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June 2004

ASHRAE® JOURNAL

The magazine of the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

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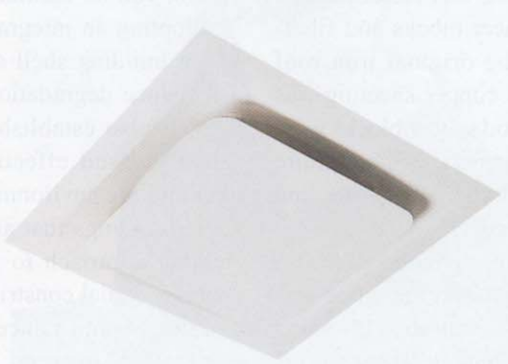


Best Practices for Selecting Diffusers

By Daniel Int-Hout III, P.E., Member ASHRAE

To keep pace with ever-changing workplace environments, new air distribution strategies are needed. The two primary drivers for making changes in the office air delivery system are new building codes/regulations and changes in interior loads.

The first driver for change concerns new building codes and regulations. ASHRAE approved Addendum *n* to ANSI/ASHRAE Standard 62, *Ventilation for Acceptable Indoor Air Quality.* This new ventilation rate procedure contains two tables. Table 6.1 redefines the minimum ventilation rates in terms of cfm per occupant *and* cfm per unit area. Table 6.2 defines values for air distribution effectiveness, which are to be applied to the values in Table 6.1. A portion of Table 6.1 is shown in *Table 1.*

The values in Table 6.1 probably do not have a direct impact on air distribution component selection, but the "rules" in Table 6.2 (shown in *Table 2*) have a great effect on all spaces with overhead heating. Table 6.2 requires that when heating from overhead (common in commercial office spaces), discharge temperatures cannot be more than 15°F (8°C) above room temperature (as has been recommended in the *ASHRAE Handbook—Fundamentals* since 1980), *and* the 150 fpm (7.6 m/s) terminal velocity from the diffuser must extend to within 4.5 ft (1.4 m) of the floor. This is to avoid ventilation short-circuiting to the ceiling return. If not followed, ventilation rates must be increased by 25%.

The second driver is changing interior loads. Many studies report, and observation confirms, that interior loads are not what they once were or are not near design. Energy concerns, productivity changes and technological advances have impacted the loads. Laptops and flat panel monitors have replaced desktop computers and CRT screens, network printers have replaced personal printers, and fewer cubicles are occupied. With 55°F (13°C) supply air, it only takes about 0.6 cfm/ft^2 (3 L/s per m²), at maximum, to maintain acceptable temperatures in many interior spaces.

Most interior systems, however, were designed for 1 cfm/ft² (5 L/s per m²). At reduced flows, many diffuser designs have insufficient projection (throw) to provide uniform temperatures and ventilation mixing. In a recent problem job involving a courthouse with severe indoor air quality problems, several hundred short-throw diffusers were replaced with diffusers with longer throw at the engineer's expense.

Good Practice

Methods for achieving an acceptable Air Diffusion Performance Index (ADPI), and for selecting diffusers are well understood. However, discomfort complaints are still common. Some building occupants can complain of being too hot, while others nearby at the same time complain of being too cold.

With a properly selected set of overhead ceiling diffusers and an HVAC supply system capable of meeting the loads in the space, it is possible to respond to variations in localized loads from 20% to 100% of designed maximum loads (at a variation in space temperature that will go unnoticed by most occupants). This system also provides a ventilation effectiveness of 100% (all the ventilation air supplied at the ceiling will be delivered to the occupants).

Heating performance with ceiling diffusers is well-documented, with discharge and room temperature difference limitations clearly spelled out in both Addendum *n* and the *ASHRAE Handbook—Fundamentals*.

So, why are building occupants complaining of discomfort? The likely culprit is improper diffuser selection, which can lead to many problems.

The first is "dumping." At low airflows, diffuser velocities may not be high enough to create the Coanda effect necessary to overcome negative buoyancy of the cold air being discharged. This causes cold air to drop into the space. As a result, it is cold under the diffuser, warm at the midpoint between diffusers, and the cold air puddles at the floor, creating a vertical stratification in the space (and cold feet).

Another problem occurs at very high airflows. Jets collide at the midpoint between diffusers, causing cold airstreams to drop into the space (where it was hot earlier). The increased induction at the recent cold spot under the diffuser now creates an upwards flow, warming that location.

TABLE 6.1
Minimum Ventilation Rates In Breathing Zone
(This table is not valid in isolation; it must be used in conjunction with the accompanying notes.)

Occupancy Category	People Outdoor Air Rate R_P		Area Outdoor Air Rate R_A		Notes	Default Values		
						Occupant Density (see Note 4)	Combined Outdoor Air Rate (see Note 5)	
	cfm/person	L/s•person	cfm/ft²	L/s•m²		#/1000 ft² (#/100 m²)	cfm/person	L/s•person
Correctional Facilities								
Cell	5	2.5	0.12	0.6		25	10	4.9
Day room	5	2.5	0.06	0.3		30	7	3.5
Guard stations	5	2.5	0.06	0.3		15	9	4.5
Booking/waiting	7.5	3.8	0.06	0.3		50	9	4.4
Educational Facilities								
Daycare (through age 4)	10	5	0.18	0.9		25	17	8.6
Classrooms (ages 5-8)	10	5	0.12	0.6		25	15	7.4
Classrooms (age 9 plus)	10	5	0.12	0.6		35	13	6.7
Lecture classroom	7.5	3.8	0.06	0.3		65	8	4.3
Lecture hall (fixed seats)	7.5	3.8	0.06	0.3		150	8	4.0
Art classroom	10	5.0	0.18	0.9		20	19	9.5
Science laboratories	10	5.0	0.18	0.9		25	17	8.6
Wood/metal shop	10	5	0.18	0.9		20	19	9.5
Computer lab	10	5	0.12	0.6		25	15	7.4
Media center	10	5	0.12	0.6	A	25	15	7.4
Music/theater/dance	10	5.0	0.06	0.3		35	12	5.9
Multi-use assembly	7.5	3.8	0.06	0.3		100	8	4.1
Office Buildings								
Office space	5	2.5	0.06	0.3		5	17	8.5
Reception areas	5	2.5	0.06	0.3		30	7	3.5
Telephone/data entry	5	2.5	0.06	0.3		60	6	3.0

Table 1: Addendum n redefines the minimum ventilation rates in terms of cfm per occupant and cfm per unit area.

Often room partitions, essential for speech privacy in an open plan office, are blamed for poor room air distribution. Partitions, as high as 7 ft (2.1 m) in an open plan office, however, have been shown to actually assist air-distribution patterns, independent of diffuser/partition location, when diffusers are properly selected.

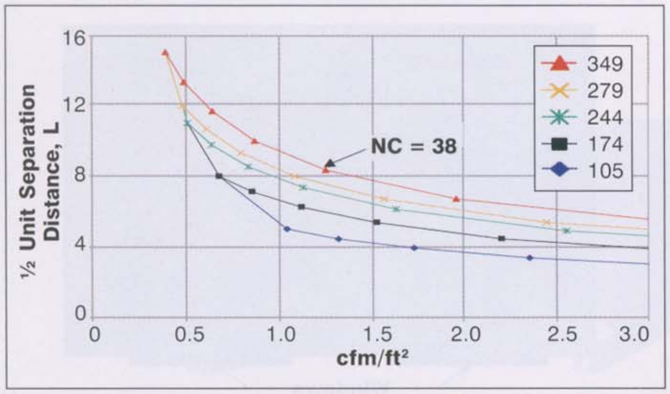
At the perimeter (where closed executive offices often are located), even worse things can happen. In the winter, air is often discharged at 15% of cooling velocities at discharge temperatures of 105°F (41°C). This warm jet has too much buoyancy and too little projection to mix with cold air, which will spill down the window and result in an 8°F to 10°F (4°C to 6°C) difference/stratification between 6 in. and 6 ft (0.1 m and 1.8 m) from the floor in the middle of the room (contrary to the minimum requirements of ANSI/ASHRAE Standard 55, *Thermal Environmental Conditions for Human Occupancy*).

TABLE 6.2
Zone Air Distribution Effectiveness

Air Distribution Configuration	E_z
Ceiling supply of warm air less than 8°C (15°F) above space temperature and ceiling return provided that the 0.8 m/s (150 fpm) supply air jet reaches to within 1.4 m (4.5 ft) of floor level. **Note**: For lower velocity supply air, $E_z = 0.8$.	1.0
Floor supply of cool air and ceiling return provided that the 0.8 m/s (150 fpm) supply jet reaches at least 1.4 m (4.5 ft) above the floor. **Note**: Most underfloor air distribution systems comply with this proviso.	1.0

Table 2: Table 6.2 from Addendum n addresses overhead heating.

In summer, heat rising from the window stratifies at the ceiling. Cold air from a perimeter-located diffuser (often still set to the original setting to blow down) stratifies at the floor. In both cases, at 43 in. (1 m) above the floor (where the thermostat is located), it is 75°F (24°C), so the direct digital control (DDC) system assumes all is well. Worse, ventilation air, also supplied at the ceiling, travels to the ceiling return and is exhausted without actually providing any conditioned air to the space.



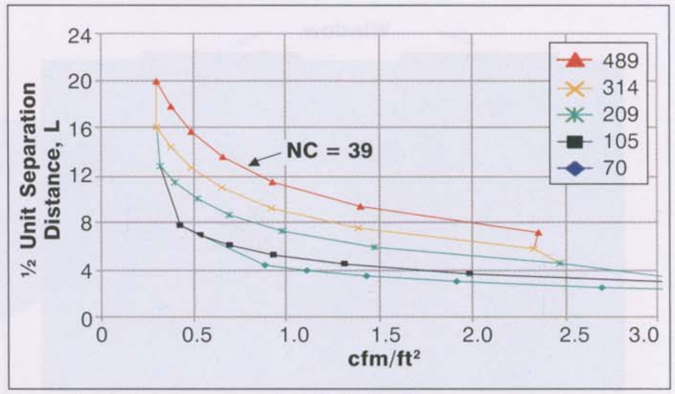


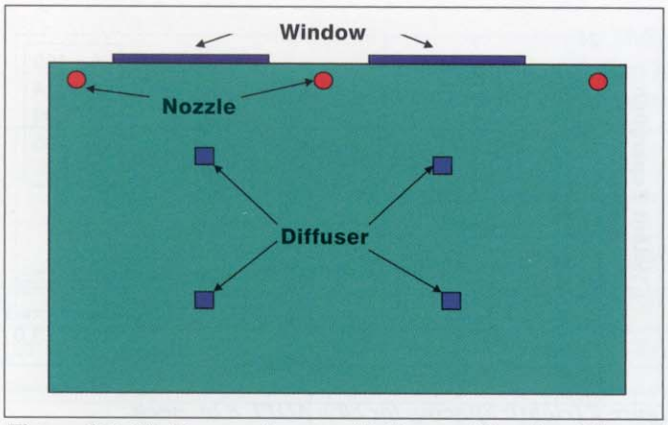
Figure 1 (left): Face-mounted deflector, 8 in. inlet, four way. Figure 2 (right): Spacing for 80% ADPI, 8 in. neck.

Possible Solutions

An uncomfortable environment can be a problem for more than just the occupants. It can be uncomfortable for engineers who have the information to provide acceptable spaces, but fail to use that data in their designs (most of which has been in the ASHRAE Handbook for 20 years).

Here are four retrofit solutions (also suitable for new construction). These solutions will provide increased occupant, and will comply with the Addendum *n* requirement, without increasing quantities of outside air.

1. **Better low velocity diffusers.** Diffusers have specific performance envelopes that can be defined in terms of diffuser spacing and airflow per unit area ("Maximizing Air Change Effectiveness," *ASHRAE IAQ Applications*, Winter 2000). Different diffusers have differing performance envelopes. Some perforated face diffusers, especially those with face-mounted



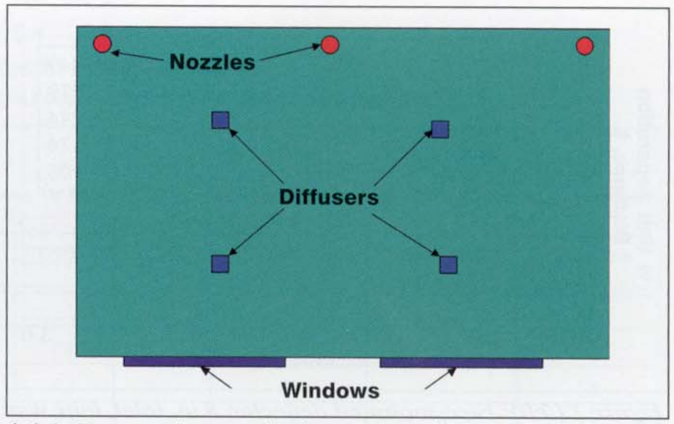


Figure 3 (left): Decoupling ventilation and heating. Figure 4 (right): Decoupling ventilation and cooling.

deflectors, have short throws. They are suitable for high airflow rates, but have lower airflow limitations that may make then unsuited for VAV applications in the interior of modern offices. *Figure 1* shows the performance of such a diffuser.

If one enters the chart at 1 cfm/ft² (5 L/s per m²) (as is a typical interior air design airflow rate), one can see that the optimum half-diffuser spacing would be about 10 ft at 350 cfm (3 m at 165 L/s). Using this as a given, the lowest airflow rate meeting the ADPI requirements would be just over 0.5 cfm/ft² (2.5 L/s per m²). If 280 cfm (132 L/s) is the maximum flow (for diffuser noise reasons), the half-spacing is 8 ft (2.4 m), and the minimum flow rate is 0.65 cfm per ft² (3.3 L/s per m²).

If a diffuser with a longer throw is selected, such as the "architectural" diffuser (a square plaque in a deep back pan) data shown in *Figure 2*, the range of airflows in much lower, and a diffuser location that will result in acceptable performance between 0.3 and 1.0 cfm/ft² (1.5 and 5 L/s per m²) can be found. One recently introduced architectural diffuser can be installed in many perforated face diffuser back pans from below the ceiling, reducing retrofit costs.

2. **Smarter reheat.** Reheat options for VAV terminals have traditionally been either hot water coils, or multistep electric heat. Often, the heating flow is constant volume, while the cooling flow is VAV. With modern DDC, it is possible to provide "dual minimum" control at no additional cost with a higher heating airflow than the cooling minimum. When reheating cooled primary air, an energy penalty exists, so the most economical solution is to have the heating airflow set as low as possible. This often results in high discharge temperatures, very poor air distribution in the heating mode, ventilation short-circuiting and temperature stratification as a result.

Modern electronics, however, allow for temperature-limited proportional electric heat at a low premium cost over staged heat. When used properly, a proportional heat controller can limit discharge temperatures to within the Addendum *n* limits (<15°F [<8°C] above room temperature). If further developed, a VAV controller can increase the airflow on further demand for heat, minimizing the amount of reheat, and still providing acceptable ventilation in the space. The changed airflow rates affect the diffuser selection to ensure proper perimeter zone air distribution.

3. **Decouple ventilation and heating.** It may be desirable to decouple the heating and cooling air supply to the room (*Figure 3*). One option is to use down-blow nozzles at the window, with heated plenum air supplied from a heating only fan coil. A parallel fan box controller on the single duct VAV terminal used for cooling can control this fan coil. This strategy has been used for more than 30 years, and is a proven and effective method of heating a perimeter. Since the nozzles are not providing ventilation air, Standard 62's heating limitations does not cover them.

4. **Decouple ventilation and cooling.** In a similar manner, it may be desirable to provide a separate ventilation system, with conditioned room temperature 100% outside air (*Figure 4*). To avoid unwanted jet collisions, the same jets described earlier may be used against an interior corridor wall to supply ventilation air. This approach has been used as a retrofit in schools, where ventilation rates can be fixed. The small (4 in. [0.1 m]) ducts can be easily threaded through building structures from a corridor installed supply duct.

Summary

The office environment is always changing. New standards, which will no doubt become codes, coupled with reduced loads have changed the requirements for air distribution in the workplace. To keep pace with these changes, new ways of looking at diffuser selection is mandatory. At the same time, reduced costs for digitally controlled circuits are allowing innovative ways of managing the zone response to heating and cooling needs. Many of these strategies can be as easily incorporated into retrofit designs as in new construction. The opportunities are there, if only we decide to act.

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