Achieving Dry Outside Air in an Energy-Efficient Manner

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ABSTRACT

The central thrust of this paper is to develop a fundamental engineering understanding of outdoor air (OA) preconditioning equipment that utilizes passive desiccant wheels, sensible heat exchangers, and deep cooling coils to reduce the first and operating costs of cooling, heating, and humidification hardware over competing approaches. The specific equipment analyzed in the paper has an application niche in buildings that employ a separate dedicated outdoor air system. The separate dedicated outdoor air system is designed to meet the ventilation requirements of ANSI/ASHRAE Standard 62-1999, Ventilation for Acceptable Indoor Air Quality *and also supply that air dry enough to remove all of the space latent loads efficiently. Application of this type of equipment allows the designer much greater equipment selection freedom with regard to meeting the remaining space sensible cooling/heating requirements. Emphasis will be placed on the physical and thermodynamic performance of each of the individual components of the system, as well as their operation together. Detailed analysis of the component and system performance is presented for all outdoor air thermodynamic conditions that fall into any of the four distinct operating regions of the psychrometric chart. Finally, a detailed analysis of the energy implications of utilizing this type of equipment will be presented and compared to various configurations and a conventional all-air VAV system.*

INTRODUCTION

The engineering community and building operators have been attempting for over ten years to meet the intent of the ventilation requirements of *ANSI/ASHRAE Standard 62-1999* (ASHRAE 1999) with all-air variable air volume (VAV) systems. Only recently has it finally become clear to the authors that it is neither energy, economically, or legally (Diamond 2000) beneficial to pursue this effort further. Many leaders in the industry are advocating, as a better alternative, the use of a separate outdoor air (OA) system (Coad 1999; Brady 1997; Meckler 1986; Scofield and Des Champs 1993; Mumma and Lee 1998) working in conjunction with a parallel (not series) mechanical system(s) to handle the building space thermal requirements. It is this migration to the expanding utilization of separate OA systems that has given rise to the growing use and prominence of the equipment analyzed in this paper. Unfortunately, many in the engineering community have not taken the time to carefully study this approach and develop a thorough understanding of its technical characteristics and economic benefits.

LITERATURE REVIEW

The trade literature has a growing number of articles presenting the basic concept of dedicated outdoor air and the use of total energy recovery; however, the archival literature is silent on the detailed thermodynamic performance of systems as presented in this paper. The trade literature is also flush with articles reporting moisture problems in public buildings, most notably schools or other places of high occupancy. The *ASHRAE Journal* (Harriman et al. 1997) has presented one article on the high sensible and latent loads associated with conditioning ventilation air. In summary, it must be concluded that the archival literature is silent on the engineering aspects of this type of system.

GENERAL CONFIGURATION OF THE DEDICATED OUTDOOR AIR SYSTEM

A general layout of the dedicated outdoor air system (DOAS), consisting of a preheat coil, an enthalpy wheel (EW),

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a deep cooling coil (CC), a sensible heat exchanger, and the prime movers, is illustrated in Figure 1. From this point on, the acronym DOAS will mean the set of equipment specified above, even though many other configurations of dedicated OA systems can be conceived. In the configuration illustrated, the sensible heat exchanger is a sensible heat wheel. For the sake of the discussion in this paper, this will be the assumed arrangement, although equally good heat exchangers in the form of plate type or heat pipe are used in the industry. As will be discussed later, the effectiveness of the sensible heat exchanger must be variable down to zero. Modulating its rotational speed alters the effectiveness of the sensible wheel. Heat pipe heat exchangers have some control of effectiveness by altering the unit tilt, but it is not possible to reduce it to zero in this manner. Plate type and heat pipe heat exchanger effectiveness must essentially be controlled by the flow rate of one of its air circuits. That means these latter two heat exchangers require face and bypass dampers to limit effectiveness, an alternative that will not be developed further in this paper. A discussion of the rational for each of the components in the system and their performance characteristics will be presented next.

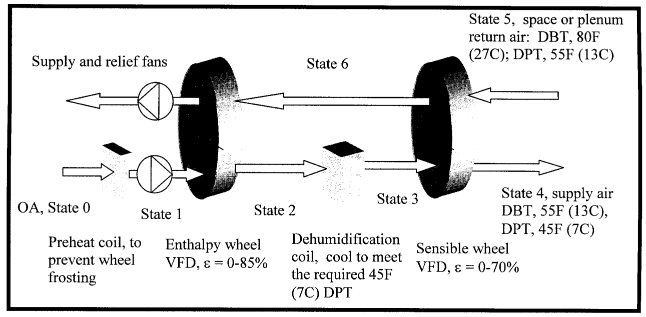

Figure 1 *General arrangement of the DOAS.*

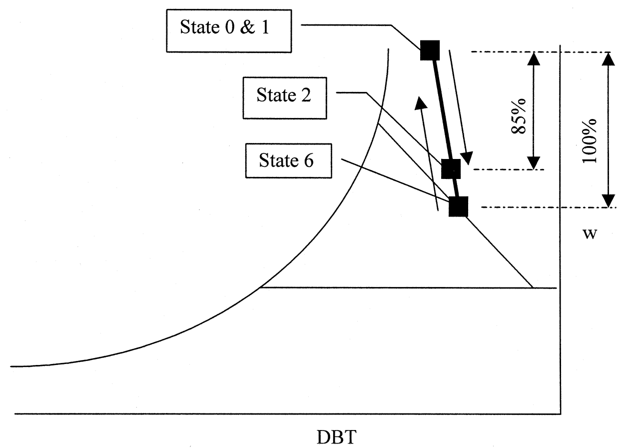

Figure 2 *Eighty-five percent effective enthalpy wheel process on the psychrometric chart.*

Enthalpy Wheel

The enthalpy wheel (also called a passive desiccant or total energy wheel) recovers both sensible (temperature) and latent (moisture) energy. The wheel's desiccant-loaded honeycomb rotor design (its appearance is like the edge of a cardboard box) provides for high heat transfer, with low pressure loss parameters. An example enthalpy wheel performance is plotted on the psychrometric chart in Figure 2. A straight line connecting the thermodynamic states of the two entering airstreams represents the process. The two entering airstreams, labeled in Figure 2, are thermodynamic state points 1 and 6 (from here on the thermodynamic state points will be referred to only as state). If the two airstream flow rates are equal and the enthalpy wheel effectiveness is 85%, as noted in Figure 2, then the outdoor airstream is taken from state 1 to state 2, which is within 15% of state 6 (i.e., dry-bulb temperature [DBT], humidity ratio [w], and enthalpy [h]). The benefits of this level of total energy recovery are very strong in the summer in terms of load and energy consumption reduction. It is equally beneficial for minimizing heating and humidification energy use during the cold winter months (a critical word of caution concerning the winter operating conditions). If the straight line joining states 1 and 6 crosses the saturation curve, condensation will occur in the wheel. If the temperature of state 1 is below freezing, then the condensate may freeze in the wheel. In order to avoid this unacceptable condition, preheat is required under certain conditions to prevent frosting of the enthalpy wheel. This situation will be illustrated later in the paper.

Preheat Coil

As noted above, a small preheat coil is required in many locations if the enthalpy wheel is to be used in the winter. With a proper preheat setpoint reset control schedule, only minor preheat energy will be required to avoid freezing, and further heating of the outdoor air is virtually eliminated with the enthalpy wheel.

Deep Cooling Coil

For the applications addressed in this paper, relatively low DPT (42-48ºF [6-9ºC]) air is required in order to remove the entire latent load from the space with the ventilation air and still maintain a target space dew point temperature (DPT) in the neighborhood of 50-55ºF (10-13ºC). During periods of outdoor air DPTs in excess of that required for supply, the CC must be controlled to maintain the low DPT. However, when the outdoor air DPTs are below that required by the supply air, the control setpoint may be reset up to the desired supply air temperature. In this case, the CC is no longer needed to perform dehumidification.

Sensible Wheel

If the space-sensible loads were always sufficiently high to permit the cold air leaving the CC to directly enter the space without local reheat, the sensible wheel would not be desirable or necessary. However, in many applications, the internal and envelope sensible cooling loads are not sufficiently high to prevent overcooling with the low temperature ventilation air. Therefore, it is desirable to elevate the supply air temperature. For the sake of this paper, it will be assumed that the supply air temperature is elevated to 55ºF (13ºC). This will be accomplished by the sensible wheel, although as mentioned earlier, other forms of sensible heat transfer equipment could be utilized. When two equal flow rate airstreams exchange energy in the sensible wheel, virtually no moisture is exchanged. For the sensible wheel illustrated in Figure 1, the 45ºF (7ºC) air leaving the deep CC is reheated sensibly to 55ºF (13ºC) with energy extracted from the return airstream. The return air is sensibly cooled by 10ºF (6ºC) in this process, thus lowering the energy content of the return airstream, reducing further the enthalpy of the outdoor air leaving the enthalpy wheel and entering the deep CC.

Supply and Return Fans

These fans must be selected to overcome the resistance to flow from the wheels, coils, and duct systems that they serve. And in some manufacturers' equipment they must also be able to handle the excess air in the purge cycles (when utilized to flush the return air from the wheel before it enters the clean supply air) of the enthalpy wheel. These fans would be required at all times the building is occupied.

PSYCHROMETRIC ANALYSIS OF THE DOAS SYSTEM UNDER ALL OUTDOOR AIR WEATHER CONDITIONS

The operation of the DOAS system (Figure 1) is best understood with the help of a psychrometric chart. The psychrometric chart illustrated in Figure 3 presents the four regions (A, B, C, and D) into which the OA may fall. Figure 3 and the discussion in this paper are based upon the following conditions:

- State 3, 45ºF (7ºC) and saturated
- State 4, 55ºF (13ºC) DBT and 45ºF (7ºC) DPT
- State 5, 80ºF (27ºC) DBT and 55ºF (13ºC) DPT
- State 6, 70ºF (21ºC) DBT and 55ºF (13ºC) DPT

It may be noted that there is a horizontal line representing 45ºF (7ºC) DPT, the supply air DPT. If the outside air conditions fall above that line, the air must be cooled and dehumidified to state 3 (45ºF [7ºC] or other DPTs as required to completely decouple the latent load) and then reheated to state 4 (55ºF [13ºC] in this example) with the sensible wheel. The sensible cooling of the relief air from state 5 to state 6 is a result of energy extraction from the return air. An identical rate of heat is added to the supply air leaving the deep CC at state 3, reheating it to state 4.

A line of constant enthalpy passing through state 6 separates the area above the 45ºF (7ºC) DPT line into regions A and B. The boundary between regions A and B (h_6) separates OA conditions, where dehumidification is required, into the two regions. In region A, full use of the enthalpy wheel dramatically reduces the CC load. In region B, any use of the enthalpy wheel increases the CC load; therefore, the enthalpy wheel must be off.

Another boundary is formed by the extension of a line through the return condition state 5 and the supply state 4. The line, which first appears at state 4 and proceeds to a humidity ratio of 0 gr/lbda (0 g/g), divides the area below the 45ºF (7ºC) DPT line into two regions, C and D. The boundary between regions C and D separates the OA conditions, where humidification is required, into two regions. In region C, sensible cooling is required. In region D, no sensible cooling is required. The operating status of the equipment in each of the four regions is presented in Table 1.

Note: Since no preheat is required in regions A, B, and C, state 0 (before the preheat coil) and state 1 (after the preheat coil) are thermodynamically the same. Therefore, in the discussion that follows, state 1 will be used to refer to the OA condition when in these regions.

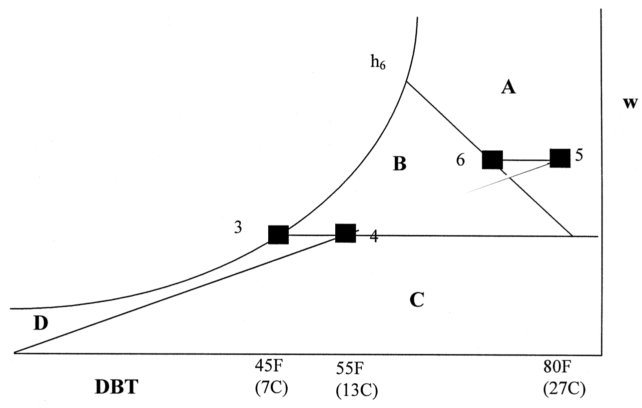

Figure 3 *Four regions on the psychrometric chart.*

TABLE 1
Control Status of the DOAS Equipment

Region	Enthalpy wheel CTL	Cooling coil CTL	Sensible wheel CTL
A	100% speed for max. effectiveness	Modulate to hold 45ºF (7ºC) LAT	Modulate to hold 55ºF (13ºC) SAT
B	Off! Must not modulate	Modulate to hold 45ºF (7ºC) LAT	Modulate to hold 55ºF (13ºC) SAT
C	Modulate to required DPT	Modulate to hold 55ºF (13ºC) LAT	Will modulate off
D	Modulate to required DPT	Will modulate off	Modulate to hold 55ºF (13ºC) SAT

Figure 4 will be used to illustrate how the DOAS system works when the OA conditions fall in either region A or B. When the outdoor conditions fall in region A, as illustrated by state 1, the enthalpy wheel operates at full effectiveness as it cools and dehumidifies the OA to state 2 without the expenditure of chiller energy. State 2 is on the straight line connecting states 1 and 6, as discussed earlier in the enthalpy wheel section. Further, since the effectiveness of the enthalpy wheel is assumed to be 85% in this illustration, state 2 is within 15% of state 6. The CC control valve modulates the coil capacity, cooling the air from state 2 to 45ºF (7ºC) at state 3. Without the enthalpy wheel, the CC would have cooled and dehumidified the much higher energy content air from state 1. Finally, the sensible wheel speed is modulated to reheat the 45ºF (7ºC) air at state 3 to the desired 55ºF (13ºC) state 4 condition without the expenditure of reheat energy.

When the outdoor conditions fall in region B, use of the enthalpy wheel operating between the OA state 1 and state 6 would increase the enthalpy of the air entering the CC. Therefore, the enthalpy wheel must be shut off when the OA conditions fall in region B. In this case where the OA is assumed to be in region B, state 1 and state 2 are equal. The CC control valve modulates the coil capacity, cooling the air from state 1 (state 2 is the same as state 1 in this case) to 45ºF (7ºC) at state 3. The sensible wheel speed is again modulated to reheat the 45ºF (7ºC) air to the desired 55ºF (13ºC) state 4 condition without the expenditure of heating energy.

Figure 5 will be used to illustrate how the DOAS system works when the OA conditions fall in region C. When the outdoor condition, state 1, is in region C, preheat is never required to prevent enthalpy wheel frosting. It must also be noted that in region C, there will never be a need for the use of the sensible wheel if the deep CC set point is reset to 55ºF (13ºC). Since, in region C the air is already below the required DPT, it is completely unnecessary to cool the air to 45ºF (7ºC) for dehumidification. The enthalpy wheel speed is modulated to bring the air up to the desired DPT for comfort (45ºF [7ºC] in this example). The resulting state 2 lies at the intersection of the enthalpy wheel process line between state 1 and state 5 (not state 6 since no reheat is required when the OA is in region C) and the 45ºF (7ºC) DPT line. In this example, the enthalpy wheel speed is reduced so that its effectiveness is reduced to about 50%. From state 2, the air is sensibly cooled by the deep CC to 55ºF (13ºC) and supplied to the building.

In region D, preheat is never needed if the OA temperature is above 32ºF (0ºC). Such a case is illustrated in Figure 6. In region D, both the enthalpy wheel and the sensible wheel speeds are modulated. The enthalpy wheel is used to bring the OA DPT up to 45ºF (7ºC), which will always result in a temperature at state 3, which is below the desired 55ºF (13ºC) at state 4. Therefore, the sensible wheel speed is modulated so the required energy is removed from the return air and added to the supply air to bring its temperature up to 55ºF (13ºC). In this illustration, the temperature difference between states 5 and 6 equals the temperature difference between states 3 and

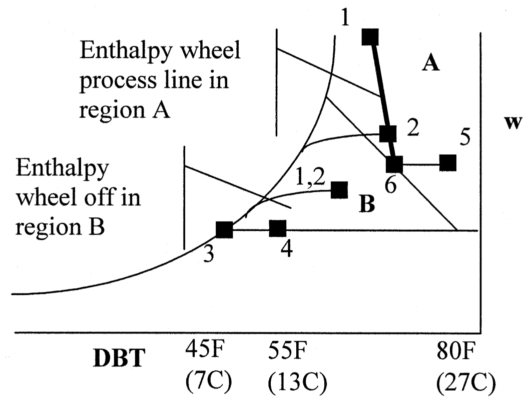

Figure 4 *Cooling and dehumidification processes in regions A and B.*

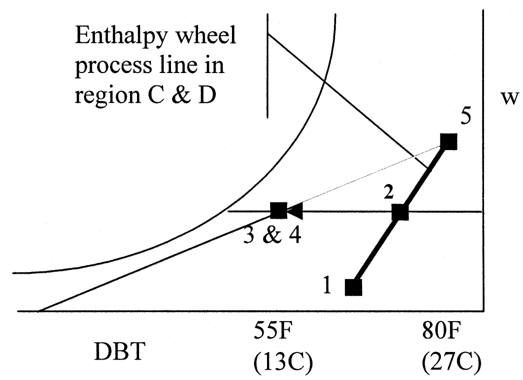

Figure 5 *Humidification and sensible cooling processes in region C.*

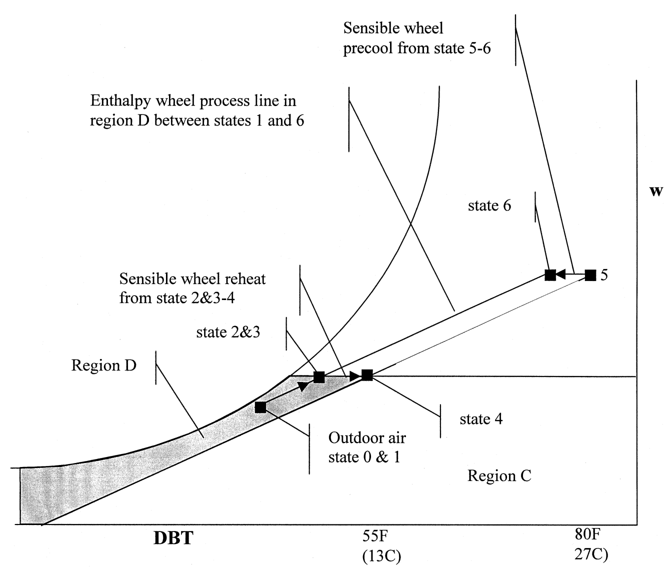

Figure 6 *Humidification and sensible heating processes in region D, no preheat needed.*

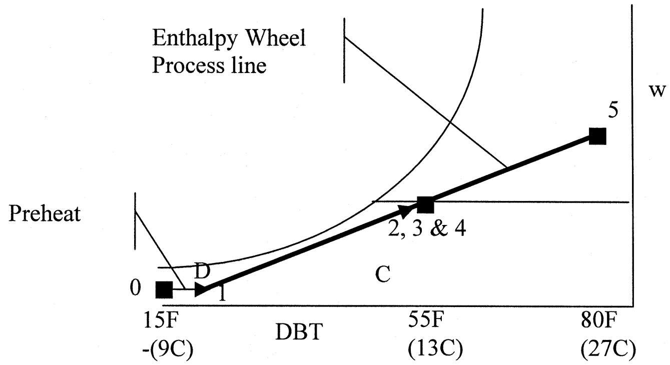

Figure 7 *Preheat and enthalpy wheel heating and humidification in region D.*

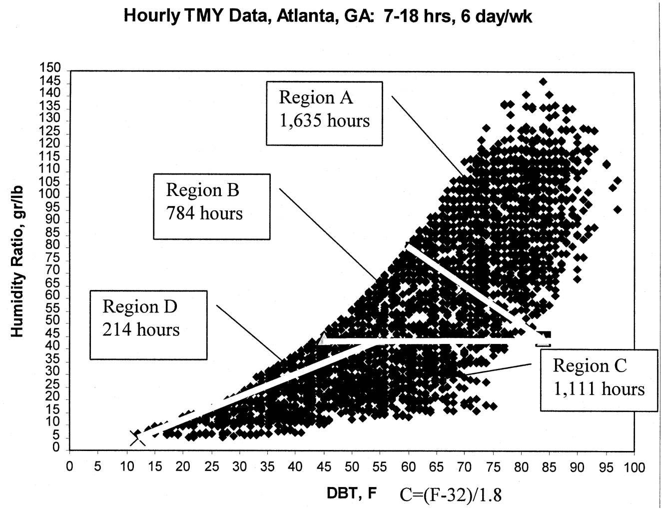

Figure 8 *Atlanta weather data on the psychrometric chart, 3744 hours.*

4 (note this is less than 10ºF [6ºC]). Finally, when the OA DBT is sufficiently low and in region D, preheat is required. This is illustrated in Figure 7. This air is preheated by the preheat coil until it reaches the line separating regions C and D at state 1. Air at state 1 is then heated and humidified by the enthalpy wheel to the desired supply air state 4. In this case, the sensible wheel is off. The deep CC is always off when the OA falls anywhere in region D.

ANNUAL HOURLY ENERGY SIMULATION RESULTS BASED UPON ATLANTA, GEORGIA, TMY WEATHER DATA

In this section, peak and annual cooling/heating/humidification duty for five configurations of separate dedicated OA systems (including the DOAS system discussed in depth in this paper) are developed. The peak and annual duty of mechanical equipment used to condition OA in a conventional all-air VAV system is also developed and presented. The intent is to present the DOAS system performance alongside the other approaches. To facilitate the analysis, the TMY weather data for Atlanta are used.

The TMY Weather Data

A tool that makes analysis of the type presented here readily accessible to the practicing engineer is a weather summary tool (GRI 1998). The analysis that follows is based upon hourly data for Atlanta, Ga., taken from this tool, which makes extraction of the desired hourly data very simple, and for this analysis, hourly data are extracted for a six-day week excluding Sundays, with 12-hour days starting at 7 a.m. and ending at 7 p.m. The specific data utilized are the DBT, w, and h. A psychrometric plot of the 3744 occupied hours of data for Atlanta is presented in Figure 8.

Configuration 1. A conventional cooling/heating/humidification arrangement without the use of heat recovery (Figure 9) will be discussed first. To determine both the peak hourly and annual energy situation for this configuration, the Atlanta TMY data are separated into three regions. The first, the humid region, consists of all data with a DPT above 45ºF (7ºC), where cooling and dehumidification are required, followed by reheat. In this region, the hourly CC load is simply the product of the OA mass flow rate (all examples were performed with 10,000 scfm [4700 L/s] of OA) times the enthalpy difference between the OA at that hour and the supply enthalpy (a constant 45ºF [7ºC] and saturated). The reheat energy required for each hour in this region is the sensible heat required to raise the 10,000 scfm (4700 L/s) 10ºF (6ºC) or from 45ºF (7ºC) to 55ºF (13ºC). A second region included all of the OA data with DPTs below 45ºF (7ºC) and warmer than 55ºF (13ºC). This region needed sensible cooling and humidification. In this region, the hourly CC load (sensible only, to 55ºF [13ºC]) is simply the product of mass flow rate, specific heat of air, and temperature difference between 55ºF (13ºC) and the OA DBT for the hour. The mass flow rate of moisture added to the OA stream for each hour is the product of the OA mass flow rate times the humidity ratio difference between the required supply air (45ºF [7ºC] DPT) and the OA. Each pound mass of water used for humidification consumed 1000 Btu/lb of energy, the latent heat of vaporization of water. The final region is the remaining region of the psychrometric chart. That region required both heating and humidification.

Configuration 2. A conventional cooling/heating/humidification arrangement with run around heat recovery will now be discussed (Figure 9). It is assumed that a constant 10ºF (6ºC) reheat (45-55ºF [7-13ºC]) is desired when the OA DPT is above 45ºF (7ºC). With widely varying OA temperatures, the reheat would be much greater than 10ºF (6ºC) at any time the OA is above about 62ºF (17ºC) if the system ran wild, as many manufacturers' systems operate. Such operation reduces the CC load in the OA system because of the increased

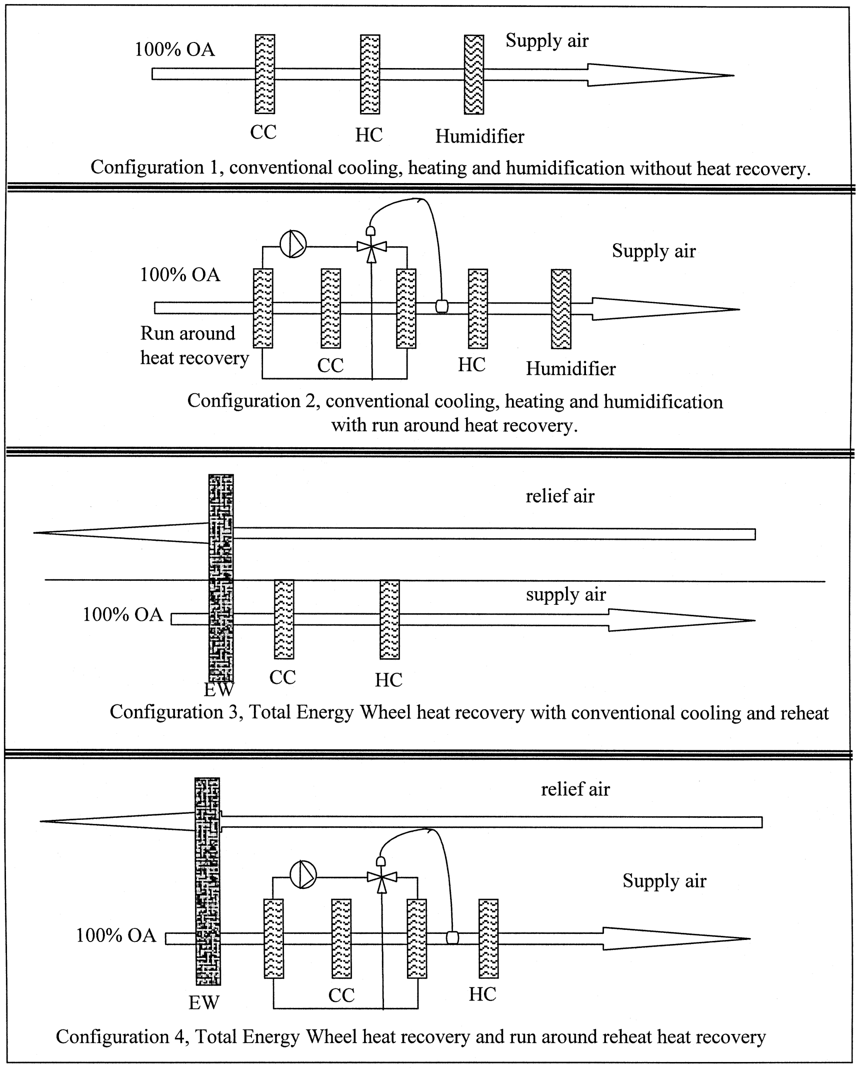

Figure 9 *Component arrangement for systems 1, 2, 3, and 4.*

precooling when compared to a constant 10ºF (6ºC) reheat. The wild reheat coil operation simply shifts more of the sensible load to the parallel system. Therefore, a larger parallel system is foolishly required. To avoid this, the flow rate of the run around heat transfer fluid through the reheat coil must have modulation capability, as illustrated in Figure 9. The configuration has one additional region, a part of the humid region of system 1. When the moist OA temperature drops below 62ºF (17ºC) (assuming a run around system effectiveness of 0.6), supplemental reheat is necessary. In the humid region, the large use of reheat energy exhibited by system 1 is greatly reduced. The precooling in this region also reduces the CC load. Performance when the OA condition falls in the dry region of the psychrometric chart is identical to that of configuration 1.

Configuration 3. A total energy wheel is added to the conventional configuration 1 system (Figure 9). This system operates much like the DOAS system but without the free reheat and precooling boost of the sensible heat recovery unit. The enthalpy wheel eliminates the need for winter humidification and most heating energy use. In the portion of the psychrometric chart with DPTs below 45ºF (7ºC), the system behaves exactly like the DOAS system in region C. A region D does exist for this configuration just like the DOAS equipment. When the enthalpy wheel is used to bring the OA DPT up to 45ºF (7ºC), the DBT will be less than 55ºF (13ºC). Consequently, since there is no way to recover heat from the relief airstream, heating energy must be supplied to elevate the supply air temperature to 55ºF (13ºC). In the wet region with DPTs above 45ºF (7ºC), two areas exist separated by the enthalpy of the return air. For OA enthalpies above the return condition, the enthalpy wheel operates at full effectiveness like the DOAS in region A. For OA enthalpies below the return enthalpy, the unit operates like the DOAS in region B where the enthalpy wheel is off.

Configuration 4. This configuration combines the heat recovery features applied in configurations 2 and 3. Analysis is based upon the same approach used in configurations 2 and 3. It requires reheat energy for OA DBTs below 62ºF (17ºC) and above 45ºF (7ºC) DBT, as is the case with configuration 2. It will also need heating when in region D, just as configuration 3 required. Space does not permit a detailed discussion of the psychrometrics as is done for the DOAS equipment. However, it should be noted that with the configuration 4 arrangement, region A is smaller than with the DOAS unit, and region B is larger. In addition, in region B, the run around coils help reduce the CC load, when no such help is available with the DOAS equipment. As in the other configurations, the weather data is sorted into regions, and the appropriate analysis is performed.

Configuration 5. This is the DOAS unit discussed in this paper consisting of the enthalpy wheel, CC, and sensible heat exchanger (Figure 1). As in each of the configurations discussed above, the TMY weather data is sorted into the appropriate regions for analysis. In this case, the data are sorted into the four psychrometric regions A, B, C, and D. During the occupied hours in Atlanta, there were 1635, 784, 1111, and 214 hours of OA occurrences in regions A, B, C, and D, respectively, as noted in Figure 8. Clearly, the frequency of occurrences in each of the four regions (or geographic location) has a bearing on the energy demand and consumption of dedicated OA systems.

Configuration 6. A conventional all-air VAV system without heat recovery is analyzed last. The first step of the analysis is to determine the total load imposed each hour on the system CC necessary to bring the OA to the return air enthalpy (the real OA load). This is a simple hourly product of the mass flow rate of OA and the enthalpy difference between the return air and the OA. It must be noted that this number is negative whenever the OA enthalpy is lower than the return air enthalpy, indicating that the OA is contributing to a reduction in CC load. In this analysis, it is assumed that no airside economizer is operating since they offer essentially no energy savings in Atlanta. This is consistent with the paper by Coad (1999) and is confirmed (using commercially available load and energy analysis software) for a 60,000 ft^2 office building located in Atlanta. It is also assumed that the OA to supply air ratio is sufficiently small so that preheating is not necessary in Atlanta (with a 12ºF [–11ºC] minimum OA temperature, this ratio may not exceed approximately 1/3 for this assumption to be true). To provide a fair comparison between the VAV systems and the dedicated OA systems, a space sensible and latent load (comparable to that handled by 10,000 scfm (4700 L/s) of OA at the required supply air conditions) is added to the requirements for the VAV system. The load is equal to the space sensible and latent cooling performed by each of the first five system configurations that supply air at 55ºF (13ºC) DBT and 45ºF (7ºC) DPT.

RESULTS

The peak load on the CC for each of the configurations is presented in Table 2. The peak CC load with the DOAS system is by far the lowest at 43 tons (150 kW). The high CC load for the conventional VAV system reflects a 51-ton load to cool and dehumidify the OA to the return condition, plus the 34-ton terminal cooling accomplished by the 55ºF (13ºC) DBT and 45ºF (7ºC) DPT supply air condition of the first four systems. The run around heat exchanger in system 2 only reduces the peak CC load by about 9 tons (32 kW). System 3, with the enthalpy wheel, realized a big drop in peak coil load compared to system 1, but not as much as the DOAS. The peak humidification energy use rate is also presented in Table 2. Systems without an enthalpy wheel and, hence, unable to

TABLE 2
Comparison of the Equipment Size Required for the Six Different Configurations Treating 10,000 scfm (4700 L/s) of OA

Configuration Number	Configuration approaches	Peak OA load on the CC, tons (kW)	Peak humidification energy use rate, k·Btu/h (kW)	Peak heating energy use rate, k·Btu/h (kW)
1	Conv. cooling/heating/humidification	94 (330)	238 (70)	465 (140)
2	Config. 1 with a run around loop at the cooling coil	85 (300)	238 (70)	465 (140)
3	Enthalpy wheel and conv. cooling/ heating	51 (180)	0	108 (30)
4	Enthalpy wheel and a run around loop at the cooling coil	43 (150)	0	108 (30)
5	DOAS	43 (150)	0	0
6	Conv. VAV	85 (300)	238 (70)	0

recover moisture from the relief airstream, systems 1, 2, and 6, require 238 k·Btu/h. The peak heating energy use rate for each of the systems is presented in Table 2. Since the peak heating requirement occurs at the lowest OA temperatures, when the run around heat recovery equipment in system 2 is dormant, the peak heating for the first two systems is an identical 465 k·Btu/h. The peak heating for systems 3 and 4 is reheat energy that occurs below 62ºF (17ºC) during the humid OA conditions. Heating is never required for configurations 5 and 6.

Table 3 presents a comparative annual energy picture for the six configurations. For configuration 1, the energy use is high for cooling, heating, and humidification. The run around heat recovery system in configuration 2 reduces the CC ton hours some and the reheat kBtu substantially. However, the high heating and humidification figures remain. The use of an enthalpy wheel with conventional heating and cooling (configuration 3) results in a large drop in the heating and humidification energy consumption. And the enthalpy wheel reduces the CC TH compared to configuration 1 some. However, it provides no relief from the high reheat energy consumption. Configuration 4, by combining the benefits of configurations 2 and 3, is able to substantially reduce the CC TH (about 35% less than configuration 1) and nearly eliminates the reheat and heating energy use. Humidification energy went to zero. Clearly, this is an energy-efficient OA conditioning configuration. It is important to note that most manufactured equipment cannot be controlled as this one is modeled, at least in part because the run around coil is allowed to run wild. The DOAS unit uses about 7% more cooling energy but less reheat and heating energy than configuration 4. On a comparable annual Btu basis, they are about equal. Proper control of the DOAS unit is an easier proposition (Mumma 2001) and, therefore, perhaps a slightly better choice. The conventional VAV system is not a bad energy choice for equal OA flow rates (rarely would they be equal since application of the multiple spaces equation of Standard 62 yields a corrected OA flow rate often 30% larger than needed with the DOAS unit). It requires only about 11% more CC energy than the most efficient system but still requires substantial humidification energy (even in Atlanta if the space RH is maintained no lower than 40%). Both of these issues could be reduced with an enthalpy wheel in the VAV system. Clearly, energy use and demand are not the major factors moving the engineering community toward dedicated OA systems. On the other hand, they are not holding it back.

CONCLUSIONS AND RECOMMENDATIONS

The thermodynamic processes for the DOAS equipment, as depicted in Figure 1, has been fully analyzed. The peak and annual operating energy characteristics of the DOAS unit have been compared with five other approaches and found to be among the leaders. Clearly, the DOAS approach can be accomplished without an energy penalty compared to the other approaches presented in the paper. And the size of the chiller necessary to condition the OA is only about half that required for an all-air VAV system. However, energy savings alone will not drive the migration to dedicated OA units. Rather, the superior comfort and productivity that accompany systems that can separate the sensible and latent loads are important. Another major reason is the ability to ensure the proper ventilation air to every zone in the building, something that cannot be ensured with a single all-air system. In conclusion, a DOAS system as discussed in this paper is highly recommended.

TABLE 3
Annual Energy Utilization When 10,000 scfm (4700 L/s) of Air Is Used with the Six Configurations

Configuration Number	Configuration approaches	Cooling coil, TH (kWh)	Reheat k·Btu (kWh)	Heating k·Btu (kWh)	Humidification k·Btu (kWh)
1	Conv. cooling/heating/humidification	113,000 (400,000)	255,000 (75,000)	110,000 (32,000)	155,000 (45,000)
2	Config. 1 with a run around loop at the cooling coil	103,000 (360,000)	14,800 (4000)	110,000 (32,000)	155,000 (45,000)
3	Enthalpy wheel and conv. cooling/ heating	103,000 (360,000	255,000 (75,000)	7000 (2000)	0
4	Enthalpy wheel and a run around loop at the cooling coil	83,000 (290,000)	14,800 (4000)	7000 (2000)	0
5	DOAS	89,000 (310,000)	0	0	0
6	Conv. VAV	92,000 (320,000)	0	0	155,000 (45,000)

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