designing ventilation air delivery

The author maintains that the testing methodology of ASHRAE 62-1989R, mired in a swamp of controversy and "continuous maintenance," may need some fresh air.

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There has always been a question as to how to apportion total air supply to a room in order to ensure the delivery of the specified minimum outdoor air for ventilation. ASHRAE Standard 62-1989R[1] attempted to do that, but problems still exist.

What is at issue? The committee, SSPC 62, had what many consider to be an impossible task in specifying how to design a system, which essentially requires building and testing it. It is no wonder that this and other unknown, controversial issues forced ASHRAE to withdraw 62-89R and to place the current Standard, 62-89, on "continuous maintenance," an opportunity to put 62-89R to a "line item veto" (more accurately, to subject each of the controversial parts to a "peer review").

With that much in mind, Table A.3.1.b. of 62-89R, which exhaustively covers minimum total air supply, should be reviewed as an indispensable requirement or specification. According to many, Standard, 62-89R gives no clear bases for it. Its Table A.3.1.b. lists both "Prescriptive Rates" for ventilation (R_{SS}), and "Simple System Rates" of total air supply or ventilation (R_{SR}). The problem may be with the total air supply and the assumptions leading to it.

For example, an office space is listed as follows (Table A.3.1.b.):

Prescriptive Rates:
R_p = 6.0 cfm/person (people component)
R_b = 0.07 cfm/sq ft (building component)

Air requirements were calculated using Simple System assumptions of six people/1,000 sq ft (density), an occupancy factor of one, and an air change effectiveness of 0.8 for overhead air delivery of cold and warm air (all in the Standard). The tabulated Simple System Rates are 0.13 cfm/sq ft of outside air and the same amount of supply air. Standard 62-89R gives the maximum supply rate (MSR) as a function of the design ventilation rate (DVR), using the following formula (6-4a):

$$\text{MSR} \geq \text{DVR Ea}$$

Substituting the above values, we would get the following:

$$\text{DVR} > 6 \times 1 \times 1 + 0.07 \times \frac{1{,}000}{6} = 17.67 \text{ cfm}$$

Finally,

$$\text{MSR} \geq \frac{17.67}{0.8} = 22.09 \text{ cfm, or,}$$

$$R_{SR} = R_{SS} \; \frac{22.09 \text{ cfm}}{167 \text{ sq ft/per}} = 0.13 \text{ cfm/sq ft}$$

The origin of the 0.13-cfm/sq-ft value was disclosed, inadvertently, in an article[2] connected to SSPC 62, and it will be shown to be questionable.

THE WRONG TOOLS

All of this has to do with room air distribution. Standard 62-89R refers to Chapter 31, Space Air Distribution[3]. Formulas in the chapter for "throw" and "terminal velocity" are strictly valid for space air diffusion, with no walls to intercept the airstream. It also provides no basis for selecting the terminal velocity to define throw, except for one means, the air diffusion performance index (ADPI).

In this regard it is stated that [the objective is to] "provide a means for maximizing the air diffusion performance index (ADPI) for mixing air diffusers…." The ADPI is a "test" protocol intended to yield a single-number rating of a room air distribution system unique for any application to be tested.

ADPI has virtually no utility as a design tool, as clearly inferred from a quotation by one of the principal researchers who developed the ADPI protocol[4]: "It must be emphasized that, although a part of the design method [ADPI] involves equations, the method must still be considered "descriptive. ... No predictions are possible of local values of variations in temperature difference or velocity. Most significantly, the method... has no known relationship to contaminant distribution."

These statements are mutually supportive. Moreover, being a test protocol, even the inclusion of the phrase "design method" would be misleading in view of the characterizations quoted above.

The calculation of air change effectiveness (Eac) is another process worth considering. Eac is determined from a tracer gas test protocol, such as ASHRAE 129, to empirically establish the value appropriate for an intended design. For the supply of cold air from the ceiling (not defined, but presumed to be horizontal air distribution), a default value of 1.0 is assigned in Table 6.2 of 62-89R. For heating, a default Eac value of 0.08 is assigned. Using the latter value and employing the equations of the "Prescriptive" procedure, one arrives at the previously tabulated value of 0.13 cfm/sq ft.

The head of the Air Infiltration and Ventilation Centre (AIVC) states in respect to the broad subject of ventilation effectiveness, that "... since [it] is often unique to each enclosure, measurement analysis [tracer gas testing] generally has a limited design role[5]."

Standard 62-89R does, indeed, reference ASHRAE Standard 129, a work in progress, as the proposed basis to determine Eac by use of tracer gas. A test protocol conducted on a mock-up would then yield values for Eac for design of the room air distribution.

Is this a realistic approach as a code design requirement?

NO DUMPING ALLOWED

The committee has taken great care in providing explanatory language and information to clarify the bases of applying the "normative" requirements that would constitute the essence of the [eventual] ASHRAE 62 updated standard. The explanations and supplementary information are virtually devoid of any bases to support the proposed [very] low values of MSR, obtained in the absence of test values of Eac to support a specific design effort.

As stated earlier, the article[2] relied heavily on interpretations of a research paper[6]. Given the available data, the article cannot support the conclusions subsequently drawn. The bases of concerns about the interpretations of the research paper (and, therefore, the questionable decision to employ a very low value of the MSR) follow.

In the referenced research paper[6], two sets of tests were conducted:

- The first set was "Thermal Measurement Test Conditions," for which very extensive spatial air speeds (not velocity) are reported, including ADPI [test] evaluations in two ranges of experiments — 0.46-0.56 cfm/sq ft (5-6 cfm/m²), and 0.93-1.02 cfm/sq ft (10-11 cfm/m²).
- The other tests cited were "Tracer Gas Test Conditions." Here, the authors report the averages of the ages of air at knee, breathing, and ceiling levels in and around the exhaust air duct and partitioned workstations. Technical limitations obliged the researchers to confine total air supply to a maximum of 200 cfm (94.4L/s); in fact, seven of the nine cooling [only] tests were at or below 110 cfm (see Table A). No spatial air speeds or air velocities are reported in conjunction with these tests.

This is very significant, as there are no alternative test data or observations reported about whether any dumping at these low rates of airflow took place.

Test No.	cfm	cfm/sq ft	$T_b(h)$	$T_e(h)$	T_e/T_b
21	100	0.31	0.40	0.43	1.08
22W	100	0.31	0.44	0.45	1.02
22	100	0.31	0.39	0.44	1.13
24	110	0.34	0.41	0.43	1.05
25	110	0.34	0.39	0.44	1.13
39	210	0.65	0.27	0.30	1.11
40	200	0.62	0.26	0.29	1.12
41	55	0.17	0.78	0.88	1.13
42w	100	0.31	0.50	0.51	1.02

Note: cfm x 0.472 = L/s; cfm/ft² x 5.1 = L/m²

TABLE A: Tracer gas tests for cooling-only air supply. Here, T_b represents the age of air at breathing level (an average value of several measurements with tracer gas,) *h*. T_e represents the age of air in the exhaust duct (derived from tests with tracer gas, *h*). Also, T_e/T_b is an index reflecting a demarcation between complete mixing of air supply with room air and pure piston flow. Depending on the degree of deviation from these theoretical flow patterns, the ratio would be ≤1 and ≥1 (but ≤2), respectively[9]. (Note: cfm x 0.472 = L/s; cfm/sq ft x 5.1 = L/m²)

Table A of this article extracts age of air test data from Table 5 of the research paper[6]. Understanding the meaning of the data and subsequent reduction of it is uncomplicated. The reader would then be rewarded by an appreciation of why using low values of rates of air delivery per unit of floor area, as earlier stated, may be unwarranted based on the published data.

The symbols and ratios given in Table A only need to be defined to comprehend the comments and analyses offered here.

The authors have estimated an accuracy of 0.12 for the ratio T_e/T_b. Not withstanding their caution, all of the ratios are greater than 1.0, which would support the probability that the air at breathing level is "fresher" (i.e., lower "age") than the age of air in the exhaust. One such scenario would involve the dumping of cold air issuing from the diffuser so that the freshest (lowest age) air would be directed, initially, to breathing level at the cubicles. There appears to be no other data reported to contradict this hypothesis.

Independent of these tests, there is reported testing[7] with cold air in an environmental chamber employing a 24- by 24-in. (0.61- by 0.61-m) four-way-pattern, perforated-face air diffuser. The temperature difference between room air and supply air ranged between 17°F (9.4°C) and 20°F (11.1°C). The report[7] states a loss of Coanda effect (i.e., dumping at 320 cfm [151 L/s], which is 80 cfm [37.8 L/s] per side).

In Table A, the highest flow rate is 210 cfm (99.1 L/s). This, given the three-way airflow pattern employed in the original report[6], obtains 70 cfm (33 L/s) per side. Thus, the data and analyses appear to indicate that dumping was occurring. At 110 cfm (51.9 L/s) and lower total flow rates, the foregoing data and analyses appear to indicate that dumping was occurring with cold air delivery.

Similar reasoning suggests that many, but not all, of the tests to evaluate ADPI may have also had dumping at air supply rates at or lower than 80 cfm/side (37.8 L/s), the critical limit for dumping to occur.

FINDING A WAY OUT

General office areas partitioned into low-height cubicles may suffer low levels of Eac with overhead horizontal air distribution. In such cubicles, multiple sources of heat and emissions simply magnify the adverse consequences of low Eac, thereby inducing poor IAQ.

A designer may wish to use a system for which existing formulae or available air velocity estimates are available. Desktop task-oriented air diffusers are one option about which, coincidentally, another paper[8] has been published, among a growing list. The desktop air diffuser could be part of a system that includes supplementary air supply from floor jets or displacement ventilation, now extant in many European installations and found increasingly in the Western Hemisphere.

Ceiling-mounted non-induction air diffusers that obtain a semi-cylindrical airflow pattern (usually downward into cubicles or a zone of occupancy) are commercially available with stated air velocity patterns. Multi-directional and linear ceiling-mounted air diffusers are available with adjustment means independent from any horizontal flow direction, to adjust airflow into the occupied zone.

In these ways, the air velocities into the zone of occupancy may either be predicted or controlled to maximize the value of Eac and, thereby, the IAQ.

CONCLUSION

The preceding proposals are recommended as the most direct means of satisfying the objective of providing ventilation stated at the outset of this article. They avoid the costly determination of Eac by full-scale model and tracer gas testing.

In these proposals, the designer needs to evaluate the likelihood of achieving "reasonable" air change at the microenvironment of the occupant(s), doing so over a range of air supply in a vav system.

In light of these efforts, achieving an "acceptable" ADPI becomes meaningless and possibly counterproductive.

Measurement of air speed and temperature, and adjusting and/or balancing air delivery to the microenvironment, could become the design and operating objectives. IAQ testing for qualifying Eac could then be conducted inexpensively by testing samples taken in the microenvironment and the exhaust air grille. Such tests could use contaminant concentrations (i.e., CO_2 levels) to quantify air change efficiency values.

References

1. BSR/ASHRAE Standard 62-1989R, Ventilation For Acceptable Indoor Air Quality, August 1966. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta.
2. Steven T. Taylor, "Series Fan-Powered Boxes," *ASHRAE Journal*, July 1996. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta.
3. *1997 ASHRAE Handbook*, Fundamentals. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta.
4. P.L. Miller Jr. "Descriptive Methods". Symposium — *Building Systems: Room Air & Contaminant Distribution*, 5-8 December 1988, published by American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, GA.
5. Martin Liddament, "A Review of Ventilation Efficiency." *Technical Note AIVC 39.* Air Infiltration and Ventilation Center, Coventry, Great Britain.
6. Bauman, F.S., D. Faulkner, E.A. Arens, W. J. Fisk, L.P. Johnston, P. J. McNeel, D. Pih, H. Zhang. "Air Movement, Ventilation, and Comfort in a Partitioned Office Space." *ASHRAE Transactions 98*, 1992.
7. Robert S. Hunka, *Acutherm Research Report Number 10-95*, December 19, 1995. Acutherm, Hayward, CA.
8. Bauman, F.S., H. Zhang, A.E. Arens, C.C. Benton. "Localized Comfort Control with a Desktop Task Conditioning System: Laboratory and Field Measurements." *ASHRAE Transactions DE-93-8-1*, 1993.
9. "A Guide To Air Change Efficiency." *Technical Notes AIVC 281990*, Air Infiltration & Ventilation Centre, Coventry, Great Britain.

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