30% Surplus OA

Does It Use More Energy?

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

The Building Sciences column "Why Green Can Be Wash" in the November 2008 ASHRAE Journal addressed overventilation this way, "Do you want to save serious energy and serious money? ... Then don't overventilate. This idea of getting green points by increasing the rates above those specified by Standard 62 is just madness."[1] The central thrust of this article is to explore the veracity of this statement as it relates to dedicated outdoor air systems (DOAS). First, the 2009 U.S. Green Building Council's LEED®[2] for New Construction and Major Renovations is briefly reviewed.

LEED Rating System

LEED is a voluntary rating system, not a formal standard. The system consists of the following sections with percentages of the maximum 110 credits:

- Sustainable sites, 24%;
- Water efficiency, 9%;
- Energy and atmosphere, 32%;
- Materials and resources, 13%;
- Indoor environmental quality (IEQ), 14%;
- Innovation in design, 5%;
- Regional priority, 3%.

The increased ventilation LEED credit is not in the energy and atmosphere category; it is in the IEQ category. One might imply, from the Journal article referenced previously, that LEED was 100% about energy. The intent of this LEED credit is "to improve indoor air quality for improved occupant comfort, well-being and productivity." The improvement is accomplished by "increasing breathing zone outdoor air ventilation rates by at least 30% above the minimum rates required by Standard 62.1-2007."

DOAS

DOAS, as defined by ASHRAE,[3] uses a separate unit to condition (heat, cool, humidify, dehumidify) all of the outdoor air brought into a building for ventilation and then deliver it directly to each occupied space or to local HVAC units serving those spaces. Meanwhile, the local units (fan coils, water-source heat

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Occupancy Category	cfm/Person	cfm/ft²	People/1,000 ft²	Combined Outdoor Air cfm/Person	Per Person Latent Load, Btu/h	Space Enthalpy at 75°F 50% RH	Required Supply Air Enthalpy, Btu/lb$_m$	Supply Air Dew-Point Temperature,°F	30% More Combined Outdoor Air/Person	Required Supply Air Enthalpy, Btu/lb$_m$ With 30% Surplus Air	Supply Air Dew-Point Temperature,°F With 30% Surplus Air
Lecture Classroom	7.5	0.06	65	8.42	200	28.14	22.86	35.96	10.96	24.08	41.63
Classroom >9 Years Old	10.0	0.06	35	11.71	200	28.14	24.36	42.75	15.23	25.22	46.08
Conference Room	5.0	0.06	50	6.20	200	28.14	20.97	24.84	8.06	22.63	34.75
Office	5.0	0.06	5.0	17.0	200	28.14	25.53	47.18	22.10	26.13	49.20
Museums	7.5	0.06	40	9.0	250	28.14	21.97	31.05	11.70	23.39	38.56

Note: Columns 1 through 4 are taken from ASHRAE Standard 62.1-2007, Table 6-1.

Table 1: Ventilation data for different categories of occupancies.

pumps, small packaged units, variable-air-volume terminals, chilled ceiling panels, chilled beams, etc.) in or near each space maintain space temperature. Treating the outdoor air separately from recirculated air can make it easy to verify that sufficient ventilation airflow reaches each occupied space and can help limit indoor humidity levels. The latter is accomplished by dehumidifying the ventilation air to remove all of the outdoor air and most, or all, of the space latent cooling load, leaving the local HVAC units to primarily handle space sensible cooling loads. (Some types of local HVAC equipment must operate dry to avoid problems associated with condensation, limiting their duty to sensible loads only.)

DOAS Issues Related to 90.1

- Air- or water-side economizers are in the prescriptive requirement for many locations. An exception is that economizers are not required for most locations east of 97°W longitude and south of 40°N latitude, i.e., Atlanta and New Orleans. An all air system with air-side economizer is generally able to meet the entire space cooling loads when the outdoor air temperature drops below 55°F (13°C). On the other hand, with the DOAS flow rate as low as 20% that of an all air system, the outdoor air temperature may need to fall below 0°F (–18°C) for it to meet the full cooling load.[4,5]
- Total energy recovery equipment (TER) with an energy recovery effectiveness (sensible and latent) of at least 50% shall be used when individual fan systems with capacities of 5,000 cfm (2360 L/s) or greater, consisting of at least 70% outdoor air, subject to exceptions.

DOAS 30% Surplus Ventilation Air Energy Hypotheses

The first hypothesis is that increasing the ventilation airflow rate will increase the energy required to cool and dehumidify, as well as temper the outdoor air (OA), but only about 20% as much as would occur if TER equipment were not used.

The second hypothesis is that increasing the ventilation airflow rate will result in a reduction in the winter cooling plant operation, saving operating costs. Consequently, any surplus air that can be brought in during the free cooling periods should prove beneficial in most locations where buildings are served by a DOAS.

The third hypothesis is that the extra free winter cooling will more than offset the increased cooling energy use during the summer months.

The analysis to follow compares, on an annual basis, the first and operating cost for DOASs with and without TER in four different geographic locations.

Exploring Typical Flow Rates Specified by 62.1

Two important issues to be explored are the required supply air (SA) dew-point temperature (DPT) and the steady-state space carbon dioxide (CO_2) concentration (a measure of contaminant dilution) as a function of combined OA per person. For that purpose, a limited number of occupancy categories (OC) found in Standard 62.1, Table 6.1[6] was selected. For those selections, the minimum combined (occupant and floor components considered) OA flow rate per person range from 6 scfm (3 L/s) per person for conference rooms to 17 scfm (8 L/s) per person for offices.

Information for the various OCs using minimal and 30% surplus ventilation air is presented in *Table 1*. The computed (solving the equation: Q_L=0.68×scfm×ΔW) SA DPT ranges from 24.8°F to 47°F (–4°C to 8°C) using the minimum OA per person. When the SA flow rate is increased by 30%, the SA DPTs are elevated, to some extent, to the range of 35°F to 49°F (2°C to 9°C).

SA DPTs below about 44°F (7°C) are difficult to obtain with mechanical refrigeration alone, and generally require the addition of active desiccants or a Type III passive dehumidification component.[7]

A word of caution, even with a DOAS designed to meet the entire space latent load, condensation on the terminal cooling surfaces will occur if they operate below the design space DPT.[8,9]

The data presented in *Figure 1,* based upon *Table 1* and indicated by diamonds, are for the minimum OA flow conditions. Those indicated by the squares are for 30% surplus OA. Points A and A′ are for the conference room OC, where the minimum combined OA flow rate is 6.2 cfm (2.9 L/s) per person and requires a DPT of 24.8°F (–4°C) for the DOAS to entirely meet the space latent load. A 30% increase in flow, a 2 cfm (0.9 L/s) per person increase, yields a 40% favorable increase in the required SA DPT to 34.8°F (2°C).

As the flow rate per person increases, a 30% increase in flow represents a progressively larger absolute change in cfm. At the same time, the magnitude of the

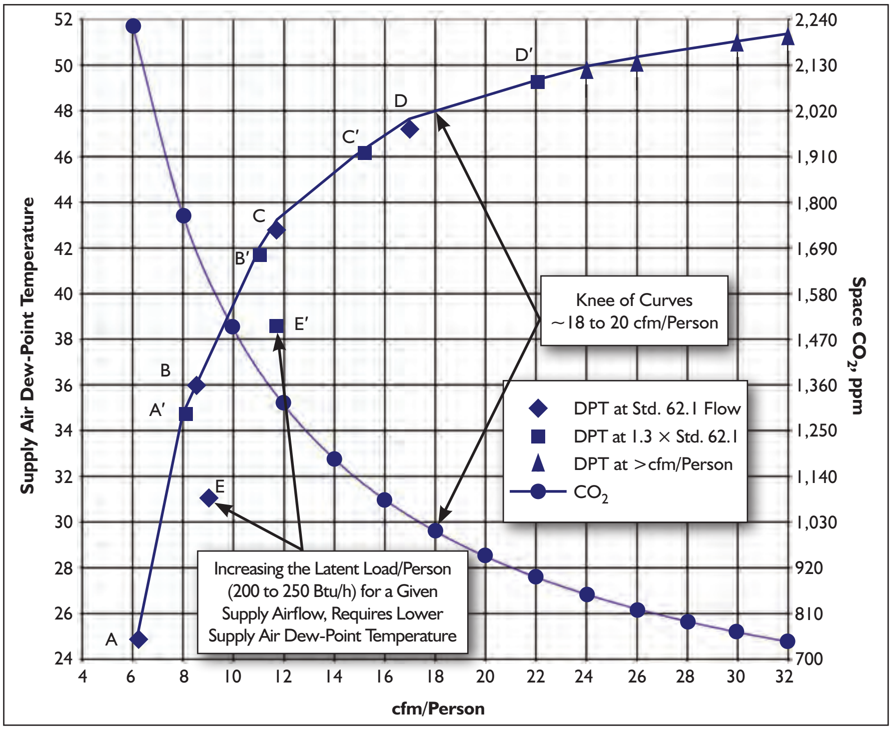

***Figure 1:** Required supply air dew-point temperature to handle the entire occupant latent load versus outdoor air/person (left axis); and space CO_2 ppm versus outdoor air/person (right axis).*

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SA DPT change decreases asymptotically. Specifically, the increase in the SA DPT from B to B′ is 16%, from C to C′ is 8%, and from D to D′ is 4%.

This relationship leads to *a key observation* that with a latent load of 200 Btu/h (59 kW) per person, increasing the airflow per person beyond about 18 to 20 cfm (8 to 9 L/s) per person, noted as the knee of the curve in *Figure 1*, has diminishing return when viewed from the perspective of elevating the SA DPTs to meet the latent loads. The triangle points on *Figure 1* are for further increased flow per person, intended to illustrate the trend beyond point D′. Finally, the same trend is visible between E and E′, but those points do not lie on the same curve since they are working with a greater latent load per person of 250 Btu/h (73 W).

Also plotted in *Figure 1* is the space steady-state CO_2 concentration (assuming an OA CO_2 concentration of 400 ppm and an occupant CO_2 generation rate of 0.01 cfm [0.31 L/min.]) as a function of cfm/person. It has a similar but opposite characteristic to the required DPT curve, with its knee also in the 18 to 20 cfm (8 to 9 L/s) per person. It would appear that increases in the OA flow per person beyond the knee would produce diminishing IEQ return. The IEQ literature reports a similar finding.[10]

Testing the 30% surplus ventilation air energy hypotheses by comparing the energy performance of a 4,600 scfm (2171 L/s) DOAS with a 6,000 scfm (2832 L/s)(4,600×1.3) DOAS. To undertake the test, a number of assumptions are required. They include:

- Space conditions maintained at 75°F (24°C) and 50% relative humidity year-round;
- All the cooling used to condition the OA in the summer is always needed with either flow (i.e., reheat never needed or used);
- Winter free cooling can be fully used by the building; or
- Winter free cooling, when reduced by the use of TER for the entire nonsummer periods, can be fully used by the building;
- Building is occupied 13 hours/day, 6 days per week, 52 weeks a year—4,056 hours/year;
- TMY weather data[11] used, and binned into 5 gr/lb_m (0.72 g/kg) bins for analysis;
- Locations are limited to: Atlanta; New Orleans; Columbus, Ohio; and International Falls, Minn.;
- Cooling plant energy characteristic, 0.7 kW/ton (2.46 kW/kW);
- Cost of electricity: $0.10/kWh;
- Use TER with an effectiveness of 0.8 on the OA side;
- DOAS combined fan/motor efficiency of 70%;

1		2	3	4	5	6	7	8
Flow Rate, cfm		Ton Hour (TH) Cooling Required Without TER	TH Cooling Required With 80% Effective TER	Operating Cost for Cooling Without TER	Operating Cost for Cooling With 80% Effective TER	Hours Where No Free Cooling Occurs	Hours Where Some Free Cooling Occurs	Lowest Temperature Leaving TER During Coldest Day, °F
Atlanta, Simulation Data								
No Free Cooling	4,600	14,826	2,965	$1,038	**$208**	1,561		
	6,000	19,330	3,866	$1,353	**$271**	1,561		
Some Free Cooling	4,600	–30,184	–7,502	**–$2,113**	–$525		2,495	
	6,000	–39,353	–9,781	**–$2,755**	–$685		2,495	65
New Orleans, Simulation Data								
No Free Cooling	4,600	31,490	6,298	$2,204	**$441**	2,292		
	6,000	41,000	8,211	$2,875	**$575**	2,292		
Some Free Cooling	4,600	–17,119	–4,031	**–$1,198**	–$282		1,764	
	6,000	–22,320	–5,256	**–$1,562**	–$368		1,764	67
Columbus, Simulation Data								
No Free Cooling	4,600	7,506	1,500	$525	**$105**	1,092		
	6,000	9,786	1,957	$685	**$137**	1,092		
Some Free Cooling	4,600	–47,084	–11,814	**–$3,296**	–$827		2,964	
	6,000	–61,387	–15,402	**–$4,297**	–$1,078		2,964	61
International Falls, Simulation Data								
No Free Cooling	4,600	1,934	387	$135	**$27**	308		
	6,000	2,521	504	$176	**$35**	308		
Some Free Cooling	4,600	–75,795	–19,210	**–$5,303**	–$1,345		3,748	
	6,000	–98,774	–25,045	**–$6,914**	–$1,753		3,748	59

Note: Negative signs indicate the free cooling delivered and savings.

Table 2: DOAS simulation results for Atlanta, New Orleans, Columbus, and International Falls.

- For systems with TER, the total pressure drop across the fan (DOAS internal and external) each direction, 3 in. w.g. (747 Pa), i.e., the flow of 4,600 or 6,000 cfm (2171 or 2832 L/s) are operating against 6 in. w.g. (1494 Pa). It is assumed that the system was designed for one or the other flow, and in each case the pressure drop is the same, i.e., increasing the flow to 6,000 cfm (2832 L/s)from a system designed for 4,600 cfm (2171 L/s) is not the case. It was assumed that the TER had bypass dampers, so when not in use, the fan power was assumed to be working against half the pressure drop.
- For systems without a TER, the pressure drop across the fan was assumed to be 3 in. w.g. (747 Pa);
- A design SA DPT of 46°F to 48°F (8°C to 9°C) was assumed to ensure that the entire space latent loads were accommodated;
- OA load is computed to be the product of mass flow rate times the difference between the *OA enthalpy and the design room enthalpy*;
- When the OA load goes negative using the enthalpy differences, some free cooling is occurring;
- Once the OA humidity ratio drops below the design supply air humidity ratio, the free cooling is computed as the product of the mass flow rate times specific heat times the difference between the OA dry-bulb temperature (DBT) and the design space DBT (75°F [24°C]) plus the design latent load of the space. Using mass flow times enthalpy difference results in an extreme overestimation of the free cooling because of the huge latent cooling potential that the dry OA has. But it can do no more than the design latent load—all that occurs is that the space RH drops below the design 50%. No supplemental humidification was assumed.
- For low OA temperatures, the DBT leaving the TER was computed for each location based on the 80% effectiveness and an exhaust air temperature of 75°F (24°C). In some locations, it may be necessary to provide some tempering beyond that of the TER. This issue will be addressed later.

The results of the cooling computations are presented in *Table 2*. Column 1 lists the OA flow rates for the periods without free cooling and with some free cooling for the four

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locations. Columns 2 and 3 present a summation of the ton hours (TH) for the periods without free cooling and with some free cooling. Columns 4 and 5 present the annual cooling operating costs or savings with and without TER. Columns 6 and 7 present the hours of no free cooling and hours where some free cooling is occurring. Column 8 lists the lowest temperature leaving the TER on the coldest bin for the location (59°F to 65°F [15°C to 18°C]).

This analysis would apply to *any* building type with space OA flows per person that have been adjusted to be within the range of 16 to 20 cfm (8 L/s to 9 L/s) per person (within the 30% OA range), not just office buildings. As noted in *Figure 1*, the required SA DPT varies by only a few degrees for OA flows of 16 to 20 cfm (8 L/s to 9 L/s) per person.

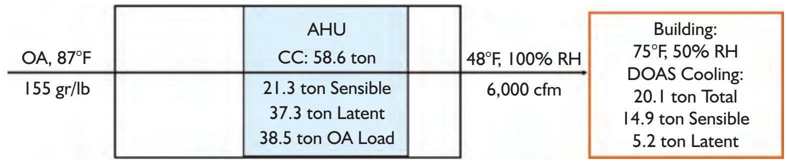
Energy Implications of 30% Surplus

In an effort to understand the annual OA cooling operating costs, and, therefore, energy implications of the extra 30% ventilation air, the differences in annual summer cooling costs displayed in Column 5 of *Table 2*, in bold, will be used. Also the savings from the added free cooling with the TER off, Column 4, in bold, will be used.

For each of the four locations, the results are as follows:

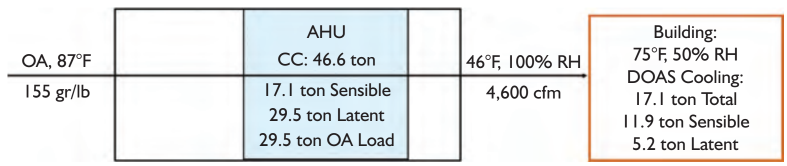
- Atlanta: ($271 – $208) – ($2,755 – $2,113) = –$579, a savings by adding 30% to the ventilation;
- New Orleans: ($575 – $441) – ($1,562 – $1,198) = –$230, a small savings by adding 30% to the ventilation;
- Columbus: ($137 – $105) – ($4,297 –$3,296) = –$969, a nice savings by adding 30% to the ventilation; and
- International Falls: ($35 – $27) – ($6,914 – $5,303) = –$1,603, an even larger savings by adding 30% to the ventilation.

One may argue that the savings for a climate such as International Falls is not accurate since tempering of the OA may be required in the winter to avoid overcooling, but was not included in the analysis. Note, however, that tempering is available "for free" from the TER, which would make supply air available, based on the data in Column 8 of *Table 2*, at 59°F (15°C) in the coldest OA weather bin. Warmer bins would yield higher leaving temperatures. *Caution*: tempering with the TER should only be done when necessary to limit the free cooling. Winter operation of the TER, with adequate frost protection, also recovers humidity from the exhaust airstream, either eliminating the need for other



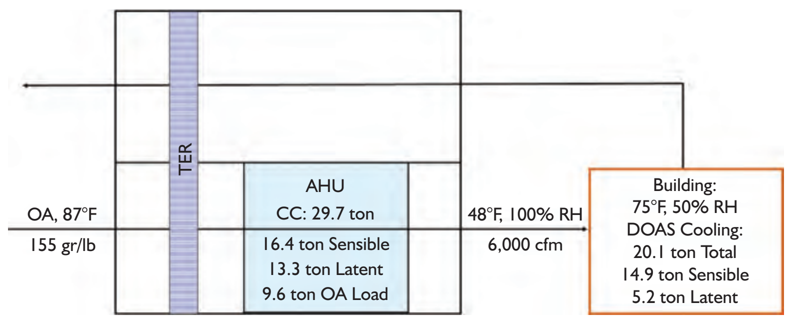
- AHU first cost: $9,900 + $9,000 installation.
- Air cooled chiller first cost: $24,900 + $5,000 installation.
- Total installed cost: $48,800.

***Figure 2:** New Orleans 6,000 cfm AHU with cooling coil, no TER.*



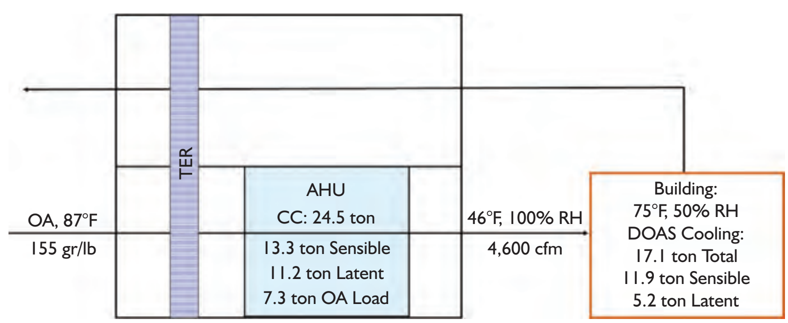
- AHU first cost: $8,100 + $6,900 installation.
- Air-cooled chiller first cost: $21,840 + $5,000 installation.
- Added FCUs to cover 3 ton of lost DOAS space sensible cooling.
  First cost: $1,440 + ($0 to $4,300 [three at $1,430 each]) installation.
- Total installed cost: $43,280 to $47,580.

***Figure 3:** New Orleans 4,600 cfm AHU with cooling coil, no TER.*



- AHU first cost: $19,800 + $12,000 installation.
- Air-cooled chiller first cost: $12,600+ $5,000 installation.
- Total installed cost: $49,400.

***Figure 4:** New Orleans 6,000 cfm AHU with cooling coil, with TER.*



- AHU first cost: $17,000 + $9,200 installation.
- Air-cooled chiller first cost: $12,110 + $5,000 installation.
- Added FCUs to cover 3 ton of lost DOAS space sensible cooling.
  First cost: $1,440 + ($0 to $4,300 [three at $1,430 each]) installation.
- Total installed cost: $44,750 to $49,050.

***Figure 5:** New Orleans 4,600 cfm AHU with cooling coil, with TER.*

humidification equipment and associated energy costs, or significantly reducing both.

If the free cooling is required, and low supply air temperatures are not acceptable, there are other ways of "free tempering" the air without losing the free cooling aspect of the OA. If the terminal equipment involves FCUs, blending the space air with the OA provides that tempering.[12] If the terminal equipment involves chilled ceilings or beams, they can be used to extract heat from the space and release it via the cooling coil (CC) in the DOAS.[13]

If the TER were allowed to operate for the entire time where dehumidification were not required, the annual cooling operating cost savings for the 30% ventilation increase for International Falls would be: ($35 – $27) – ($1,753 – $1,345) = –$400. This huge loss of free cooling with DOAS,[14] as a result of any unnecessary operation of the TER, must be avoided.

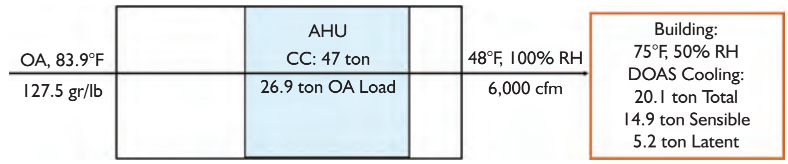
Columns 6 and 7 give insight as to the outcomes presented previously. Only New Orleans has a free cooling duration shorter than the cooling duration—528 hours. The other three regions have longer free-cooling durations than cooling, ranging from 934 to 3,440 more hours.

First-Cost Consequences

In an effort to estimate the first-cost consequences of supplying surplus OA, equipment was selected for the New Orleans and Columbus climates. The selections were for the 6,000 cfm (2832 L/s) and 4,600 cfm (2171 L/s) flow rates with and without TER. Current fourth quarter 2008 equipment street prices were obtained from a leading manufacturer. Current installation cost data were obtained from an experienced DOAS design-build mechanical contractor.

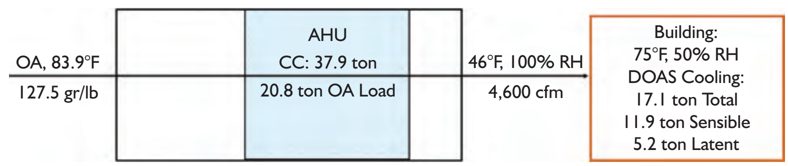
The equipment arrangements, design duty, and first and installed costs are presented in *Figures 2* through *9* for the two flow rates without and with TER. To match the space latent loads for the two flow rates, the SA DPT at the 4,600 cfm (2171 L/s) flow needed to be 2°F (1°C) lower than that of the 6,000 cfm (2832 L/s) flow.

In all cases, the higher flow with the surplus air provides more space sensible cooling, 14.9 ton versus 11.9 ton (52 kW versus 42 kW). The 3 ton (11 kW) difference in space sensible cooling must be accommodated by adding capacity to the terminal equipment, FCUs assumed for this comparison. The extra 3 ton (11 kW) capacity must also be added to the coil loads at the reduced flow when pricing the chillers.



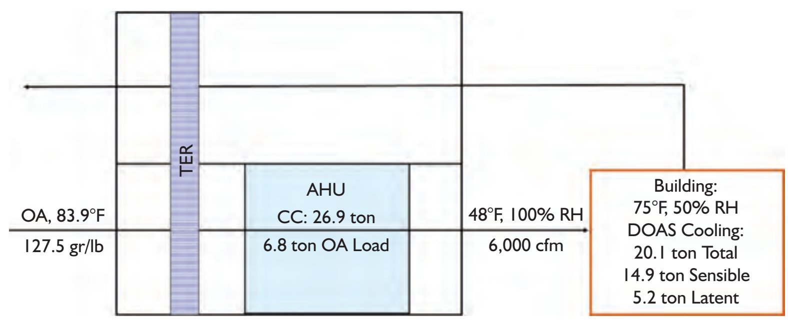
- AHU first cost: $9,900 + $9,000 installation.
- Air-cooled chiller first cost: $20,000 + $5,000 installation.
- Total installed cost: $43,900.

Figure 6: Columbus, 6,000 cfm AHU with cooling coil, no TER.



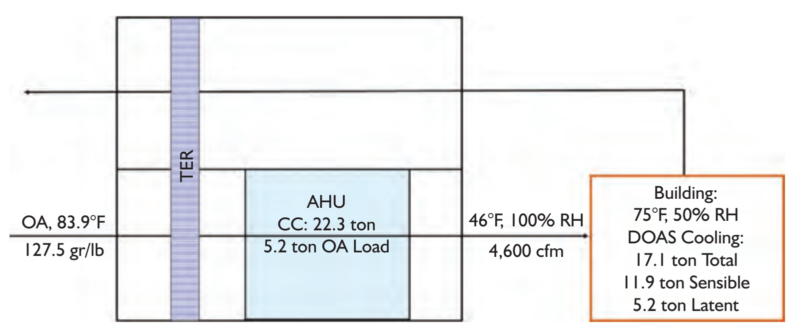
- AHU first cost: $8,100 + $6,900 installation.
- Air-cooled chiller first cost: $18,010 + $5,000 installation.
- Added FCUs to cover 3 ton of lost DOAS space sensible cooling.
  First cost: $1,440 + ($0 to $4,300 [three at $1,430 each]) installation.
- Total installed cost: $39,450 to $43,750.

Figure 7: Columbus, 4,600 cfm AHU with cooling coil, no TER.



- AHU first cost: $19,800 + $12,000 installation.
- Air-cooled chiller first cost: $11,400 + $5,000 installation.
- Total installed cost: $48,200.

Figure 8: Columbus, 6,000 cfm AHU with cooling coil, with TER.



- AHU first cost: $17,000 + $9,200 installation.
- Air-cooled chiller first cost: $11,130 + $5,000 installation.
- Added FCUs to cover 3 ton of lost DOAS space sensible cooling.
  First cost: $1,440 + ($0 to $4,300 [three at $1,430 each]) installation.
- Total installed cost: $43,770 to $48,070.

Figure 9: Columbus, 4,600 cfm AHU with cooling coil, with TER.

Case I New Orleans: Economic Comparison of 6,000 and 4,600 cfm Flow Without Total Energy Recovery (TER)			
Flow	**First Cost**	**Operating Cost Outdoor Air**	**Fan Operating Cost**
6,000	$48,800	$2,875 to $1,562 = $1,313	$1,230
4,600	$43,280 to $47,580	$2,240 to $1,198 = $1,042	$950
Extra Money for Surplus Air	$5,520 to $1,220	$271	$280
Payback Years With Surplus Air	No		
Case II New Orleans: Economic Comparison of 6,000 and 4,600 cfm Flow With TER			
Flow	**First Cost**	**Operating Cost Outdoor Air**	**Fan Operating Cost**
6,000	$49,400	$575 to $1,562 = –$987	$1,920
4,600	$44,750 to $49,050	$441 to $1,198 = –$757	$1,476
Extra Money for Surplus Air	$4,650 to $350	–$230	$444
Payback Years With Surplus Air	No		
Case III Columbus: Economic Comparison of 6,000 and 4,600 cfm Flow Without TER			
Flow	**First Cost**	**Operating Cost Outdoor Air**	**Fan Operating Cost**
6,000	$43,900	$685 to $4,297 = –$3,612	$1,230
4,600	$39,450 to $43,750	$525 to $3,296 = –$2,771	$950
Extra Money for Surplus Air	$4,450 to $150	–$841	$280
Payback Years With Surplus Air	8 to 0.3 Years		
Case IV Columbus: Economic Comparison of 6,000 and 4,600 cfm Flow With TER			
Flow	**First Cost**	**Operating Cost Outdoor Air**	**Fan Operating Cost**
6,000	$48,200	$137 to $4,297 = –$4,160	$1,562
4,600	$43,770 to $48,070	$105 to $3,296 = –$3,191	$1,204
Extra Money for Surplus Air	$4,430 to $130	–$969	$358
Payback Years With Surplus Air	7 to 0.2 Years		
Case V New Orleans, Economic Comparison of 6,000 cfm Flow With and Without TER			
TER Present	**First Cost**	**Operating Cost Outdoor Air**	**Fan Operating Cost**
Yes	$49,900	$575 to $1,562=–$987	$1,920
No	$48,800	$2,875 to $1,562=$1,313	$1,230
Extra Money for TER	$1,100	$326	$690
Payback Years With TER	0.7 Year		
Case VI Columbus: Economic Comparison of 6,000 cfm Flow With and Without TER			
TER Present	**First Cost**	**Operating Cost Outdoor Air**	**Fan Operating Cost**
Yes	$48,200	$137 to $4,297 = –$4,160	$1,562
No	$43,900	$685 to $4,297 = –$3,612	$1,230
Extra Money for TER	$4,300	–$548	$332
Payback Years With TER	<20 Years		

Table 3: First and operating cost summary.

The variation in the installation costs for the FCUs depends upon how the extra 3 tons (11 kW) are added. If it can be accomplished by upsizing equipment instead of installing additional FCUs, the added installation cost is zero. If three new 1 ton (4 kW) FCUs were added, with a per FCU installation cost of $1,430 each, regardless of size, the installation cost could be $4,300.

The cost data presented in *Figures 2* through *9*, and the operating cost data presented in *Table 2* are summarized in *Table 3*. The cross comparisons reveal the following:

- In New Orleans, a DOAS without TER (Case I) or with TER (Case II): 30% surplus OA cannot be justified based on the economics.
- In Columbus, a DOAS without TER (Case III) or with TER (Case IV) has simple paybacks from 0.2 to 9 years when a 30% surplus of OA is supplied. The time frame is sensitive to the installation costs of the extra FCU capacity required for the 4,600 cfm (2171 L/s)cases.
- Case V and VI compare the utility of TER for a fixed flow of 6,000 cfm (2832 L/s) (i.e., the issue of excess air is not present) in New Orleans and Columbus. In New Orleans, TER provides a payback of 0.7 year. In Columbus, the added first cost was also accompanied with a small decrease in operating expense, resulting in a long 20-year payback.

If the cost of electricity were doubled to $0.20/kWh, the simple payback times would be cut in half.

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Economic factors not taken into account in this analysis may yield different results than presented above, and each job must be considered independently.

Conclusions

First, the veracity of the Journal article claim concerning the cooling energy waste “madness” of garnering a LEED credit in the IEQ category has been disproved. Even Atlanta and New Orleans, locations that are not required, by Standard 90.1, to have economizers, used less cooling energy with 30% surplus OA. Significantly more energy savings were demonstrated for Columbus and International Falls, where economizers are required.

Second, the three hypotheses set forth previously were confirmed:

- A TER device substantially reduces the summer cooling energy used to treat OA;
- 30% surplus air is beneficial in the winter at reducing the cooling plant energy use; and
- The winter savings offsets the added cooling energy use during the warm months was found to be true for the locations explored.

Three, *increasing the ventilation air to spaces* with low OA cfm/person *yields big dividends* in terms of allowing the SA DPT to be elevated while still accommodating all of the occupant latent loads. *This strongly suggests a nonuniform ventilation increase strategy*.

If a space combined minimum OA/person is in the 18 to 20 cfm (8 to 9 L/s) per person range, do not increase those values at all. But for spaces with the 6 to 18 cfm (3 to 8 L/s) per person range, increase those values upward close to 18 to 20 cfm (8 to 9 L/s) per person range, then step back and assess how close the entire building ventilation has approached a total 30% increase. If, after *equalizing the flow rate per person* to about 18 cfm (8 L/s), the 30% surplus ventilation has been achieved, *take the LEED credit*.

Otherwise, abandoning the goal of gaining a LEED credit by this method may be best. Such an approach should make gaining the LEED credit possible while *significantly simplifying the equipment choices* and *avoiding elevated first cost by eliminating the need for below freezing DPTs to some spaces*. Conversely, increasing the OA flow rate beyond 18 cfm (8 L/s) per person yields diminishing returns in terms of required SA DPT or IEQ achievement.

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Three (3) Letters to the ASHRAE Journal Editor ref: June, 2009 ***30% Surplus OA*** article

From: Dr John Straube, P.Eng.
Associate Professor
Faculty of Engineering and Architecture
University of Waterloo
Waterloo, Ontario
Canada

I read Professor Mumma's article entitled "30% Surplus OA: Does it use more energy" in the June ASHRAE Journal with some dismay. DOAS are almost always the best way to provide ventilation in commercial buildings. Mumma deserves a lot of credit for getting this message out to the community. But his analysis that over-ventilation saves energy is flawed.

Even if super-efficient total energy recovery (TER) is used (like the 80% assumed), additional ventilation results in additional energy to condition and move a larger volume of air during hours when the outdoor air requires cooling, heating and/or dehumidification. Period.

Yes TER reduces the penalty, but it remains a penalty.

There is obviously a real benefit to increasing outdoor air flow, not for ventilation, but when free cooling is available. But there is no reason to continue to run an economizer all year long, under the guise of ventilation, regardless of outdoor conditions! For true high-performance buildings I often recommend the design of a VFD-driven DOAS and TER system 25-30% larger than the ASHRAE design conditions because of some large energy savings that can accrue: by operating the system at much lower flow rates than the peak design, the fan energy and noise drop dramatically, the TER rate increases. When appropriate, the extra capacity can be used as an airside economizer of course as Mumma suggests.

I might also point out that the article may be misunderstood or miss- used. Most systems installed today are, alas, not DOAS (the sole focus of the article) and the 30% over ventilation credit in LEED drives designs with both energy and moisture problems in commonly used systems. Also, in large buildings with cooling towers, a well-designed water-side economizer almost always uses less energy to cool than an air-side economizer (DOAS or not).
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From: Joseph Lstiburek

DOAS is an awesome approach - and I applaud Professor Mumma's work in this area. However, as magnificent as DOAS is it shouldn't be used to justify poor judgment on ventilation rates. The smart play is to take advantage of free cooling whenever it is available - and not over-ventilate when it is not available.

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From: Mark R. Heizer, PE LEED AP, Member ASHRAE. Associate, Sr. Mech. Eng.

Regarding Dr. Mumma's article in the June Issue of the ASHRAE Journal, I know that Mr. Lstiburek can respond much better than I, but I wish to add my own comments:

Dr. Mumma's article "30% Surplus OA: Does It Use More Energy" presented a fair comparison showing that Dedicated Outside Air Systems (DOAS) should be optimized for the building type. Contrary to intuition, the paper shows that increasing OA above the 62.1 minimum levels can save energy under most circumstances, in almost any climate type, when using DOAS.

Unfortunately, this article's focus is to counterpoint Joe Lstiburek's "Why Green Can Be Wash" article, which stated that overventilation just to get a LEED (R) point is not a sustainable practice. Dr. Mumma shows that there is one circumstance where increased ventilation can be a benefit: DOAS. Dr. Mumma's respected work has led the charge on DOAS. DOAS is sadly underutilized by our industry, in spite of it being one of the most energy efficient, cost effective HVAC systems (when natural ventilation can't be done). But for the remaining 98% or more HVAC systems that are being installed today, a 30% OA increase over 62.1 will likely increase energy use.*

Dr. Mumma should have spent less time arguing against Mr. Lstiburek. More effort showing DOAS should be considered/installed: when site optimized, it can have the bonus of higher efficiency when OA is increased. It can give engineers more data to show that DOAS is a viable, cost effective, energy saving system.

> * Members of SPC 189.1, which wanted to make this <u>CODE</u> mandatory for schools and offices, stated to me that the average building would see a 5% energy increase. I think they were under-estimating, especially once lighting, envelope, and other 189.1 improvements mandatory, the OA load becomes a much higher percentage the overall energy use. On a positive note, nearly all of the schools our office has under design use a DOAS/energy recovery system. Wish that we could get more commercial buildings to go this direction.

Dr. Stanley A. Mumma, P.E., FASHRAE
Consolidated reply to Heizer, Lstiburek, and Straube ASHRAE Journal letters to the editor.
Ref: June, 2009 ***30% Surplus OA*** article.
July 2, 2009 response.

Thanks to each of the 3 commenters for their thoughtful and excellent letters, and especially for their support and enthusiasm for DOAS.

1. My surplus-ventilation air perspective, as presented in the article, is as follows: Ventilation rates to occupied spaces in excess of those specified by ASHRAE Std. 62.1 are recommended wherever the combined (floor and occupant components) rates are less than about 15 to 20 cfm/person (see both the article's Table 1 and Figure 1). **KEY POINT:** elevating low Std. 62.1 flow rates (anything less than about 18 cfm/person) allows the SA DPT's to be sufficiently increased so mechanical refrigeration is practical, while decoupling the space sensible and latent loads. The increases are **NOT** for the purpose of garnering a LEED point; rather the LEED point, if it can be taken, is a by-product.
2. **Another KEY POINT:** the futility of increasing the combined OA/person beyond about 18 cfm, as viewed from either a dilution or SA DPT perspective, was illustrated. So to reiterate the article; take the LEED point, if it is there, after bringing all space combined OA/person flow rates up to about 18 cfm. Note; this is likely a non-uniform increase in OA/person ranging from 0 to 300%.
3. **A final KEY POINT:** the article illustrated that increasing the low OA/person cases beyond Std 62.1, for SA DPT reasons, did not result in additional annual DOAS energy use—LEED point or not so long as about 20 cfm/person is not exceeded.
4. The letters correctly note: surplus OA will increase the annual energy use and operating cost for all systems, especially VAV, with the *single exception* being DOAS for the cases illustrated in the article.
5. Reference was made to over sizing of the DOAS AHU, an excellent idea. I have continually recommended ***reserve capacity*** in all my DOAS lectures and short courses. As for ***reserve capacity***:
    a. It is needed to accommodate unforeseen latent loads
    b. It improves the total energy recovery (TER) effectiveness and provides overall energy use reduction
    c. It can expand the system free cooling capabilities.
6. Yes, buildings served by DOAS must be pressurized as noted in the letters! As a result, the supply and return air flow rates are often unbalanced, lowering the TER effectiveness compared to balanced flow. However, by employing *reserve capacity*, excellent elevated TER effectiveness is achieved. Space here does not permit discussion of options that provide 24/7 building pressurization and balanced flow in the main DOAS AHU during occupancy.
7. Finally, water side economizers are excellent where water cooled chillers are used. Realistically, the vast majority of DOAS AHUs use packaged air cooled DX equipment. On the other hand, a DOAS without a full economizer generally easily beats the energy cost budget of ASHRAE Std. 90.1. I have addressed the DOAS economizer issue in prior ASHRAE articles.

Another reader writes:

Your recent article certainly generated some interesting letters. I think that is the sign of a good article. Based on the letters, I went back and reread your article. I think I follow your premise pretty well. By dehumidifying 30% more air through the DOAS, you are able to achieve the full dehumidification of the space and then you can operate your sensible system based on higher and more efficient temperatures.

There is another way to do this, though that is sometimes seen in swimming pool designs and other similar systems. The trick is to introduce return air after the thermal energy recovery (TER) unit to get the added airflow needed to still achieve the full dehumidification. If 30% more is the "sweet-spot", then achieve that with use of return air (RA) not additional OA. This will result in less energy use. Depending on the climate, you would still need an economizer (water or air) but once you are in economizer, you should bypass the TER anyway.

I think your article was well done and helped to clarify this dehumidification potential effect. In reading the letters, I don't think they were very clear in describing how you can do everything you say without using RA. I am not sure if this is exactly what they were thinking, or if they fully understood your analysis. Anyway, I thought I would offer my 2-cents worth.

Mumma responds:

Your suggestion is a good one.

However I have always tried to stay away from any recirculation with DOAS, particularly if it will operate using demand controlled ventilation [since I choose to supply the air cold--i.e.~48F, preventing overcooling to lightly occupied or unoccupied spaces is required via either demand controlled ventilation (Mumma preference) or terminal reheat].

Verification of the proper ventilation is not likely to be achieved with systems that employ any recirculated air, as proposed in the letter above, any more than can be expected with traditional VAV systems. Further, the initial motivation of DOAS was to always meet std. 62 without invoking the multiple spaces mess.

Whenever the multiple spaces approach is a requirement, the building always needs to be over ventilated to meet the critical space, and the expected energy savings the author above expects to see, is hard to realize and is rarely obtained.