A two-way, high-induction diffuser.

Dedicated Outdoor Air Systems

Air Diffusion Performance

By S.A. Mumma, Ph.D., P.E., Fellow ASHRAE

Dedicated outdoor air systems (DOAS), when used with a hydronic parallel sensible cooling system, generally deliver far less supply air to the spaces than a conventional all-air system supplying a large amount of recirculated air. This has led some to question the DOAS's ability to provide sufficient air movement to meet comfort requirements. This column presents test results from a DOAS delivering the minimum ventilation air via overhead high-induction diffusers.

Mumma

The Test Facility

The portion of the test facility used in this experiment consists of a 40 ft × 50 ft (12 m × 15 m) space housing 35 students and their drafting tables arranged as shown in *Figure 1*. The facility has one exterior exposure, 50 ft long by 14 ft high (15 m by 4 m) with nine 3 ft × 8 ft (9 m × 2 m) single-glazed windows. All of the other room enclosure surfaces are interior.

Six overhead Thermal-Core 24 in. (0.6 m) two-way blow high-induction diffusers, spaced uniformly at the 9 ft (2.8 m) elevation and along the 50 ft (15 m) centerline of the room, provide 108 cfm (51 L/s) each, or a total of 650 cfm (304 L/s) of 100% OA to the space. At the ventilation air delivery rate, 0.325 cfm/ft^2 (1.7 L/s per m^2) of air is delivered to the space.

This delivery rate is only about 30% as much air as typically delivered by a conventional all-air system. The test facility is equipped with ceiling radiant cooling panels for sensible cooling in the summer. To minimize the impact of the radiant cooling on natural convection air movement, the tests were conducted in winter when all of the space cooling (less than 2 tons [7 kW] sensible plus latent during the test) could be accomplished with the DOAS. During the testing, the OA temperature was 37°F–39°F (3°C–4°C), and the wind velocity around 5 mph (8 km/h).

The test space has no active means of heating, other than the 24–7 internal generation of lights, equipment, appliances, and occupants. Cooling is required in the space year-round. The constant volume flow of OA is tempered as necessary by the DOAS using recovered heat at the enthalpy wheel. During the testing, the supply air temperature (SAT) was between 55°F and 57°F (13°C and 14°C). A schematic of the system is illustrated in *Figure 2*.

Effective Draft Temperature

Thermal comfort is a function of the following variables that influence metabolic heat transfer:

1. Dry-bulb temperature (DBT),
2. Relative humidity,
3. Mean radiant temperature,
4. Air movement,
5. Metabolism, and
6. Clothing worn by the occupants.

Provided there is sufficient heating or cooling to meet the thermal and humidity control requirements, comfort is almost completely a function of the space air distribution. Proper air distribution prevents thermal stratification and stagnation in order to approach a homogenous mixture of room air.

Based upon the local and room air temperatures and velocities, an effective draft temperature (EDT) may be calculated using the following relationship:[1,2]

$$\theta = (T_L - T_R) - 0.07(V_L - 30)$$

where

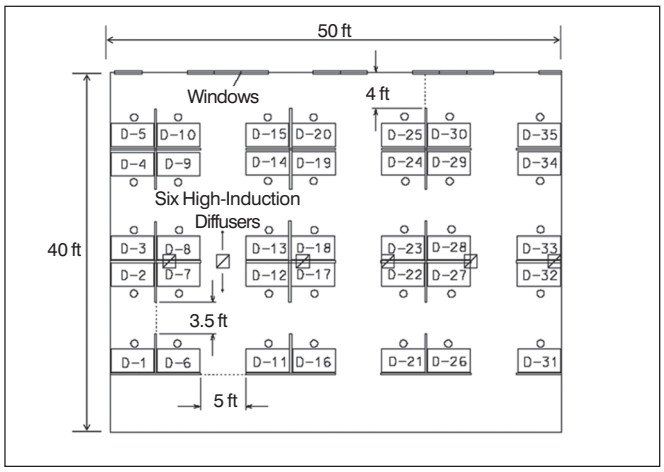

Figure 1: Test facility floor plan with 35 drafting tables.

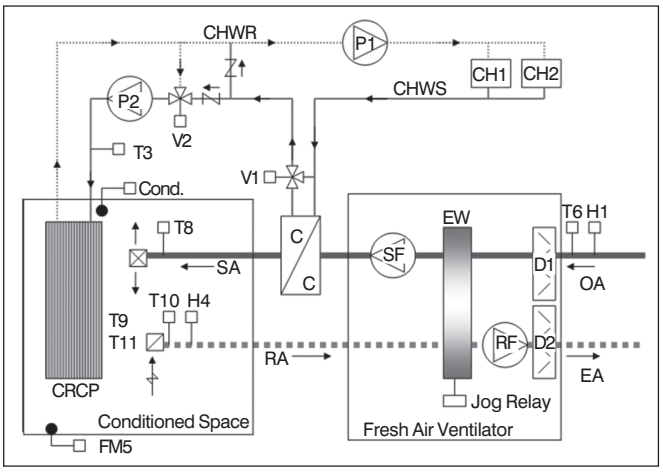

Figure 2: Schematic of the test facility.

- θ, EDT, °F
- T_L, local mean airstream DBT, °F
- T_R, average room DBT, °F
- V_L, local mean airstream velocity, fpm

Research has shown[3] that a high percentage of people in sedentary occupations are comfortable when the EDT is between –3°F and 2°F (–1.7°C and 1°C).

Using the EDT and a local velocity upper limit of 70 fpm (0.36 m/s) as the criteria, the comfort level of a space can be determined based upon the air diffusion performance index (ADPI). The ADPI is an indication of the percent of locations in a space with a local velocity of 70 fpm (0.36 m/s) or less and an EDT between –3°F and 2°F (–1.7°C and 1°C). When the ADPI approaches 100%, the most desirable comfort conditions are achieved.

Data Gathering

Local mean velocity and temperature measurements were taken at each of the 35 drafting tables. The raw data is presented in *Table 1*, along with the computed EDT based upon an average room air temperature of 76.1°F (24.5°C). While this winter average room temperature may seem hot and energy wasteful, this is the condition that the occupants requested for comfort, and the OA was completely tempered with recovered heat at the enthalpy wheel. The data in *Table 1* was collected with a low velocity analyzer capable of providing mean values for both the local velocity and temperature. The average room temperature was measured at the common return with a calibrated thermistor and taken from trend plots. The outdoor air conditions reported above are from the campus weather station.

station location	1 min. mean vel. ft/min	1 min. mean local temp. °F	θ, eff. draft temp. °F	
1	11.8	76.1	1.3	
2	23.6	74.84	-0.8	
3	13.2	75.38	0.5	
4	17.9	73.94	-1.3	
5	12.0	73	-1.8	
6	19.7	75.56	0.2	
7	14.8	77.18	2.1	max.
8	27.8	75.2	–0.7	
9	24.8	74.12	–1.6	
10	20.9	73.1	–2.4	min.
11	13.8	76.82	1.9	
12	12.8	76.1	1.2	
13	20.5	75.56	0.1	
14	30.1	73.94	–2.2	
15	29.1	74	–2.0	
16	23.4	75.02	–0.6	
17	12.6	75.92	1.0	
18	12.4	75.74	0.9	
19	21.7	74.84	–0.7	
20	14.0	74.48	–0.5	
21	18.7	75.74	0.4	
22	14.8	75.38	0.3	
23	13.8	75.56	0.6	
24	13.0	74.3	–0.6	
25	12.0	74.48	–0.4	
26	13.4	76.82	1.9	
27	16.5	75.74	0.6	
28	18.5	74.84	–0.5	
29	14.8	76.1	1.1	
30	14.6	76.28	1.3	
31	22.0	76.46	0.9	
32	25.0	74.66	–1.1	
33	12.4	76.1	1.2	
34	28.5	75.38	–0.6	
35	13.8	75.38	0.4	

Table 1: Raw data and the resulting EDT.

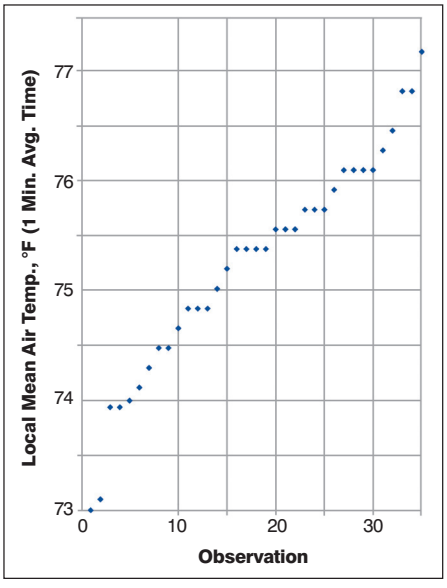

Figure 3: Local mean air temperature.

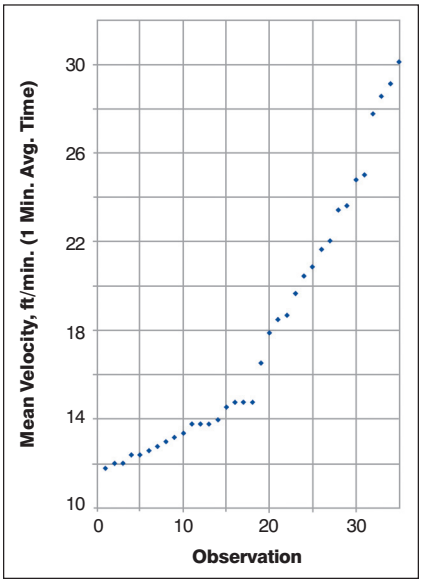

Figure 4: Local mean air velocity.

Data Analysis

The mean local temperature data were sorted in ascending order and presented in *Figure 3*. The lowest temperature, 73°F (23°C), occurred near a window (drafting table 5). In general, the local temperatures were lowest at drafting tables near the exterior wall. The highest temperature, 77.2°F (25°C), which was 1.1°F (0.6°C) hotter than the average room temperature, was recorded under a diffuser near the left center of the room.

Similarly, the mean local velocity data were sorted in ascending order and presented in *Figure 4*. The minimum velocity, 11.8 fpm (0.06 m/s), occurred at drafting table 1. The maximum velocity of 30.1 fpm (0.15 m/s) (well below the 70 fpm [0.36 m/s] upper limit of the ADPI requirements) occurred at drafting table 14.

Both the air motion and temperature were impacted by the 5.5 ft (1.7 m) high partitions around the drafting tables, but no pattern is apparent to this author concerning either velocity or temperature data variations.

The EDT resulting from the above local temperature and velocity data ranged from a low of –2.4°F (–1.3°C) to a high of 2.1°F (1.2°C). The 4.5°F (2.5°C) spread is less than the 5°F (2.7°C) EDT spread used in the ADPI calculations, indicating that a reduction in the SAT means that the range would shift the range toward the minus side, bringing all of the observations within the –3°F to 2°F (–1.7°C and 1°C) EDT range. However, this would have caused overcooling of the space.

The 35 mean local temperature-velocity observation pairs are presented in *Figure 5*. All but one of the data pairs falls between EDT of –3°F and 2°F (–1.7°C and 1°C). And all of the local mean velocities are below 70 fpm (0.36 m/s). Consequently, 34 of the 35 observations fall within the EDT range, leading to an ADPI of 97%. And, the one point that is out of range is barely above 2°F (1.1°C).

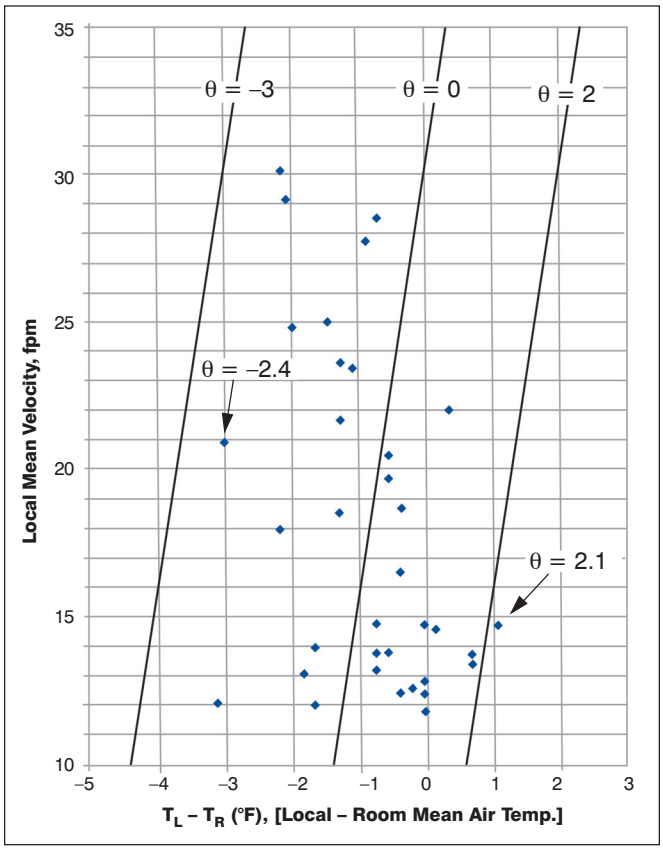

Figure 5: Effective draft temperature,* θ. *ADPI = (34/35)×100 = 97%, or 97% of the observations were between –3°F<θ<2°F.

Conclusions

With an ADPI of at least 97%, the DOAS project demonstrates that spaces served with relatively low air quantities can have very satisfactory thermal comfort based upon air motion and temperature criteria.

The expected maximum ADPI range[4] for ordinary diffusers of various designs and locations is 61 to 94. The superior ADPI results obtained in the experiment reported here are thought to be the result of using high induction diffusers with the low supply air quantities at constant volume. Clearly it is not necessary to have supply air quantities in the order of 1 cfm/ft^2 (5 L/s per m^2) to achieve good air distribution.

Finally, by ensuring good air motion with the ventilation air flow alone, it is not necessary to move large quantities of air.[5] As a result, there can be significant air movement energy savings when a hydronic parallel system, such as ceiling radiant cooling panels, is used to meet the balance of the space sensible load not met with the ventilation air.

References

1. Ryberg, J. and P. Norback. 1949. "Air distribution and draft." *ASHVE Transactions* 55:225.

2. Kosetel, A. and G.L. Tuve. 1955. "Performance and evaluation of room air distribution systems." *ASHRAE Transactions* 78(2):235.

3. Nevins, R.G. and E.D. Ward. 1968. "Analysis, evaluation and comparison of room air distribution performance." *ASHRAE Transactions* 61:533.

4. *ASHRAE Handbook—Fundamentals* pg. 32.14, Table 4.

5. Mumma, S.A. 2003. "Meeting air change criteria." *IAQ Applications* 4(3):18–19.

S.A. Mumma, Ph.D., P.E., is a professor of architectural engineering at Penn State University, University Park, Pa.