\# 10516

Demand-Controlled Ventilation of an Entertainment Club

Gaylen V. Atkinson
Member ASHRAE

ABSTRACT

Entertainment clubs, nightclubs, theaters, restaurants, and coliseums, with their highly variable occupancy rate, are excellent candidates for demand-controlled ventilation. The dynamic thermal requirements of both heating and cooling, coupled with the need to control indoor air quality because of the large number of patrons who also may be smoking during the highest occupancy, provide an opportunity to integrate the temperature controls with an indoor air quality control system. Significant energy savings may be realized by controlling the ventilation of outdoor air to match the heating, cooling, and humidity requirements as well as maintaining acceptable indoor air quality. This paper describes a demand-controlled ventilation system that was installed in an entertainment club in Boise, Idaho, using a multigas indoor air quality sensor to measure the level of indoor air pollutants, which, when combined with a mixed-air temperature sensor to provide economizer cooling, introduces outdoor air at a rate required to adequately ventilate the space.

BACKGROUND

The entertainment club described in this paper is located in Boise, Idaho. It has a seating capacity of 244 around tables where drinks and food are served while local and celebrity comedians perform their acts during scheduled showtimes. Smoking is permitted during the performances; therefore, local zoning and code requirements call for a high air change rate to meet indoor air quality requirements. The intermittent operation of low occupancy during the day and very high occupancy during evening performances, coupled with the need to maintain the air quality by diluting the secondary tobacco smoke during the performances, generated a concern over energy use. It was determined that a mechanical air-handling system was needed with a maximum outdoor air capacity of 5,800 cfm (2,738 L/s) to handle peak crowds yet also take advantage of mixing return air with the outdoor air for economizer operation, especially during non-showtimes or less-than-capacity performances. This figure came from 290 people total occupancy at 20 cfm (9.4 L/s) per person. A control method was desired that would provide the outdoor air necessary for thermal loads and indoor air quality during the performances but go back to a standard mixed-air cycle during other occupied times.

DESCRIPTION OF MECHANICAL SYSTEM

Several different mechanical design options were considered. Among them, using up to 100% outdoor air was considered desirable for the performances, with a standard mixed-air economizer cycle during other occupied times. Although it was considered, the size of the mechanical system and project budget did not justify heat recovery for a 100% continuous outdoor air operation. Finally, a single constant-volume package unit was selected with a 5,800-cfm (2,738-L/s) capacity; outdoor air, return air, and relief air damper sections; an evaporative cooling section; and two stages of gas heating to handle 100% outdoor air at −10°F (−23°C).

Direct evaporative cooling was selected because it was a good match for the dry Boise climate with very low summer wet-bulb temperatures along with the requirement to cool 100% outdoor air. A direct expansion (D/X) cooling section, sized to handle 100% outdoor air, would have been considerably more expensive in both first cost and cost of energy for operation. In addition, the D/X section either would have had considerable excess capacity for smaller crowds or nonperformance occupied periods or would have needed an expensive control package to meet temperature control requirements at light loads. It would have been difficult to control the space temperature during the cooling mode without wide temperature swings and short-cycling the mechanical equipment. The potential necessity for 100% outdoor air during performances was a good match for the direct evaporative cooling selection because evaporative cooling is usually only operated when the unit is in full outdoor air mode.

Gaylen V. Atkinson is president of Atkinson Electronics, Inc., Salt Lake City, Utah.

#1061

CONTROL CONSIDERATIONS

A standard constant-volume package unit control system was considered but felt to have several drawbacks. A typical programmable commercial-grade multistage heating/cooling thermostat with the mixed air being the first stage of cooling probably would have been adequate for the space temperature control if there was not a requirement for 100% outdoor air during capacity performances. Three different modes of operation were determined.

I. *Unoccupied Mode*: The unit would normally remain off outside of business hours because no one would be present in the building. The unit fan, however, would start along with the heating section if the space temperature dropped below 60°F (15°C) and the outdoor air damper would remain closed. At the other extreme, if the space temperature rose above 85°F (29°C), the unit fan would start along with the evaporative cooling and the outdoor air damper would open.

II. *Occupied Mode*: The unit fan would start, the outdoor air damper would open to a minimum ventilation position, and the space temperature would be controlled by cycling the heating stages, when needed, to maintain a 72°F (22°C) heating setpoint. For cooling, a 74°F (23°C) cooling setpoint would be maintained by enabling the mixed-air controller (set for 60°F [15°C]) for the first stage and then starting the evaporative cooler with opening the outdoor air damper to 100% for the second stage of cooling. With this cycle, the "standard economizer function" of closing the outdoor air damper to a minimum position when starting the cooling section would be disconnected as full outdoor air is needed to properly operate the direct evaporative cooling section.

III. *Performance Mode*: The unit would already be operating in the occupied mode but would then have the capability of opening the outdoor air damper to a 100% position if required to ventilate the space with higher air changes per hour to maintain the desired indoor air quality. Several modes of operation were considered.

(1) A time clock could be installed to open the outdoor air damper to 100% during performances. This would be simple but would have a cost impact of heating unnecessary amounts of outdoor air when there were less-than-capacity crowds.

(2) An "air quality switch" could be installed in the stage manager's office to open the outdoor air damper to 100% if the air quality during performances became objectionable. This option had the liability of allowing the indoor air quality to become poor before someone would have to take manual action to correct the problem. With either option, because of the possibility of very low supply air temperatures causing comfort problems, a discharge air low temperature limit thermostat would be required to cycle the gas burner to maintain tempering of the outdoor air until the space thermostat determined that heating was required.

(3) The third option was installing some kind of sensor to measure the indoor air quality and override the thermostat, modulating the outdoor air damper to bring in the quantity of outdoor air necessary to ventilate the space and maintain an acceptable indoor air quality level. It was decided to pursue this option if an inexpensive yet acceptable air quality sensor could be selected. A carbon dioxide (CO_2) sensor was considered but not used because even though CO_2 would be a good indicator of crowd size, in this case it would not necessarily reflect the total indoor air quality problem because of the potential of considerable amounts of secondary tobacco smoke.

INDOOR AIR QUALITY SENSOR DESCRIPTION

Therefore, the air quality sensor selected was the multigas type that was sensitive to volatile organic compounds (VOCs). It was installed in the return air duct and consisted of a stannic (tin) oxide element heated to a constant temperature. Molecules of VOCs are catalytically reduced or oxidized on the heated surface, thus changing the conductivity of the element. This change in resistance is amplified electronically and converted to a 0 to 10 volt DC linear analog signal, with 0 volts corresponding to "poor air quality" and 10 volts DC corresponding to "good air quality." An increased number of VOC molecules indicates a "poorer air quality." The sensor responds well to butyric acid, which is a component of body odor along with the components of tobacco smoke such as carbon monoxide and other secondary smoke contaminants. The sensor is also reactive to VOCs produced by the offgassing of new carpet and furnishings and is calibrated to a 3.5-volt DC output corresponding to 50% "good air quality" at 1,000 parts per million methane at 70°F (21°C) and 50% RH.

The sensor does not require field recalibration but must be maintained in a continuously powered mode. The manufacturer's literature recommended that the sensor be installed and powered for four weeks before final occupancy adjustments. Intermittent power interruptions require a sensitivity recovery time of 12 hours for a one-hour interruption and two days' recovery time for a power interruption of less than one day. The recommended field calibration procedure is to observe the indoor air quality and then make gradual setpoint adjustments until the desired air quality is achieved. Figure 1 shows a sensor output graph. Air quality is measured in "AQs," with 10 AQs corresponding to 10 volts DC and "perfect air," which would only be obtainable with synthetic air. Zero AQs or 0 volts DC would correspond to very poor air quality, such as that found in a wintertime thermal inversion with smog and other pollutants. The typical desirable indoor air quality level would correspond to between 5 to 7 AQs. In this range there would be a noticeable freshness to the air quality. Figures lower than this would indicate that noticeable levels of pollutants would be present.

CONTROL SELECTION

Figure 2 shows a schematic diagram of the mechanical unit along with the control hardware that was selected. The VOC air quality sensor is shown in the return duct. The packaged rooftop unit's mixed-air dampers and actuators were used, but a direct digital control (DDC) terminal unit controller with a built-in discharge air sensor capability and multistage heating-cooling control with outdoor air/return air function was installed. A

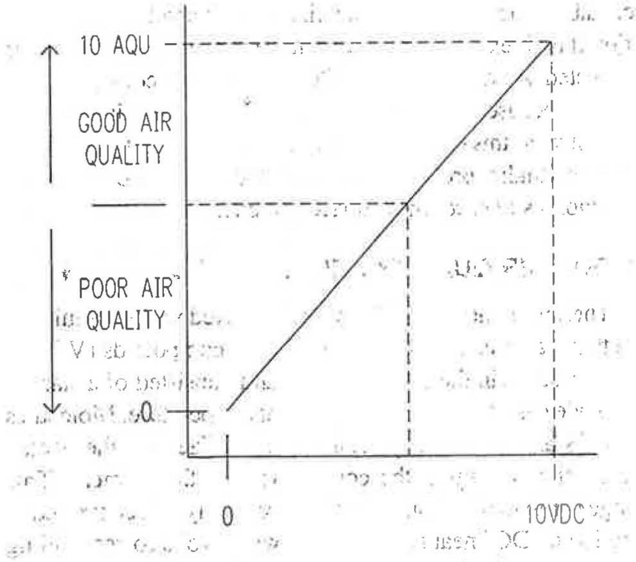

Figure 1 Multigas IAQ sensor calibration chart.

programmable DDC controller was used to input the signal from the air quality sensor and compare it against a setpoint in AQs. If the air quality was less than setpoint, then the mixed-air control signal to the outdoor air damper actuator was overridden and the outdoor air damper modulated to a position necessary to ventilate the space to maintain the desired air quality level. The discharge air sensor through the terminal unit controller cycled the gas heating to temper the outdoor air to avoid space temperature swings as much as possible. The terminal unit controller receiving a signal from its space temperature sensor provided all the desired occupied and unoccupied temperature control functions that were explained earlier in this paper. A constant-temperature (60°F [15°C]) mixed-air control loop was used for "occupied" operation in lieu of a minimum position control, providing the benefit of minimum outdoor air ventilation.

SYSTEM OPERATION

The system was installed and commissioned. A DDC network panel was installed temporarily to store sensor and

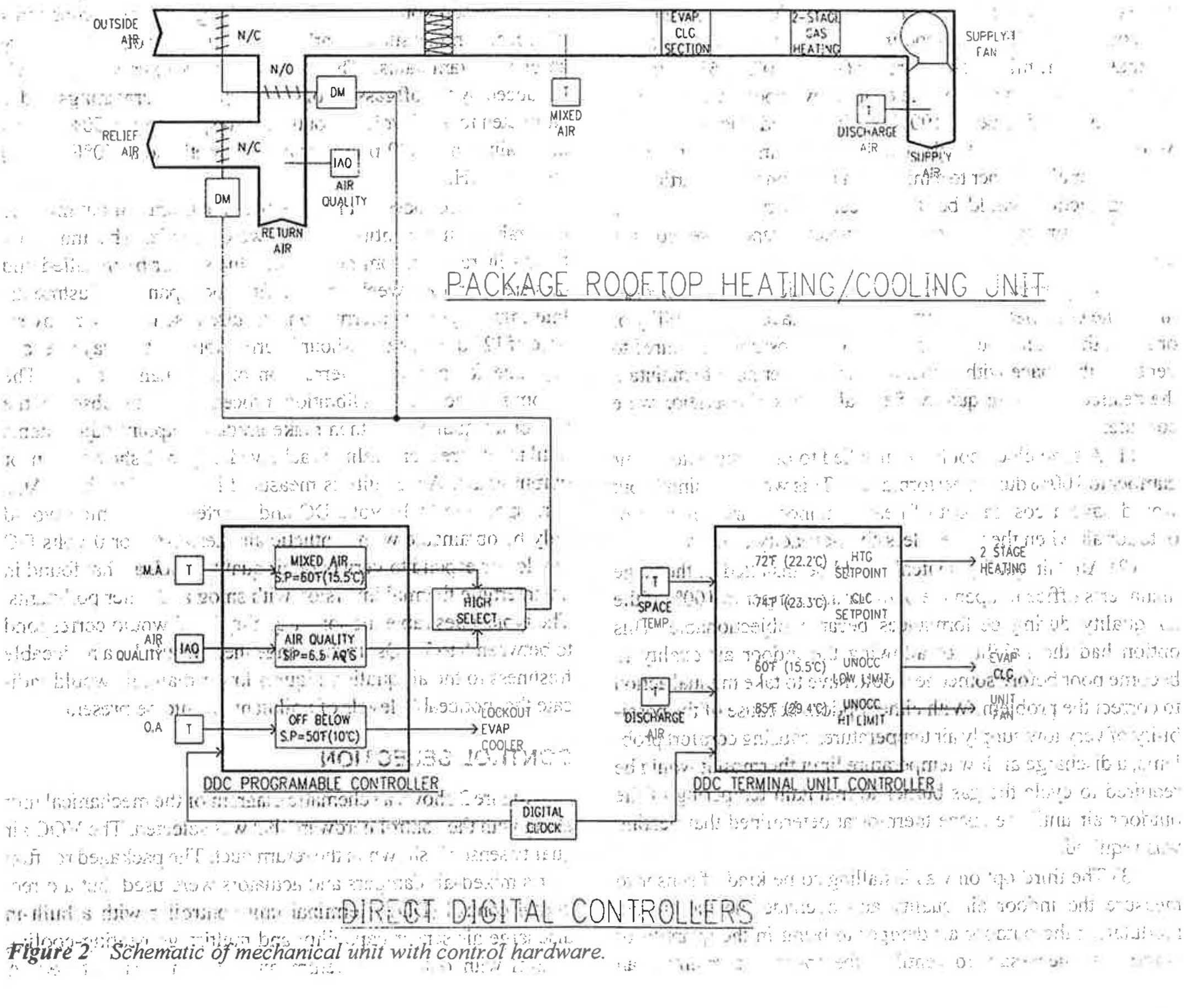

Figure 2 Schematic of mechanical unit with control hardware.

control system values and trend the control system operation so that the desired setpoint for air quality could be determined after observing a number of performances. The building that was renovated into the comedy club did not have any other DDC controls so the field panel was removed after the desired setpoint adjustments had been made. Figures 3 and 4 show graphs of the data collected by the field panel over several different days while it was connected to the DDC controllers. The outdoor air temperature, indoor air quality, zone temperature, and outdoor air damper position are graphed over time corresponding to 24 hours each day on February 18 and 19, 1995.

Figure 3 shows the building switching into "occupied" mode at 10:00 a.m. The outdoor air temperature only rises to about 60°F (15°C) by 2:00 p.m. and then starts to drop. The space temperature cycles in the heating mode around 70°F (21°C), and the outdoor air damper position stays around 60% until the first show starts at 7:00 p.m. The indoor air quality remains around 70% AQs during the same period. When the first show starts, the indoor air quality drops rapidly to less than 50% AQs. The outdoor air damper is modulated open to 100%, which keeps the air quality from dropping any further. The first show ends at about 9:00 p.m. and the air quality rapidly improves, returning the damper to the mixed-air control position of about 60%. At 10:00 p.m., the next show starts and the same cycle repeats itself, opening the damper to 100% outdoor air. The second show is over at about 11:30 p.m., the crowd leaves, and by midnight the air quality has improved and the outdoor air damper position returns to the mixed-air control. The system switches to "unoccupied" after midnight.

Figure 4 shows the system operation on the next day, February 19. On this day the outdoor air temperature rises to almost 70°F (21°C). The mixed-air controller opens the damper to the 100% position during the afternoon, but by 5:00 p.m. the outdoor air temperature has dropped to below 60°F (15°C), which allows the damper position to return to the mixed-air controller. At 7:00 p.m., the show starts and the indoor air quality drops rapidly. The damper position modulates again to 100% outdoor air, which prevents the air quality from getting any worse. The single show is over about 10:00 p.m. and the indoor air quality improves rapidly, returning the outdoor air damper to mixed-air control. The unit once again switches to the "unoccupied" mode around midnight.

From observing the rapid change in indoor air quality on multiple days it became apparent that the setpoint adjustment for the indoor air quality in AQs was about right. The temperature controls functioned normally, keeping the space temperature within desired limits. The air quality sensor rapidly detected the show beginning with the crowd showing up and starting to smoke. The instantaneous response of the demand-controlled ventilation system prevented the air quality from dropping below acceptable levels during the performances. Once the crowd left, the indoor air quality improved rapidly and the system reverted to normal mixed-air control.

One could estimate energy savings from not having to heat 100% outdoor air for non-showtime occupied periods on February 18, 1995, as shown in Figure 3. Assuming that the unit is on for 14 hours from 10:00 a.m. to midnight, one could take the average outdoor air temperature for the period along with the average outdoor air damper position and calculate the savings in heating a reduced amount of outdoor air up to the space temperature. A more accurate figure could be obtained by knowing the

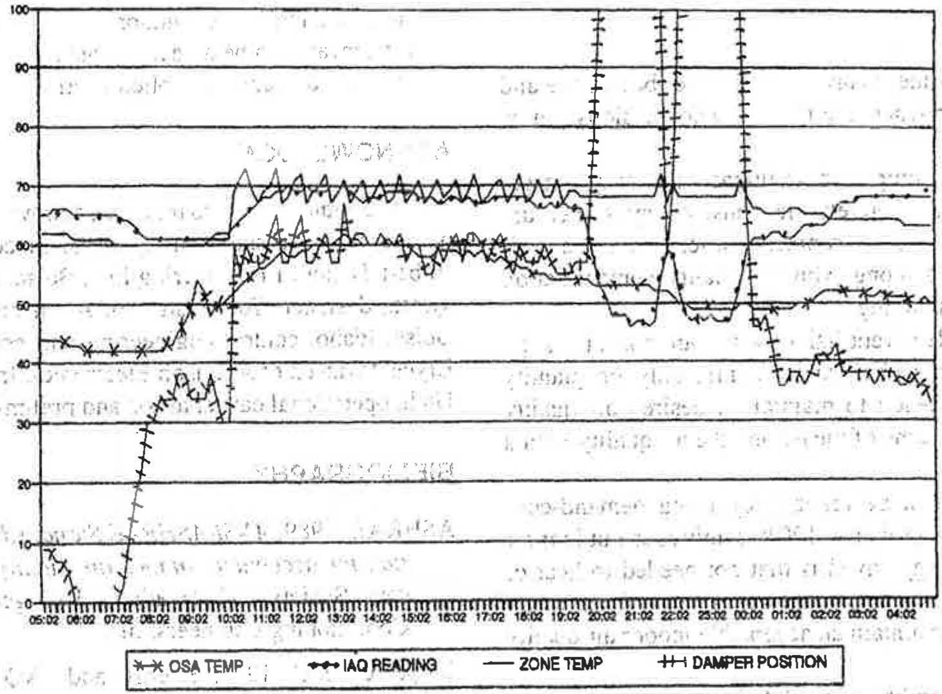

Figure 3 *IAQ data—February 18, 1995.*

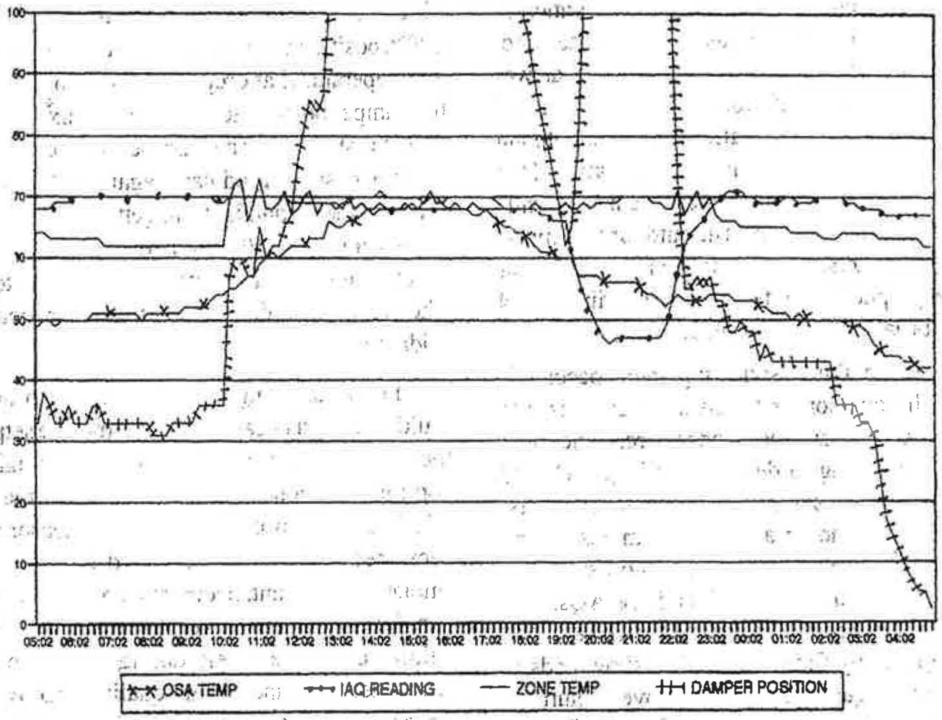

Figure 4 *IAQ data—February 19, 1995.*

supply air temperature, but unfortunately these data were not collected.

$$10(\text{nonshow hours}) \times 5800 \text{ cfm} \times 40\%(\text{OA}) \times (70 - 55°\text{F}) = 348\text{MBH/Day energy savings}$$

This corresponds to 102 kilowatts. One should not use this daily figure to calculate the savings for a year, as the outdoor air temperature and heating load would vary each day.

CONCLUSIONS

After installing the control system described above and observing its performance, the following conclusions can be made.

1. Volatile organic compound, multigas air quality sensors are an acceptable, relatively inexpensive way to measure indoor air quality in environments where there are variable people loads along with air quality contamination sources such as smoking.
2. Demand-controlled ventilation with outdoor air is an acceptable control technique that admits only the quantity of outdoor air needed to maintain a desired air quality when there is a means of measuring the air quality with a control sensor.
3. Energy savings can be realized by using demand-controlled ventilation in lieu of 100% continuous outdoor air systems. The energy saved is that not needed to heat or cool excessive amounts of outdoor air beyond that quantity necessary to maintain an acceptable indoor air quality level.
4. The long-term stability of the VOC-type air quality sensor appears to be adequate because the system described in this paper has been installed for more than two years, operating as designed, without requiring any recalibration.
5. The multigas VOC indoor air quality sensor used in the project appeared to react favorably to people load. Whether other sensors, such as the CO_2-type sensor, track occupancy load more accurately is not for discussion in this paper. Combining the requirement to sense both people count and overall indoor air quality, the multigas sensor appeared to be an acceptable selection for a sensor to drive the demand-controlled ventilation system.

ACKNOWLEDGMENTS

The author wishes to thank the following individuals who have contributed substantially to the success of this project: Robert Tikker of Engineering Inc., Boise, Idaho, mechanical system designer; Tony Sutton of Atkinson Electronics, Inc., Boise, Idaho, control engineering and commissioning; and Bryce Atkinson of Atkinson Electronics, Inc., Salt Lake City, Utah, operational data analysis and preliminary write-up.

BIBLIOGRAPHY

ASHRAE. 1989. *ANSI/ASHRAE Standard 62-1989, Ventilation for acceptable indoor air quality.* Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Carpenter, S.C. 1996. Energy and IAQ impacts of CO_2 demand-controlled ventilation systems. *ASHRAE Transactions* 102(2).

Elovitz, D.M. 1995. Minimum outside air control methods for VAV systems. *ASHRAE Transactions* 101(2): 613-618.

Haghighat, F., and G. Donnini. 1992. IAQ and energy-management by demand controlled ventilation. *Environmental Technology* 13: 351-359.

Levenhagen, J.I. 1992. Control systems to comply with ASHRAE standard 62-1989. *ASHRAE Journal* 34(9): 40-44.

Staefa. 1995. Application bulletin S1-02.50. San Diego, Calif.: Staefa Control Systems, Inc.