

Field Experience Controlling a Dedicated Outdoor Air System

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ABSTRACT

Dedicated outdoor air systems (DOAS) are first defined. Then the single-space operating system, whose performance is presented here, is shown to conform to that definition. The controlled components are identified, along with the minimal instrumentation needed for control. Control of each component individually, and in the context of the whole, is presented. Supplemental controls to reduce terminal reheat energy consumption in multi-space applications are developed. Issues surrounding the use of CO₂-based demand-controlled ventilation, in light of ASHRAE Standard 62.1-2004 (former addendum 62n), are presented. Finally, experimental data are presented on the overall success of the DOAS design and control to provide air distribution and thermal comfort.

INTRODUCTION

The use of dedicated outdoor air systems (DOAS) has increased significantly in the last few years; however, little has been presented in the literature on experiences controlling them. The central thrust of this paper is to explore the DOAS control needs and present operational field experience. Important performance metrics include controllability, response characteristics, complexity, role of demand-controlled ventilation, continual diagnostic capabilities, and the resulting thermal comfort and air diffusion performance.

DOAS DEFINED FOR THIS PAPER

Many papers and articles have appeared in the literature (Coad 1999; Butler 2004; Mumma 2001a; Morse 2003; Fischer and Bayer 2003; Manuel 2003) addressing DOAS. Those papers illustrate the diversity of DOAS concepts presently held by the US HVAC&R engineering design commu-

nity. Therefore, in an attempt to minimize potential confusion, DOAS will be defined for this paper.

The block diagram in Figure 1 illustrates, in simplistic terms, the DOAS whose control is addressed in this paper. Key elements of the DOAS approach are:

- One hundred percent outdoor air (OA) in quantities defined by ASHRAE Standard 62.1-2004 is supplied directly to each space of the building via its own system of ductwork and diffusers.
- No recirculated air is used.
- For most situations, the DOAS is constant volume.
- The supply air dew-point temperature (DPT) is selected sufficiently low to enable it to handle the entire space latent load, thereby decoupling the space sensible and latent loads.

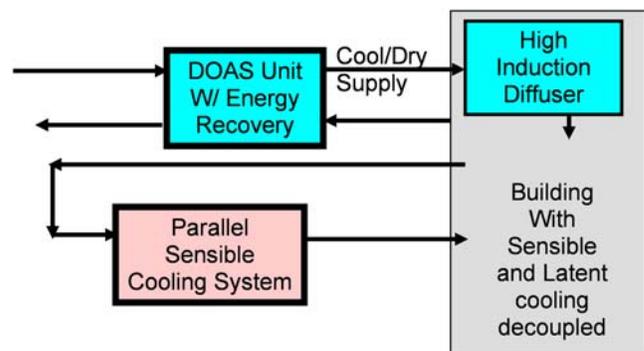


Figure 1 DOAS block diagram.

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- The supply air dry-bulb temperature is set at the required DPT, especially for low occupancy density spaces.
- Total energy recovery, or enthalpy wheel (EW), is utilized for energy conservation and chiller-boiler-humidifier size reduction and to comply with ANSI/ASHRAE Standard 90.1-2004. It requires an EW for systems supplying greater than 5000 cfm (2355 L/s) and 70% OA. Control of the EW is a significant issue.
- Many parallel sensible cooling technologies are possible, including fan coil units, packaged terminal devices, air systems such as variable air volume (VAV), and ceiling radiant cooling. For the operational system discussed in this paper, ceiling radiant cooling is the parallel system. Control of the radiant cooling will not be the subject of this paper, since it is addressed in earlier papers (Mumma and Jeong 2005).
- Utilize high induction ceiling diffusers (HICD) to ensure adequate air motion and delivery of the ventilation air to each person's breathing zone. The use of HICDs permits the use of low supply air temperatures (45°F-52°F [7°C-11°C]), impacting the sensible cooling capacity of the DOAS and the resulting size and first cost of the parallel sensible cooling system. Note: the increasingly popular underfloor air distribution (UFAD) approach (Woods 2004) may be suitable for ventilation purposes in the special case of displacement ventilation (DV); however, in general, UFAD supply air is a mixture of OA and recirculated air and cannot ensure proper ventilation (Mumma and Lee 1998).
- The operating DOAS is serving a leaky, early 1900s second floor 3200 ft² (300 m²) fifth-year architecture studio space (40 ft by 80 ft [12 m by 24 m] floor plan), occupied around the clock, 7 days per week, housing 40 students and 5 or so faculty consultants along with their computers, overhead lights, task lamps, refrigerators, coffee pots etc. It has one glazed SW exterior exposure (384 ft² [36 m²] single-glazed movable sash, upper and lower sections), three interior partitions adjacent to non-conditioned spaces, and two doors that open to unconditioned space. The floor and ceiling are also adjacent to unconditioned spaces. The ceiling height is 14 ft (4 m), with pendent illumination at the 9 ft (3 m) elevation. There are eight evenly spaced, two-way 20 ft (6m) throw, high induction diffusers also at the 9 ft (3 m) elevation, delivering 850 ft³/min (400 L/s) of 100% outdoor ventilation air. There are eight free-hanging CRCPs (i.e., heat transfer on both top and bottom surfaces), each 2 ft by 40 ft (0.6 m by 12 m) (20% of the floor area), evenly spaced and at the 9 ft (3 m) elevation. Finally, the DOAS-radiant system has no heating equipment to control. The space does have old cast iron steam radiators under the windows, with uninsulated steam piping traveling vertically to equipment on the third floor. The steam shutoff valves are closed on all radiators, so the only heat from the original system is uncon-

trolled heat loss from a few pipes running vertically through the space.

An alternate use of the term *DOAS*, found in the literature, conditions the OA in parallel with recirculated air, often to a DPT sufficient to handle all of the space latent load, then introduces the OA downstream of the main VAV AHU coils (Khattar and Brandemuehl 2002). The mixture is then delivered to the space via a single duct. This approach minimizes the space humidity control problems, characteristic of VAV systems at off-design loads, but does nothing to address the ventilation distribution problems that characterize VAV systems (Mumma and Lee 1998). Rarely do these arrangements utilize an EW, since ANSI/ASHRAE Standard 90.1-2004 does not require it for conventional VAV systems.

THE CONTROLLED DEVICES

The controlled devices in the DOAS system serving a space in a university setting consist of: supply and return fans, OA and relief dampers, EW, cooling coil (CC), chiller, and chilled water supply (CHWS) pumps. Figure 2 is a schematic of the actual system being controlled, illustrating all of the controlled devices and associated instrumentation.

Fans And Dampers

The fans and dampers are either on/open or off/closed since the DOAS is constant volume. Their on/off status is dictated by:

- Scheduled occupancy.
- Frost formation potential in the EW, to be discussed with the EW control.
- Sash position (one of three condensation control options for the ceiling radiant cooling technology). The two exterior doors and twelve large movable sash windows are fitted with normally open magnetic proximity switches wired in series to detect an opening. One of four selections available with the option for condensation control places the system in the unoccupied mode whenever a door or window remains open for ten minutes or more, placing the fan/dampers in the off/closed position. In fact when in the unoccupied mode, everything is off (chiller, pumps, fans, EW).
- Occupied space low-temperature limit to prevent overcooling with the OA during the winter (necessary since the system has no active preheat, heat, or reheat). The EW recovers the majority of the internal generation (illumination, both overhead and task; computers and peripheries; refrigerators and other convenience appliances; and people); however, when the OA is very cold, the space temperature will drop below the space low limit (i.e., adjustable and set around 65°F [18°C]) and mechanical equipment shuts down and the dampers close. This rarely occurs and only when the space is empty.

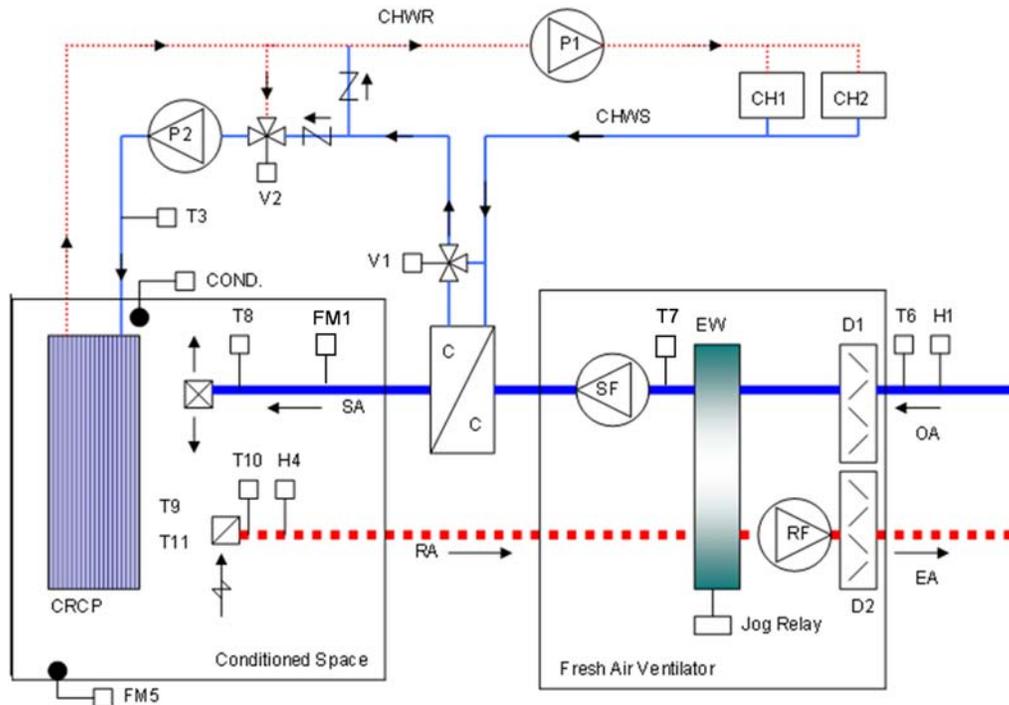


Figure 2 DOAS schematic, with controlled devices and instrumentation.

ENTHALPY WHEEL

In the DOAS control discussed in this paper, the EW has two operating modes: on or off. Short cycling associated with on-off control is avoided with a minimum “on/off” relay. The relay is set to prevent the EW from receiving more than two “on” signals per hour. When the OA DPT is above setpoint (adjustable but about 52°F [11°C]) and the OA enthalpy exceeds the space enthalpy, the EW is on. In the small triangle illustrated in Figure 3 where the OA DPT is above setpoint but its enthalpy is below that of the return, the EW is off. Operating the EW in this triangle would increase the enthalpy of the air entering the CC and, thus, the load on the CC.

EW manufacturers seldom sell equipment with the ability to move air through the wheel when it is not rotating, since it can become an unwanted filter (i.e., particles collect on the leading edges of the wheel flutes). An alternative might be to employ bypass ductwork and dampers, but since the EW equipment is already large, it is seldom done. In the EW control discussed here, even when it is thermodynamically beneficial to stop the EW rotation (i.e., when the OA conditions are in the small triangle of Figure 3), it is forced to rotate three minutes per hour. This control has successfully kept the EW from becoming an unwanted filter since the airflow through the wheel is counterflow. Therefore, particles collected on the leading edge of the stationary wheel flutes are blown off when the wheel rotates into the airstream flowing in

the opposite direction. At 20 rpm, the wheel experiences 60 flow reversals in the three-minute cleaning cycle—once an hour.

When the OA DPT is below setpoint (OA DPT less than 52°F [11°C]), the OA is dry and dehumidification is not required. In fact, humidification may be needed. The controls enable the system operator to select between either temperature or temperature and humidity control.

- If temperature control has been selected, and the OA dry-bulb temperature (DBT) is higher than that required by the system, the CC must provide sensible cooling. If the OA DBT is lower than required, the EW is cycled on-off as necessary to maintain space temperature (minimum on-off cycle is currently set at 15 min to limit the number of cycles per hour). The space DBT will rise and fall within a 1°F (0.5°C) range on a minimum one-hour cycle (i.e., slow dynamics). Cold OA supply has presented no comfort problems because the high induction diffusers mix the air so well that the discharge jet temperature nearly equals the room air temperature within about 18 inches from the diffuser. And when it is cold, the EW is operating, continually tempering the OA.
- If temperature and humidity control has been selected, the EW is cycled on-off to maintain the desired space DPT. This can result in the OA being heated more than

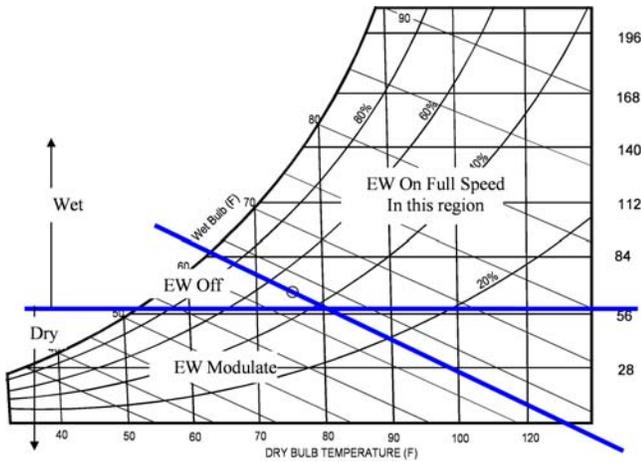


Figure 3 Psychrometric chart showing the “EW off” triangle.

desired, in which case the CC provides sensible cooling (see next section for control). Alternatively, the sensible cooling can be delivered with the parallel system. In either case, under these “dry” weather conditions (often a large percentage of the annual occupied hours); water-side economizer can save significant energy.

The operating system discussed here did not use any type of economizer for cooling (other than the constant volume ventilation air delivered by the DOAS, which is able to provide 100% free cooling from mid-October through mid-April). Economizer options will not be discussed further in this paper.

A final but important aspect of the control of the EW is frost prevention. During the winter, frost will form on a rotating EW if the OA temperature is sufficiently low and the space DPT is around 55°F (13°C). The potential for frost formation on the wheel exists if the straight line between the OA and space thermodynamic state points crosses the saturation curve on the psychrometric chart. The controls used for the EW in this operating DOAS continually check to make sure that the saturation curve is not crossed and the OA temperature is above freezing. If frost formation is possible, the control logic puts the system into the unoccupied mode, shutting down the fans and closing the dampers. Due to the lack of humidification in the space (other than the occupants’ latent contribution), the space dew-point temperatures, when it is cold outside, remain low enough that the line between the state points does not cross the saturation curve. As a result, the system has, to date, not shut down to prevent EW frosting. A tight building, with either some humidification or longer periods of full occupancy, would either shut down or use another means of frost control. The author’s preference for another means of frost control, unlike most EW manufacturers due to

first cost, would be preheat. A discussion of frost control prevention and preheat was developed in an earlier paper (Mumma 2001b).

COOLING COIL CONTROL VALVE

The CC is used as needed to both cool and dehumidify the OA. In the operating single-space DOAS case considered in this paper, the supply air temperature is modulated as necessary to avoid overcooling but not allowed to drop below 52°F (11°C). Therefore, terminal reheat is never required. By limiting the supply air DPT to no lower than the 52°F (11°C) setpoint, the DOAS is rarely able to meet the entire space sensible load. The balance of the sensible load is taken care of by the parallel ceiling radiant cooling panel (CRCP) system.

A three-way mixing control valve, supplied with approximately 45°F (7°C) chilled brine solution, is modulated to control the supply air DPT. Much of the time, this valve is nearly wide open, with little modulation required. This occurs for two reasons. First, the brine solution flow to the CC is half of design when only one of the two compressor loops is operating—the case for the majority of time. Second, preconditioning the OA with the EW causes the OA conditions, seen by the CC, to be almost constant (reducing the maximum enthalpy variation from about 26 Btu/lb (60 kJ/kg) to about 3.5 Btu/lb (8 kJ/kg). This is an important control benefit realized with the EW, i.e., very stable OA loads result in extremely stable control.

CHILLER AND CHWS PUMPS

The chiller consists of two distinctly separate 5 ton (18 kW) refrigeration/compressor circuits in parallel. Each circuit has its own air-cooled condenser. The units are designed for computer room applications and are equipped with hot gas bypass for load trim control, if desired, and heaters for low ambient air temperature operation (0°F [-18°C]). The chilled water system utilizes two CHWS pumps in series, with each pump devoted to one of the chiller circuits. When only one refrigeration circuit is operating, along with its CHWS pump, the brine circuit through the other refrigeration circuit is valved off, and its CHWS pump is off. The chiller has all of the standard safeties of high and low temperature, pressure, flow switches, etc. The DOAS control simply sends an “on” signal (switch closure) to the chiller when it is needed.

The logic used to establish the need for an “on” signal to the chiller is based upon three criteria. First, at some times of the year, the space is not used around the clock. So a time clock schedules the on and off time for the chiller. The clock can be easily reset as use patterns change. Second, the chiller is only enabled when the OA temperature is above setpoint. Experience has shown that the chiller is not generally needed when the OA temperature is below 50°F (10°C). A logical “if” statement is used with a hysteresis of 5°F (3°C). In other words, if the OA temperature is above 55°F (13°C), the chiller is enabled and will not be disabled by this logical device until the temperature drops below 50°F (10°C). This has proven to be a satisfactory setpoint and hysteresis for this application.

Hysteresis is an important feature of control logic to avoid short cycling!

Third, it is possible to operate the entire system as an on-off control. Rather than allowing the chiller to unload as the CC and radiant panel controls modulate closed, they can be forced to be nonresponsive (except for condensation control) to space temperature and the chiller operated at nearly full capacity until the space temperature falls to a desired level. When the chiller/system is operated this way, the chiller is activated when the space temperature reaches 75.5°F (24.2°C) and deactivated when it drops to 73°F (23°C). Chiller cycling primarily occurs during the spring and fall when it is cool outside. Otherwise, for most cases, one of the two 5 ton (18 kW) refrigerant circuits is just able to meet the load and does not cycle.

Should the space temperature rise above setpoint (adjustable but about 76°F [24°C]) with just one refrigeration circuit operating, the second brine loop isolation valve is opened, and the refrigeration circuit compressor and brine loop pump are activated. The second compressor operates with an adjustable (about 2°F [1°C]) hysteresis.

When the chiller is cycling on and off, the capacity and dynamics of the space/system are such that the chiller runs for at least one hour. When it is turned off, it stays off for at least one hour. Part of the reason that the chiller on-off cycles are as long as they are is that the CHWS pump continues to operate for 40 minutes after the chiller is shut down (part of the radiant cooling panel condensation control). As a result, cooling continues until the CHWS has heated up to nearly the OA temperature. While the space humidity increases without the chiller, it does so gradually. The EW is able to transfer much of the moisture in the incoming stream to the exhaust airstream, slowing the room humidity increase when the chiller is off. Of course it does the same thing when the chiller is on, but the room RH remains essentially constant when the DOAS and chiller are operating.

INSTRUMENTATION

The instrumentation necessary for a DOAS (Figure 2) is, for the most part, not much different from that used in conventional all-air VAV systems. The major objectives of the DOAS controls are: thermal comfort, pressurization, continual performance monitoring to detect problems before they lead to IAQ symptoms, and perhaps demand-controlled ventilation (DCV).

CONTINUAL PERFORMANCE MONITORING

This performance assessment need, achieved by continual monitoring, is not unique to buildings served by DOASs. In fact (Woods 2002), when the entire nonindustrial building stock served by conventional HVAC systems is analyzed, 20% to 30% have problems either with building-related illnesses (5% to 10%) or sick building syndrome (10% to 25%). These facilities began as buildings without known problems and then degraded. In all facilities, early detection and correction are

required to avoid disabling problems later. Woods (2002) estimates the consequences of ongoing system performance degradation in the US are as follows: 20% of workers are experiencing health-related symptoms, 20% of workers are experiencing hampered performance, and 50% of workers have lost confidence in management's ability to deal with the situation. A major economic investment is needed to mitigate the problem or renovate/replace the facility to recover "goodwill" after system performance degradation. Fisk (2002) estimates the economic impact on US businesses is as much as \$208 billion per year, including increased respiratory diseases (\$6 billion to \$14 billion per year), increased asthma and allergies (\$1 billion to \$4 billion per year), sick building syndrome (\$10 billion to \$30 billion per year), and reduced worker productivity (\$20 billion to \$160 billion per year).

Causes of system performance degradation can be divided into three categories: insufficient diagnostic and alarm tools built into the system for early warning of degradation, a lack of awareness of problem buildings' economic consequences, and indifference. Degradation of the DOAS performance can occur in three major areas: the supply air quantity can be compromised, the building pressurization function can be compromised, and the supply air conditions can be compromised.

Compromised supply air quantity could result from failures in the supply fan motor, bearings, or belts. It could also be the result of dirt loading at the filters, enthalpy wheel, CC, or other unintended filters such as grilles, diffusers, etc. These could all impact the DOAS's ability to meet the ventilation requirements, latent load duty, and its portion of the space sensible load. A failure to deliver in these areas will be immediately noticed by the occupants. Degraded air delivery can be directly monitored with a flow-measuring station in the supply air ductwork, as indicated by FM 1 in Figure 2.

If the magnitude of envelope leakage entering or leaving the building becomes excessive (compromised building pressurization), the enthalpy wheel thermal performance degrades. This adversely impacts the supply air conditions and ability to cool and dehumidify. The loss of building positive pressure will cause leakage through the envelope, leading to excessive latent loads in the space that may be beyond the DOAS system's capacity. Humidity control problems may develop along with the potential for mold and fungi formation. These conditions may be noticed by the building occupants once the situation has become critical, which is generally too late. Therefore, the system must have an instrument to monitor building pressurization. While many tools exist to achieve this task, in this project an envelope flow magnitude and direction sensor is employed. This instrument is labeled FM 5 in Figure 2.

Degradation in the ability to hold the desired supply air temperature (compromised supply air conditions) impacts the DOAS's ability to deliver the required space sensible and latent cooling. Such degradation will lead to occupant thermal discomfort, diminished humidity control, and potential micro-

bial growth problems. The supply air thermal condition degradation can be caused by something as simple as enthalpy wheel drive belt or motor failure, deterioration of the enthalpy wheel effectiveness, loss of cooling capacity at the CC (because of insufficient chilled water temperatures or flows, or compromised direct expansion function), or fouling of the CC. The instrumentation used in this project to sense enthalpy wheel sensible effectiveness are three temperature sensors. Two are in the OA stream, one before and one after the enthalpy wheel (identified as T6, and T7, in Figure 2), and a third is in the relief air upstream of the EW (identified as T10 in Figure 2). CC performance degradation is sensed by measuring the supply air temperature (identified as T8 in Figure 2, also used for control) downstream of the CC.

MINIMIZING TERMINAL REHEAT IN MULTI-SPACE FACILITIES

While the single-space operating system discussed in this paper uses no terminal reheat, an extremely important design decision that must be made when designing a multi-space constant volume DOAS is the method of preventing overcooling at part-load conditions. In general, to decouple the space sensible and latent loads, the supply air dew-point temperature needs to be below 50°F (10°C), which is accomplished with a conventional CC (as opposed to the use of an active desiccant). A common practice in the industry, then, is to reheat the ventilation air to a neutral temperature with recovered heat (either from the refrigeration condenser or the exhaust airstream via a sensible wheel, SW). Such a decision, while avoiding terminal reheat, impacts the first cost of the parallel system since the neutral temperature ventilation air has no capacity to provide space sensible cooling. Cold (45°F-50°F [7°C-10°C]) DOAS supply air can reduce the first cost of the parallel system from 32% for low occupancy density spaces (offices) to 90% for certain high occupancy density spaces (conference rooms, classrooms). Further, if the reheat is achieved with heat recovery from the exhaust airstream, the chiller load increases (Shank and Mumma 2001).

There are two ways to gain the advantage of the full sensible cooling capacity of cold DOAS supply air temperatures in multiple space applications, while limiting terminal reheat. Both involve added complexity, hardware, and controls. While discussed next, it should be noted that they are not used in the operating DOAS discussed in this paper.

The first method is to apply demand-controlled ventilation, thus reducing the supply air quantity from the DOAS with occupancy. The benefits of any type of demand-

controlled ventilation approach for reducing the supply air quantity and the need for terminal reheat were diminished with the introduction of ANSI/ASHRAE Standard 62.1-2004, which specifies a floor and an occupancy ventilation component. Nonetheless, the ventilation air can be supplied cold and the flow rates reduced with occupancy, reducing the energy consumed for terminal reheat (reheat is permitted by ANSI/ASHRAE Standard 90.1-2004).

The second method is to use a critical space supply air temperature (SAT) reset with a constant volume DOAS. In this control approach, the supply air DBT (not the DPT) from the DOAS would be elevated as high as possible so the temperature of the space with a load nearest its design is just met, while maintaining the dehumidification requirements (mechanical dehumidification is required until the OA DPT drops below about 50°F [10°C]). The supply air would be tempered as needed with heat from the refrigeration condenser or the exhaust airstream using a sensible wheel or plate heat exchanger. This will not totally eliminate terminal reheat energy use but will reduce it. The reheat energy saved is a function of the part-load occupancy levels and distribution.

Each zone/space would be equipped with both a parallel sensible cooling appliance (radiant cooling panels, fan coil units, VAV, etc.) and a terminal reheat coil. The terminal reheat and the parallel cooling appliance must work in sequence. Then, at off-design loads/occupancies, the DOAS supply air temperature would be elevated until one space temperature exceeds setpoint. In that critical space, the parallel system would be at full capacity. The supply air temperature can be elevated no farther, and terminal reheat in any of the other spaces would be reduced.

DEMAND-CONTROLLED VENTILATION

Standard 62.1-2004 introduces a ventilation rate procedure based upon a floor component and an occupant component. The standard also permits dynamic reset of the design outdoor air intake flow during several conditions. One condition quoted from 62.1-2004 is “an estimate of occupancy or ventilation rate per person using occupancy sensors such as those based on indoor CO₂ concentrations.”

Controls to strictly maintain the minimum floor and occupant components may be tricky. To illustrate, consider this controlled operating facility designed for 45 people, i.e., students and their studio critics. Employing the design data from 62.1-2004, Table 6.1, the ventilation rates are summarized in the table below:

Floor Area, ft ² (m ²)	OA, cfm/ft ² (L/s-m ²)	Floor Component OA, cfm (L/s)	Occupancy	cfm/Person (L/s-person)	Occupant Component, cfm (L/s)	Total OA, cfm (L/s)	Total OA/Person, cfm/Person (L/s-person)	Resulting Space CO ₂ Concentration, ppm
3,200 (300)	0.12 (0.6)	384 (180)	45	10 (5)	450 (225)	834 (405)	18.6 (9)	1,000
3,200 (300)	0.12 (0.6)	384 (180)	20	10 (5)	200 (100)	385 (280)	29 (14)	750

Note: the CO₂ concentration setpoint is now a function of occupancy, changing from 1000 ppm at full occupancy to 750 ppm at 44% occupancy! When the ventilation was occupancy-only based, as was the case prior to 62.1-2004, the CO₂ setpoint was independent of occupancy.

Given this new and difficult situation, an experiment to test the ability of space CO₂ concentrations to accurately predict dynamic occupancy was undertaken. If the dynamic occupancy can be determined, the space CO₂ concentration setpoint can be computed dynamically, and conventional PID control loops may be used to modulate the OA flow to meet the new 62.1-2004 requirements.

The supply and relief air CO₂ concentrations and the supply airflow rate are monitored continually. The signals are connected to the system's DDC hardware, where the software computes the dynamic occupancy.

COMPUTING OCCUPANCY FROM MEASURED DATA

Both steady-state and transient equations can be used to solve for occupancy (Ke and Mumma 1997). The transient equation in difference form is

$$Pep = (V * (N - N_1) / \Delta\tau + SA * (N - C_i)) / (G * 1,000,000), \quad (1)$$

where

- Pep = number of occupants,
- V = the space air volume, ft³,
- N = the space CO₂ concentration at the present time step, ppm,
- N_1 = the space CO₂ concentration one time step back, ppm,
- $\Delta\tau$ = the time step, min,
- SA = the supply airflow rate, scfm,
- C_i = the CO₂ concentration in the supply air, ppm, and
- G = the CO₂ generation rate per person, scfm.

Implicit in the equation is the assumption that the space is held at a positive pressure (part of the continual performance monitoring instrumentation or flow measuring transducer FM5 of Figure 2), so that air with a CO₂ concentration higher or lower than the space air is not drawn inward, which would alter the concentration balances. When the space is held at a positive pressure, all of the room air leaves with an assumed uniform CO₂ concentration equal to that measured in the return airstream, an excellent assumption with high induction diffusers (Novoselac and Srebric 2002). In the case of the test facility, the use of high induction supply air diffusers produces well-mixed air (to be discussed later).

CO₂-BASED ESTIMATES, OBSERVED OCCUPANCY

Walk-through occupancy counts were conducted during several months and compared with the computed occupancy. The actual occupancy count agrees with the estimated occupancy count within two people. It also gives accurate counts

when there is a rapid change in occupancy since the transient concentration equation was used, not the steady-state concentration equation.

POTENTIAL TO DYNAMICALLY RESET THE VENTILATION RATE

For the system and control discussed in this paper, the ventilation was not dynamically reset using CO₂-based occupancy estimates. A procedure for doing so, however, was published (Mumma 2004) in an article titled, "Transient Occupancy Ventilation By Monitoring CO₂." To date, the authors are unaware of any industry-accepted method of achieving DCV under 62.1-2004.

ADPI ACHIEVED WITH THIS SYSTEM AND CONTROL

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 percentage of recirculated air. This has led some in the industry to question the DOAS's ability to provide sufficient air movement to control comfort requirements.

Six overhead Thermal-Core 24 in. (0.6 m) two-way blow high-induction diffusers (originally designed for low air temperature applications, thus using an injection molded core to avoid condensation), spaced uniformly at the 9 ft (2.8 m) elevation provide 108 cfm (51 L/s) each, or a total of 650 cfm (304 L/s) of 100% OA to that part of the space tested. At the ventilation air delivery rate, 0.325 cfm/ft² (1.7 L/s per m²) 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. 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 between 37°F and 39°F (3°C and 4°C) and the wind velocity around 5 mph (8 km/h).

As noted above, the test space has no active means of heating other than the 24 hours/day-7 days/week 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 EW. During the testing, the supply air temperature (SAT) was between 55°F and 57°F (13°C and 14°C).

EFFECTIVE DRAFT TEMPERATURE

Thermal comfort is a function of the following variables (ASHRAE 2001) that influence metabolic heat transfer:

1. Dry-bulb temperature (DBT)
2. Relative humidity
3. Mean radiant temperature

4. Air movement
5. Metabolism
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:

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

where

- 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 (Nevins and Ward 1968) 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 average room air temperature was 76.1°F (24.5°C). While a winter average room temperature of 76.1°F (24.5°C) may seem high and energy wasteful, it is the condition that the occupants requested for comfort, and the OA was completely tempered with recovered heat at the EW (so no heating energy was used). The data were collected with a low-velocity analyzer (DISA Type 54N50) capable of providing mean values for both the local velocity and temperature. The average room temperature was measured at the common return with a field-calibrated commercial-grade thermistor. The outdoor air conditions were from the campus weather station.

DATA ANALYSIS

The EDT resulting from the 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 toward the minus side, bringing all of the observations within the -3°F to 2°F (-1.7°C and 1°C) EDT range.

The 35 mean local temperature-velocity observation pairs are presented in Figure 4. 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).

THERMAL COMFORT ACHIEVED WITH THIS SYSTEM AND CONTROL

A person who is thermally neutral does not know whether he/she would like to be warmer or cooler. The subjective and physiological reaction of a person to the thermal environment is determined by the rates of bodily heat generation and heat emission.

The six parameters impacting thermal comfort discussed above can be measured and used to predict the human subjective response to any given combination of environment, cloth-

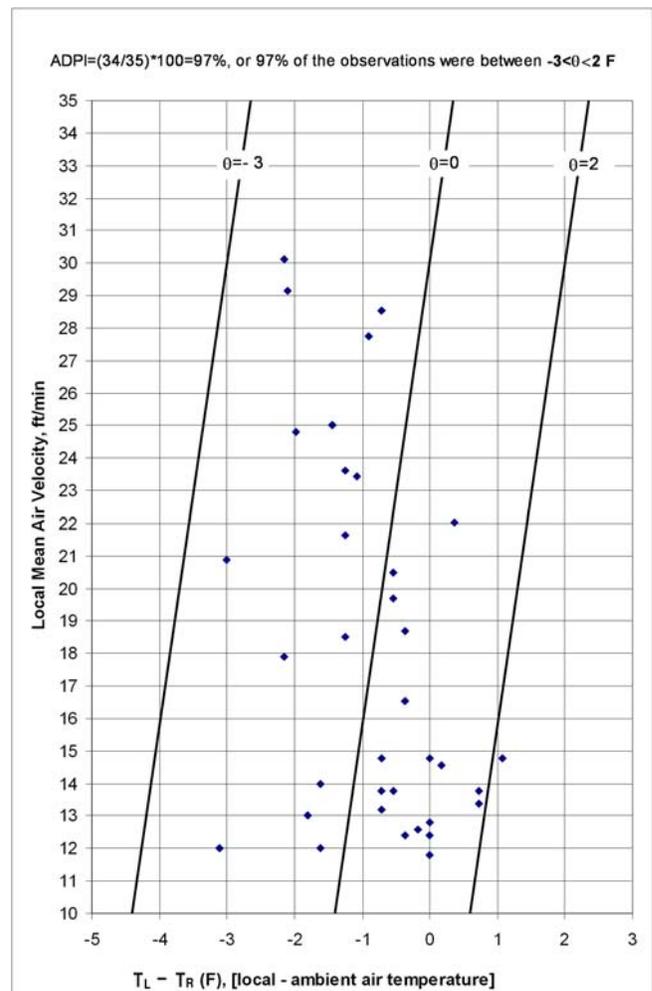


Figure 4 Effective draft temperature, θ .

ing, and activity level. These reactions follow a normal distribution about a mean, which is termed the predicted mean vote (PMV) (ASHRAE 2001). The PMV is an index that predicts the mean value of the subjective ratings of a large group of people on a seven-point thermal-sensation scale as follows: +3 = hot, +2 = warm, +1 = slightly warm, 0 = neutral, -1 = slightly cool, -2 = cool, and -3 = cold. While not everyone has identical thermal comfort preferences, no more than 95% of the occupants of any given thermal environment will be satisfied. Satisfying 80% of the occupants is a more realistic and more common goal. A related indicator of thermal comfort is the predicted percent dissatisfied (PPD). It is an index that predicts the percentage of a large group of people likely to feel thermally uncomfortable, i.e., voting +3 or +2. There is a functional relationship between PMV and PPD, but it will not be presented here.

Thermal Comfort Measurement Instrumentation

A thermal comfort meter can measure the influence of air motion, temperature, and mean radiant temperature on thermal comfort for prescribed values of clothing, activity level, and space humidity. The instrument uses a heated ellipsoidal (6.25 in. \times 2.25 in. [16 cm \times 6 cm] major and minor axis) transducer designed to simulate the thermal pattern of a human being. It contains a surface temperature sensor and a surface-heating element whose power is adjusted automatically by the thermal comfort meter to bring the surface to a temperature similar to that of a thermally comfortable human. The rate of heat production needed to attain this temperature is used as a measure of the environmental conditions.

The shape of the transducer is determined by the need to obtain the same ratio of horizontal and vertical projected radiant surface areas as for a human being. This is important where significant difference between horizontal and vertical radiation temperature can occur, as with overhead lighting or when CRCPs are used.

With the experimental results reported here, the occupants were assumed to have a clothing insulation value of 1 clo. This corresponds to an office worker dressed in slacks, shirt, shoes and socks. The occupants were also assumed to be doing sedentary work, with a metabolic rate of 1.2 met.

Test Space Conditions

The test thermal conditions were: room dry-bulb temperature = 73°F (23°C); room DPT = 53°F (12°C); average CRCP surface temperature = 60°F (16°C); DOAS supply air-temperature = 62°F (17°C); and average temperature of the ceiling = 74°F (23°C).

Thermal Comfort Measurements

Measurements were taken systematically throughout the space, directly under the radiant panels, directly under the illumination, near the exterior windows, near interior walls, and interstitially. The variations in measurements were very modest, and can be summarized as follows:

- Predicted Mean Vote (PMV): -0.01 to +0.07
- Predicted Percent Dissatisfied (PPD): 5.1% to 5.4%

RESULTS

The variation in the thermal conditions was minimal. For the assumed clothing and metabolism, a remarkably low PPD was measured. ASHRAE's accepted thermal comfort design guidelines permits PPD to be as high as 20%. Satisfying nearly 95% of the occupants is certainly far superior to the ASHRAE target of 80% satisfied. The DOAS-CRCP system offers extremely fine control of both the temperature and relative humidity of the space while maintaining good air circulation. Therefore, other levels of clothing and activity levels can be accommodated easily with the system to produce the same PPD.

CONCLUSION

Dedicated outdoor air systems, when properly designed and controlled, are capable of delivering very stable and comfortable environments (PPD = 5%). If the tests were performed in a space with a better envelope, the results could be expected to be even better. Even so, the authors have not experienced any difficulties making the control system perform as desired. Perhaps the keys to success are the proper control of the enthalpy wheel and control of the cooling equipment to ensure that the space latent load is completely handled by the ventilation air.

It has been demonstrated that good air motion is achieved (ADPI of 97%) with ventilation airflow alone (typically around 20% to 30% of that required for thermal control), so it is not necessary to move large quantities of air. 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.

Finally, with the ability of the DOAS to decouple the space latent control from the sensible control, space relative humidity levels are maintained at the desired design level. The literature is loaded with information concerning the enhanced environmental and indoor air quality benefits of controlling the relative humidity in occupied spaces.

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DISCUSSION

Kathy Hauck, Product Manager, Reznor, Memphis, Tenn.: The evolution of the dedicated outdoor air systems has brought about conflict regarding how the air should be treated. Most sensible manufacturing companies try to “extend” their designs to work as makeup air units; however, the latent capabilities are limited. How can we determine which group or technical committee is governing the standard?

Stanley A. Mumma: If the question is about the heat recovery requirement in ANSI/ASHRAE/IESNA Standard 90.1-2004, it does not specify a sensible or latent effectiveness, only a total effectiveness. In most climates a total energy recovery device is required to meet the total effectiveness requirement, but a sensible only recovery device is enough in some arid climates. However, in such climates, total energy recovery is a huge help in the winter to prevent occupied buildings from operating at single digit relative humidity. I only recommend the use of total energy recovery devices, also know as enthalpy wheels or passive desiccant wheels.

I believe the cognizant TC for 90.1 is 7.6 Energy Utilization, but that doesn't mean they are the experts on this specific question (TC 5.5, Air-to-Air Energy Recovery, is the expert in this area). It also does not mean that 7.6 or 5.5 “governs” the standard. SSPC 90.1 writes the standard, and its recommendations are approved by the ASHRAE Board of Directors and ANSI.