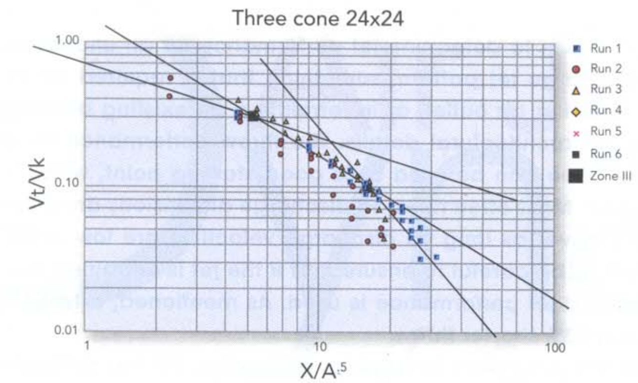

Figure 2 - Straight-line approximations

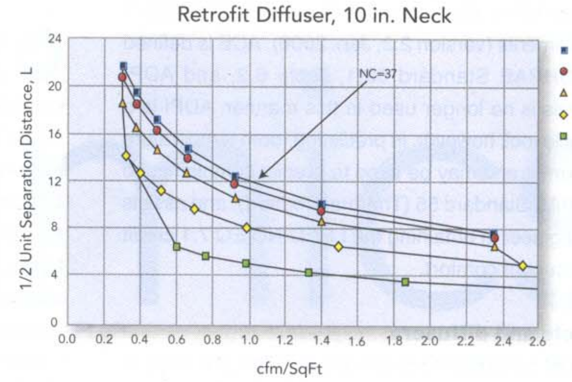

Figure 4 - Test conducted with isothermal conditions

straight line plots, labeled Zones II, III and IV. These lines have slopes of -0.5, -1 and -2 respectively, and reflect the zones of jet/room air interaction. Zone II is typically a zone of very high induction, Zone III a transition zone and Zone IV a region of momentum induced air movement. The processes involved in these zones are not really important to the actual analysis, however.

The straight line approximations allow post process analysis in a simple manner (Figure 2). It also is possible to utilize a polynomial analysis of the curve, but the polynomial approach does not lend itself as easily to data reduction for multiple curves. For many air devices, data from different volumes as well as different sizes will fall on a single curve, within the limits of measurement accuracy. For some devices, a series of curves will result, as a function of size, or adjustable blade angle (see Figure 2).

Once plotted and intersected (or regressed), the throw graph is used to determine catalog performance. The throw performance data is calculated by solving the plotted coordinates for the value of T, or throw distance, for a known volume, discharge velocity (Volume/A_k), and area factor, at a desired terminal velocity.

If the straight line approach is used, it is possible to utilize the coordinates of the intersection of the Zone II and III, and the III and IV

IP Data

	Neck Velocity fpm	Air Flow cfm	Pt in. wg	Ps in. wg	Throw ft	NC
6-in. Dia.	200	39	0.004	0.002	1-1-4	-
	300	59	0.009	0.004	1-3-6	-
	500	98	0.026	0.010	3-5-8	-
	600	118	0.037	0.015	4-6-9	-
	700	137	0.050	0.020	4-6-10	-
	800	157	0.066	0.026	5-7-10	12
	900	177	0.083	0.033	6-8-11	16
	1000	196	0.103	0.041	6-8-11	19
	1100	216	0.124	0.049	7-8-12	22

Terminal Velocity (fpm)		
150	100	50
↓	↓	↓
7	8	12

Figure 3 - Throw data at three terminal air speeds

lines in an equation to calculate throw for any size and volume within the range of the input data. If a regressed approach is utilized, the regression is used to solve for the plot coordinates, and the equation solved for throw.

In most cases, throw data is presented for catalog purposes at three terminal air speeds: 150, 100 and 50 fpm terminal velocities. (see Figure 3).

Several manufacturers now have programs to calculate throw for many types of diffusers at almost any terminal velocity. Some even include throw data at different room-to-discharge temperature differentials. Most throw data is shown for an entrained jet (air moving along a surface), unless stated otherwise. An entrained jet typically is reduced in throw by about 30% when it is freed from a surface.

Delta-T effects

The effect of Delta-T on the performance of a diffuser are estimated, or are determined directly. If it is to be measured, it is essential that steady-state conditions be established for the test. Typically, tests will be conducted with a -20°F Delta-T condition, with a 55°F supply and 75°F average space temperature. In order to maintain stable conditions, this requires that these temperatures be maintained throughout the test, and some form of heat load will have to be applied. The heat load should be uniformly distributed, and not create any measurable air currents, which is difficult to achieve. ANSI/ASHRAE 70-2006 sets requirements for tests of this type. Most tests are conducted, and data reported, with isothermal conditions (see Figure 4).

To estimate the effect of Delta-T on throw, one can use a simple rule of thumb. Throw is affected by buoyancy by 1% per degree delta-T at 75 fpm. (The 75 fpm throw lies about halfway between the reported 50 and 100 fpm throw distances. Some computer programs can calculate the 75 fpm throw directly). Remembering that hot air rises and cold air falls, it can be seen that horizontally supplied hot air will travel further, cold air shorter. Here are three examples:

1. 20° Delta-T Cooling, Vertical Down = +20% projection
2. 20° Delta-T Heating, Vertical Down = -20% projection
3. 20° Delta-T Heating, Along Ceiling = +20% projection

When observing throw data curves, it becomes immediately apparent that diffusers of the same type exhibit similar throw plots. Air outlets of a given type will exhibit similar throw plots due to the physics of jet dynamics. The now-obsolete ADC 1062 test standard published a "Simplified Grill Throw Curve," implying that all grills will follow this simple plot. This data assumed that a grill's jet was entrained along a surface. Linear diffusers provide a thin and flat jet along the ceiling. Tests have shown that most linear diffusers exhibit a very similar plot on the throw graph. Free jets from drum louver type devices exhibit a common, but different characteristic curve. The effect of adjustable pattern controls will be different from device to device, and may result in a family of curves on the throw graph. Additionally, some devices, such as common face ceiling diffusers with differing inlet sizes, will exhibit a family of curves, one of each inlet size (see Figure 5).

The throw of a diffuser should be a primary determinant in the selection process of an HVAC system layout, especially with VAV systems. The *ASHRAE Handbook - Fundamentals,* gives a guide to the selection of diffusers as a function of throw, room size and ADPI. The ratio of throw (to a 50-fpm terminal air speed) to a characteristic room length (distance to the wall, or half the distance to the next diffuser) is listed as a function of the room load and diffuser type. Linear diffusers use 100 fpm in this analysis. The table gives expected ADPI values for different ratios. If the predicted ADPI is less than 75%, it will result in occupant dissatisfaction, and it is likely room temperature stratification will exceed the allowable 5°F stratification limit of ASHRAE Standard 55.

With the ADPI table, and with manufacturer's isothermal throw data, the range of volumes that can be distributed by a diffuser can be calculated. It is therefore possible to calculate the range of cfm/ft² for a given diffuser and diffuser spacing (see *ASHRAE Journal,* "Best Practices for Selecting Diffusers," by Daniel Int-Hout, June 2004). Given this analysis, an engineer can determine the effective VAV range of a diffuser. At least one manufacturer's catalog data presents ADPI performance data as a function of flow rate per unit area and diffuser spacing over a range of flows for many types of ceiling diffusers. Electronic computer-based catalogs are now available that will calculate both throw and ADPI data at almost any flow rate, spacing or diffuser size.

Terminal Device	Room Load, BTU/h-ft²	T_{50}/L for Max ADPI	Max ADPI	For ADPI > than	Range of T_{50}/L
High sidewall grilles	80	1.8	68	-	-
	60	1.8	72	70	1.5-2.2
	40	1.6	78	70	1.2-2.3
	20	1.5	85	80	1.0-1.9
Circular ceiling diffusers	80	0.8	76	70	0.7-1.3
	60	0.8	83	80	0.7-1.2
	40	0.8	88	80	0.5-1.5
	20	0.8	93	90	0.7-1.3
Sill grille, straight vanes	80	1.7	61	60	1.5-1.7
	60	1.7	72	70	1.4-1.7
	40	1.3	86	80	1.2-1.8
	20	0.9	95	90	0.8-1.3
Sill grille, spread vanes	80	0.7	94	90	0.6-1.5
	60	0.7	94	80	0.6-1.7
	40	0.7	94	-	-
	20	0.7	94	-	-
Ceiling slot diffusers (for $T_{100}L$)	80	0.3	85	80	0.3-0.7
	60	0.3	88	80	0.3-0.8
	40	0.3	91	80	0.3-1.1
	20	0.3	92	80	0.3-1.5
Light troffe diffusers	60	2.5	86	80	<3.8
	40	1.0	92	90	<3.0
	20	1.0	95	90	<4.5
Perforated, louvered ceiling diffusers	11-50	2.0	96	90	1.4-2.7
				80	1.0-3.4

Figure 5 – Air diffusion performance index (ADPI) selection guide

Low flows and high ceilings

At low flows, the cold air coming out a diffuser can fall off of the ceiling and drop into an occupied space without mixing with room air. Sometimes referred to as dumping (but more properly known as excessive drop), the result is a room that is cool under the diffuser, warm everywhere else, and often cold at the floor. The room thermostat is often very slow to respond to changes of load when this is happening, as the heat rising from local loads has to essentially fill the room before coming in contact with the thermostat sensing element. The resulting stratification, while possibly not unacceptable to occupant's whose exposed skin is the same distance above the floor as the thermostat, can be quite uncomfortable to occupants not wearing socks, due to cold air stratifying at the floor level.

Heating performance cannot be predicted using ADPI techniques. Most space interior zones are in cooling mode year-round, so heating analysis is typically needed only in the outer 10 ft. to 15 ft. of a building. With typical office ceiling heights, tests have shown that a maximum diffuser-to-room differential of 15°F should be maintained to avoid significant stratification in the occupied zone, and to assure ventilation air reaches the building occupants and doesn't short-circuit out of the ceiling returns (see *ASHRAE Journal*, "Overhead Heating: A Lost Art," by Daniel Int-Hout, March 2007). ASHRAE Standard 62.1 2007, Table 6.2, indicates that if the discharge temperature exceeds 15°F and if the 150 fpm throw doesn't come to within 4.5 ft. of the floor, one must increase the quantity of ventilation air by 25%. Standard 55's vertical temperature stratification limit will also likely be exceeded.

When analyzing high-bay heating and cooling applications, it becomes apparent that getting heating air to reach the floor is a challenge. ADPI analysis/prediction is only valid for ceiling heights of 8 ft. to 11 ft. For large spaces, cold air will always make it to the floor.

One sees that in many cases, heating airflow should exceed, often by twice, the cooling airflows, if occupant comfort is to be maintained.

Summary

Throw data is the primary discriminator in selecting diffusers for the best air diffusion in a space. With VAV systems, throw data is the only means of determining the acceptability of a diffuser at reduced flows. Using the *ASHRAE Handbook,* the room ADPI can be predicted using throw data, and data can be prepared to show the range of acceptable flows for a given diffuser spacing. Computer-based systems allow rapid determination of diffuser performance over a range of conditions.

Poor diffuser selection, however, can result in occupant discomfort, very slow thermostat response, and ventilation short circuiting. Understanding how air jets work is essential. |cse|