Observations, ref ASHRAE May 2013 Journal article Vol. 55, NO 5, pg. 18
"VAV Reheat vs. Active Chilled Beams & DOAS".

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Prepared by: S. A. Mumma, Ph.D., P.E., ASHRAE Distinguished Fellow (DFASHRAE)
Penn State College of Engineering Professor Emeritus

Executive Summary.

Conclusions Published in the referenced ASHRAE Journal Article. In this competition, VAVR was the clear winner versus ACB+DOAS. VAVR had much lower first costs, much lower energy costs, and similar floor-to-floor heights. These conclusions strictly apply only to the analyzed systems and this building, which is in a relatively mild climate. However, in the last five years the authors' firm has conducted detailed life-cycle cost analyses comparing several ACB variations to VAVR for several buildings across the country and has yet to come across a single case where ACB was more efficient or lower cost.

Observation 1. The design team and article authors are to be complemented for recognizing that the central benefit of DOAS is ***fatally compromised*** when air is ***centrally recirculated***. To their credit, the DOAS was 100% outdoor air.

Observation 2. The 33,900 cfm of ventilation air used in the design exceeds by 250% the 13, 574 cfm minimum required by ASHRAE Std. 62.1. The table below illustrates the requirements for the two diverse spaces.

Space category	Combined OA/person (cfm/p)	Combined OA/ft^2 floor area (cfm/ft^2)	Required SA DPT, F
Private and open offices, 62.1	17	0.09	46.7
Classrooms, 62.1	8.5	0.55	34.9
Building as designed	24	0.6	49.4

Observation 3. At 24 cfm/p it is difficult to achieve the 49.4F DPT with the 45 F campus CHWS, and even more difficult when supplying 17 cfm/p at 46.7F SA DPT. Both SA flow rates suggest the use of a passive dehumidification wheel (PDW). A DOAS employing a PDW delivering 17 cfm/p (1.1 cfm/ft^2 for the classrooms) with a SA DBT of 56F and a DPT of 46.7F opens up alternatives for the classrooms.

Observation 4. A PDW equipped DOAS would eliminate the need for ACBs in the classrooms. The highly variable occupancy classrooms should be equipped with DCV. As a result, the DOAS SA flow rate would vary with occupancy, constrained by a space DBT override, thus operate much like a conventional VAV system.

Observation 5. The office design SA flow is 41% greater than required by 62.1. Providing the extra air adds significantly to the first and operating cost, particularly since all free cooling is eradicated by reheating to 63F. The up to 41% extra air can be delivered by pairing the 17 cfm/p DOAS with locally recirculated air from ***parallel*** booster fan/sensible cooling units to increase sensible cooling output during peak load conditions.

Observation 6. Reheating air from below 50F to 63F in an effort to minimize terminal reheat is an absolute energy and first cost travesty. Davis, CA has about 4,500 hours when the OA must be cooled and dehumidified. Reheating that air wastes 145,000 ton-hours of cooling plus the reheat energy, and this is just the tip of the problem. Minimal if any reheat would be required in the classrooms employing DCV, and it would be most difficult to overcool the office areas with 0.09 cfm/ft^2 of SA at 56F. A neutral SA temperature is rarely beneficial.

Conclusion. This observer gets the sense that two straw men systems were set up to be knocked off by VAVR. A different approach as described above would have substantially reduced fan energy, cooling and heating energy, and equipment first cost. If the two alternates to VAVR were not straw men, it illustrates the urgency for ASHRAE to put a DOAS design guide into the hands of its membership.

Detailed Technical Support for the Observations above start next page.

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Introduction

The observations to be discussed here will be limited to the active chilled beams (ACB) with DOAS system. This Observer, a DOAS scholar, has been very cautious about recommending active chilled beams (ACB) with DOAS since, as noted in the articles, most ACBs require more supply air than is needed to meet the ASHRAE ventilation criteria in order to deliver their published sensible cooling capacity. As a result, most ACB jobs use centrally recirculated air with the ventilation air. Using centrally recirculated air makes meeting the ventilation criteria of Std. 62.1 much more difficult, and generally elevates the energy use. Overcoming these problems is the central reason to use DOAS. One may ask, couldn't it still be called DOAS when using centrally recirculated air, to which the answer must always be NEVER, since its central benefit is sacrificed.

The ACB design team referenced in the article did not make this mistake, to their credit. Unfortunately, other observations to be discussed were not as positive.

Defining the ventilation criteria for the UC Davis facility.

The article defines the building as 56,500 ft^2 with the floor area equally divided between private offices, open offices, and classrooms. Since the article was published in the ASHRAE Journal, targeting a worldwide audience well beyond California and its Title 24 criteria, this observer has chosen to use ASHRAE 62.1 ventilation criteria as summarized in the following table.

use	area, ft^2	p/1000 ft^2	scfm/p p-comp	scfm/ft^2 FL-comp	OA	avg. OA scfm/ft^2	scfm/p
private Office	18,833	5.00	5.00	0.06	1,601	0.09	17.00
open office	18,833	5.00	5.00	0.06	1,601	0.09	17.00
classroom	18,833	65.00	7.50	0.06	10,372	0.55	8.47

Notice that the ASHRAE ventilation criteria is met with a total of 13,574 scfm of 100% OA, or the flow required by the DOAS unless thermal criteria govern. The article states that the constant flow DOAS in the ACB design supplies 33,900 scfm, or 0.6 scfm/ft^2. This is 250% the ventilation requirement, said to be made necessary to get the desired sensible capacity out of the ACBs. This will be addressed in some detail later**.** Had this observer used the California Title 24 ventilation criteria, the numbers would have been a little different, but not the basic conclusions, which likely explains why the authors of the article did not highlight which criteria was used for design. One of the authors served as past chair of the ASHRAE Std. 62.1 committee, so is well qualified to address this issue and chose not to.

Next, let's explore another critical design variable, the required DOAS SA DPT necessary to avoid condensation while accommodating the space latent load.

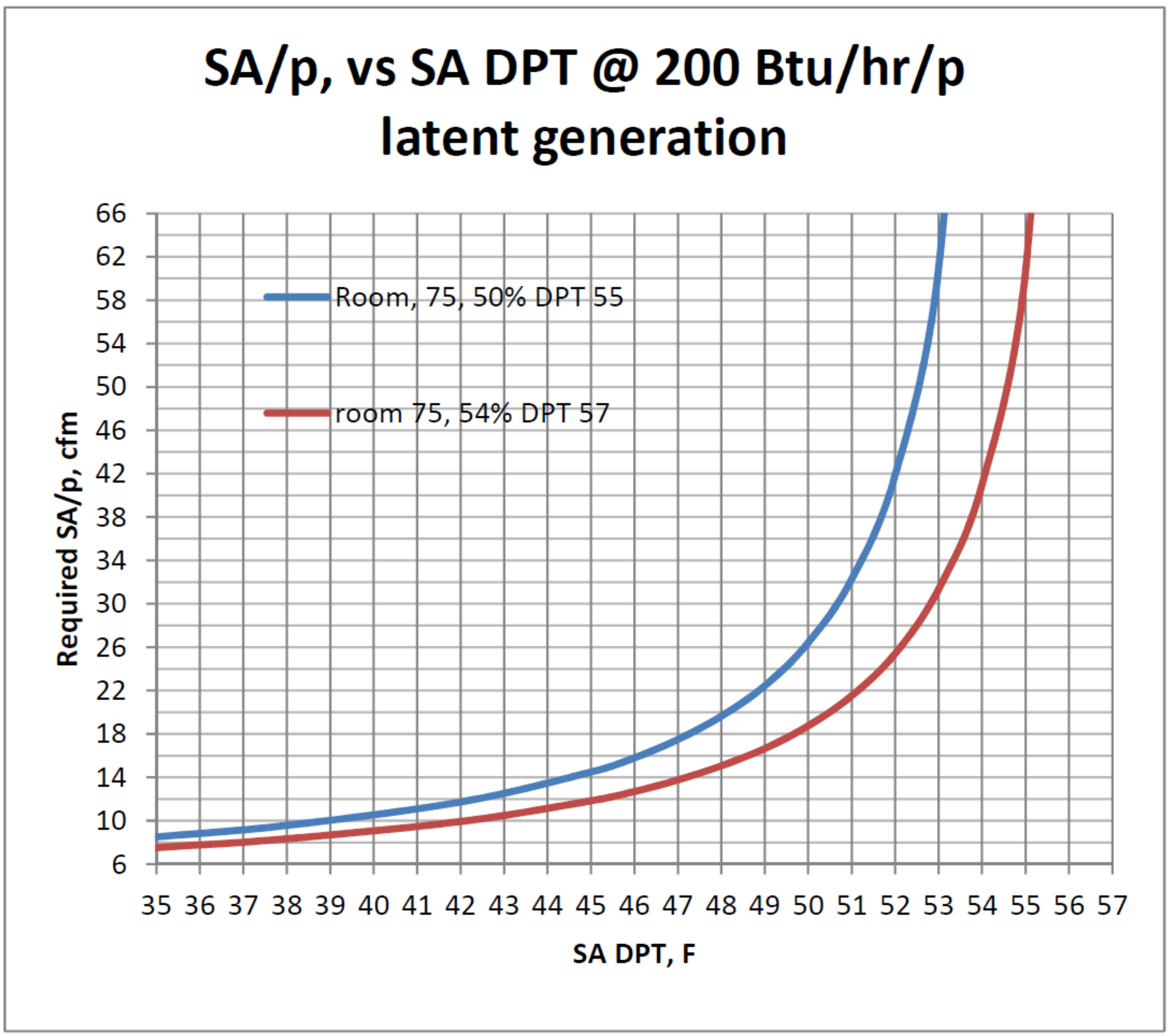
Criteria necessary to meet the space latent loads

If we assume that the building design and operation preclude infiltration, then the internal latent load is essentially occupancy based. If we assume that the latent load for moderately active office workers and non-sleeping engaged students are about 200 Btu/hr, and that for condensation control the space is maintained at 75 F DBT and 50% RH (humidity ratio of 64.9 gr/lbm).

Solving Q_L/p =0.67*scfm/p*(Wspace-Wsupply) for Wsupply, *[200=0.67*17*(64.9-Wsupply)],* one gets Wsupply= 47.3 gr/lbm (i.e. required SA DPT= 46.7F) with the office flow rate per person of 17 scfm. At the classroom ventilation flow rate of 8.5 scfm/person, the computed Wsuppy= 29.8 gr/lbm (i.e. required SA DPT= 34.9 F). The article does not provide the design SA DPT, but since there is just one DOAS unit, all spaces receive the same DPT. If so, in order to remove the occupant generated latent load in the highly variable occupancy classrooms, and maintain the space DPT around 55F to avoid ACB condensation, the OA to the classrooms must be increased from 8.5 to 17 scfm/p, i.e. doubled. The result of this step would be to increase the total flow for the DOAS from 13,574 to 23,946 scfm.

Commonly, the cooling water to ACBs is held about 2 F above the design space DPT. While not stated in the article, it is assumed that space design DPT is 55 F for condensation avoidance since 57 F cool water is supplied to the ACBs. Employing the relationships above, it is easy to compute the required SA per person for latent load control. The figure below illustrates this relationship. As the SA DPT increases, the required SA flow increases rapidly approaching infinity. The ASHRAE Std. 62.1 default occupancy density, noted in the table above, yields 188 people in offices, and another 1,225 in classrooms for a total building head count of 1,413 occupants. With a published constant volume DOAS design flow of 33,900 scfm, the resulting SA flow rate per person is 24 scfm. At 24 scfm/p, the SA DPT required to maintain a space DPT of 55 F is 49.4 F. This SA DPT is difficult to achieve with the 45 F CHWS published in the article. **Note**: the majority of DOAS designs do not have access to a campus chilled water system, and hence use DX DOAS (generally equipped with exhaust air energy recovery rather than passive dehumidification wheels) that can easily deliver the required low DPTs and a “high” temperature (55 F) and efficiency chiller for a myriad of parallel sensible cooling equipment including ACBs.

In order to drop the SA DPT, when supplying just the Std 62.1 required 17 scfm/person (as opposed to the design 24 scfm/p), to 46.7 F using 45 F CHWS with assurance would seem to suggest the introduction of a ***passive dehumidification wheel*** in the DOAS unit. In that combination, 51 F saturated air (enthalpy of 20.85 Btu/lbm) leaving the CC served with 45 F CHWS would be nearly adiabatically dried and heated to the required 46.7 F SA DPT with an approximate leaving DBT of 56 F. Introduction of the passive dehumidification wheel eliminates concern about chiller energy performance degradation when required to produce approximately 46F SA, and the associated fear of condensation in the ductwork and ACBs if supplied with 46F DBT air.



Understanding these conditions and criteria, causes this observer to ask: is the huge published increase in DOAS OA flow necessary, are ACB necessary in the classrooms, and is there a potential for overcooling?

<u>One of many alternative ways to handle the space sensible and latent loads in the *CLASSROOMS.*</u>

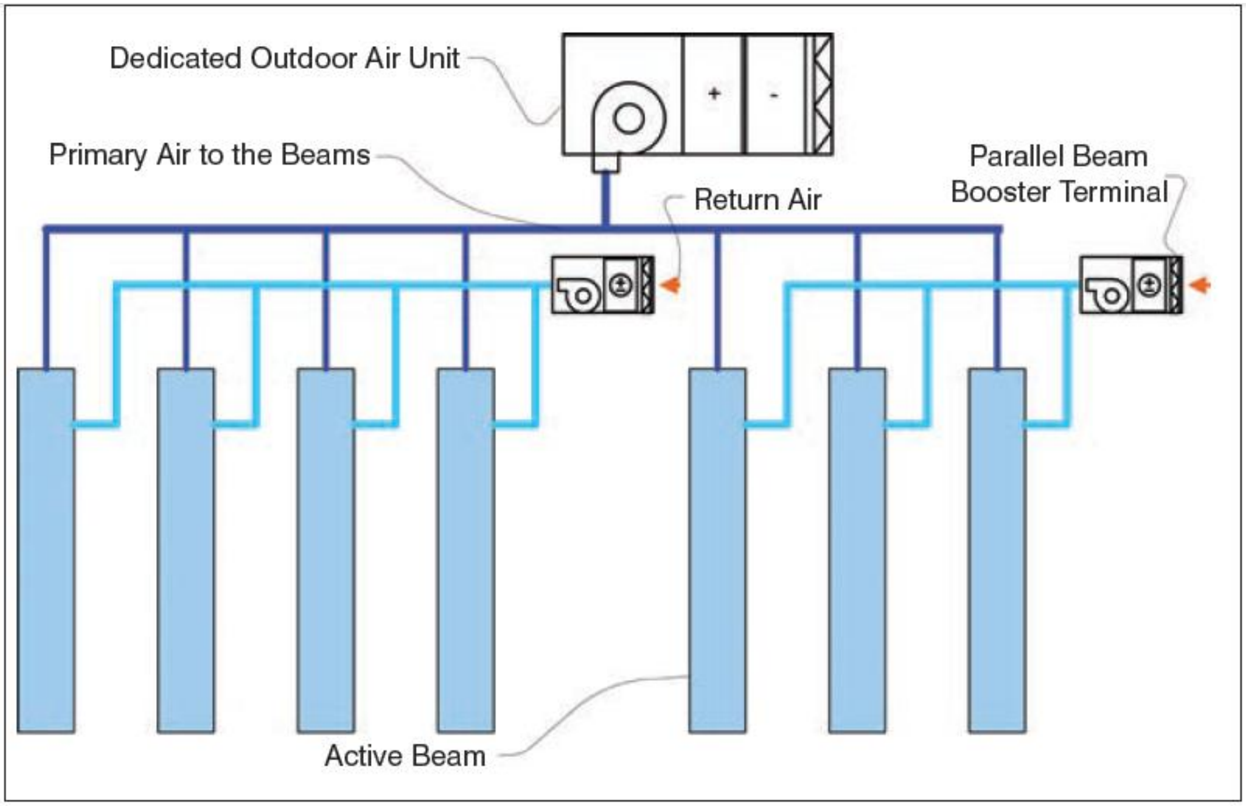
Classrooms are generally highly variable occupancy density facilities. As such they should be equipped with demand controlled ventilation to both conserve central cooling and heating energy use as well as terminal reheating energy. Unfortunately, to avoid or minimize terminal reheating, many designers are of the persuasion that the DOAS air should be centrally tempered, as was done by the ACB designers in the article, i.e. the constant volume primary air was delivered at a constant 63F. This introduces a serious energy penalty, as well as a first cost penalty reflected by the added cooling capacity required and sensible cooling terminal equipment. The consequences of this elevated SA DBT, i.e. 63F as opposed to the 50F necessary for latent control, will be discussed later.

When a DOAS unit equipped with passive dehumidification delivers 17 scfm/p (1.1 scfm/ft^2) of air at the required 46.7 F DPT (associated supply DBT of 56 F), its sensible cooling capacity essentially equals that of an ordinary VAV system supplying 55F DBT air. In other words, there is more than enough capacity to obviate the elimination of ACB in the classrooms. Once the beams are gone, ACB condensation issues in the classrooms are eliminated, and the space DPT can be allowed to rise to 65F (84.7 gr/lbm). Using the above equation *[200=0.67*scfm/p*(84.7-47.3)]* to solve for the new scfm/p, one finds that only 8 scfm/p is required to meet the space latent load.

Conclusion, the classrooms can be supplied with the DOAS air as necessary to meet the space sensible loads employing its own variable volume SA delivery arrangement that also uses DCV. And as noted above, since the DOAS supply air DBT essentially equals that of a conventional VAV system, any proclaimed fan energy penalty is eliminated. Further, the DOAS will always use less terminal reheat to meet the ventilation requirement compared to conventional VAV when in the recirculation mode. Example, let’s say a space needs 100 scfm of ventilation air, which is that supplied by a DOAS, but a conventional VAV recirculation system using 50-80% completely viciated recirculation air needs to supply 200 to 500 scfm of SA (100/0.5 to 100/0.2) to the room to actually deliver 100 scfm of OA. In the conventional VAV, all 200-500 scfm are subject to terminal reheat, not just the necessary100 scfm supplied by a true DOAS.

<u>Alternate way to boost the sensible capacity of the ACBs in the office areas while using just the ventilation requirement and no central recirculation.</u>

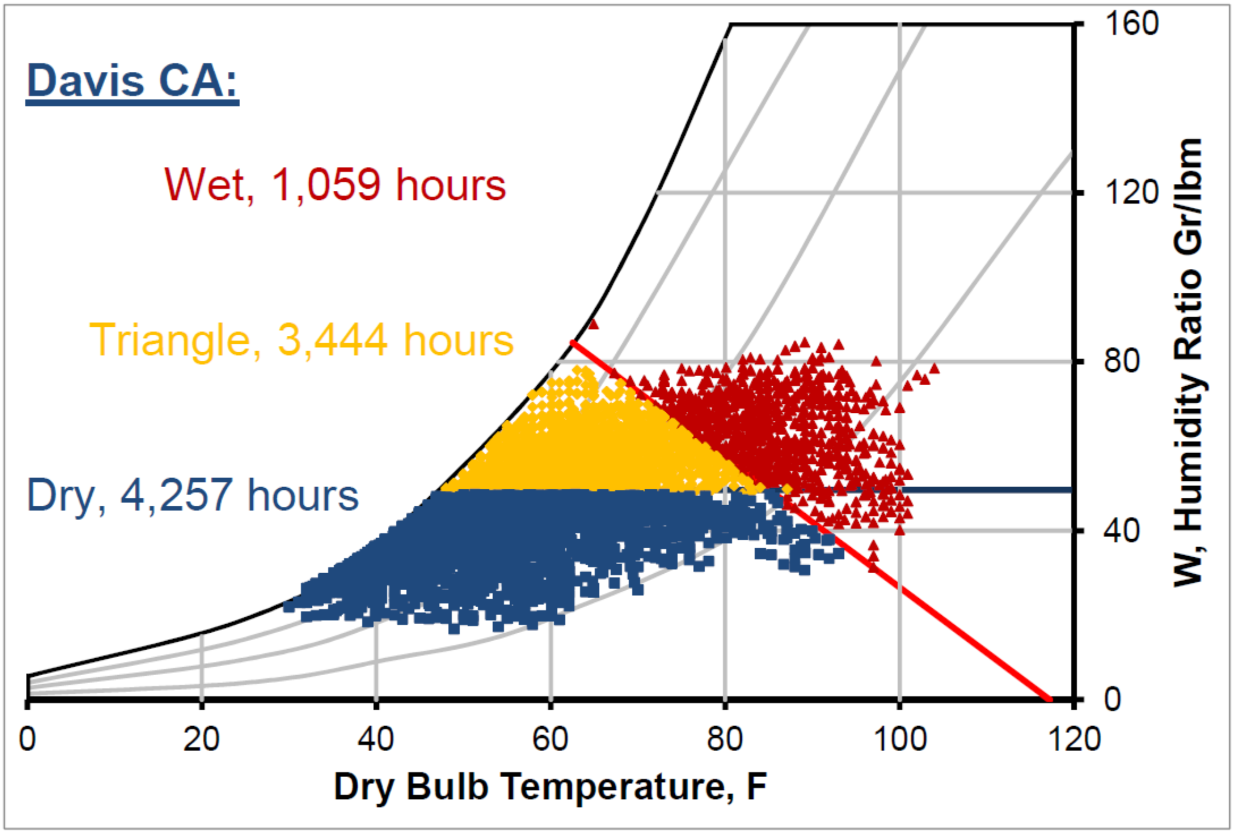
This solution first came to the attention of this observer via the April 2012 ASHRAE journal article by Livchak entitled "Don't Turn Active Beams into Expensive Diffusers". The idea is to pair ACBs with booster fans to increase air flow and sensible capacity at peak loads. A graphic illustration is as follows:



Engineers can pair an ACB system using 100% DOAS supply along with a small quantity of locally recirculated air from *parallel* booster fan/sensible cooling units to increase cooling output during peak load conditions.

Impact of climate on the design and controls of the ACB approach.

The project discussed in this article is located in Davis, CA, a location extremely climate friendly compared to virtually all locations east of the Rockies where 80% of the USA population lives. Engineers designing for this area of the USA, largely dominated by conventional VAV systems, virtually never have a concern for humidity problems. This paradigm must change when the design uses ACBs. A plot of the 8,760 hours of weather data is presented below.



The climate is temperate with just a few hours where the DBT drops below 40F, and it is relatively dry where the humidity ratio rarely even reaches 80 gr/lbm. Many of us living east of the Rockies see both sub 0 F winters, and summers with the humidity ratio above 140 gr/lbm for a great many hours. As a result, the utility of exhaust air energy recovery differs vastly East to West. In climates like Davis, where the OA humidity ratio is below that needed to take care of the entire space latent loads for half the year, the DOAS SA controls need to reset the SA DBT during these hours to just meet the critical space sensible cooling loads (see these links: http://doas.psu.edu/OR-05-3-3.pdf, & http://doas.psu.edu/DOAS_Field_Experience_Denver_05_2_1.pdf).

Since the Davis weather data has been illustrated, this is a good place to contrast the OA cooling/dehumidification energy use and cost difference to treat the DOAS required OA and that used by the ACB designers, 13,574 vs. 33,900 scfm.

OA flow, scfm	Annual TH of cooling based on dehumidification hours	Annual energy cost based on 0.7 kW/ton & $0.10/kWh
13,574	57,890	$4,050
33,900	144,575	$10,120

As bad as this looks in terms of energy waste, if untempered OA were supplied when the OA DBT is below 63F, the extra 20,326 scfm of OA would do about $7,250 worth of extra free cooling. In other words the penalty from cooling and dehumidifying the extra air is more than offset when it is dry outside and properly controlled. Problem with the extra OA is its adverse impact on equipment first cost and fan energy use.

<u>Consequences of elevating the SA DBT to 63F</u>

Based upon the weather data for Davis, CA presented above, there are about 4,500 hours where the air must be dehumidified to meet the space latent loads. The ACB-DOAS approach supplied 33,900 scfm of 100% OA, or an estimated 24 scfm/person. That air flow per person was shown above to require a SA DPT around 50F. Assuming a saturated coil leaving condition, the SA DBT would also be 50F. Elevating the SA DBT mechanically to 63F makes about 40 tons of cooling unavailable via the DOAS air (1.08*33900*(63-50)/12000=39.7 ton). This loss would occur for every hour dehumidification is required, or 4,500 hours, requiring the expenditure of 180,000 ton-hours of cooling wasted, worth $12,600 assuming the cooling plant operates at 0.7 kW/ton and electricity costs $0.10/kWh. The diagrams in the article show no heat recovery as the reheat energy source, so if it were assumed that the temperature is elevated 13F with electric resistance heat (downstream of the cooling coil but not shown), the added energy would be (1.08*33900*(63-50)=475,977 Btu/hr or 140 kW.

> An observation about the schematic Figure 2, Active Chilled Beam Design: if the heating coil and cooling coil with bypass damper were used to produce the required SA conditions, all of the OA would need to be heated to about 145F then 19% of it blended with the remaining 81% entering the cooling coil at 145 F and leaving at a saturated 45F to produce the 63F SA DBT. This would be a profound energy penalty for many reasons. Conclusion, Figure 2 is incomplete.

This would also occur for the entire 4,500 hours representing 2,140 Million Btu, or 630,000 kWh. If electricity cost $0.10/kWh, this reheat would cost $63,000 just for the time the air is being mechanically dehumidified. As bad as this is, the space loads must still be met requiring that the cooling energy wasted by heating the SA to 63F must be borne by the terminal ACBs and the cooling plant serving them. The result is significant added energy use cost, added demand charge, and first cost for the added ACB capacity and central plant capacity. Unfortunately the bad news does not stop here.

During the periods when no dehumidification of the OA is required, i.e. when the OA DPT is below about 50F, there are many hours when free cooling can be quite effective, in this case all hours when the OA DBT is below 63F which is about 3,700 hours. Unfortunately, the article does not provide building design and use details to the extent necessary to compute how much of this free sensible cooling can be used, but if it could all be used, it would amount to 173,000 ton-hours valued at about $12,000. Instead, it appears that all OA at a DBT below 63F is tempered to 63F, requiring another 681,000 Kwh of resistance heat at about $68,100.

Conclusion regarding tempering the SA to 63F, it is a serious error that totally skews the results and the voracity of the Journal article. Further, there is little or no need for tempering with DCV in the classrooms, and only 0.09 cfm/ft^2 of 56F SA to the offices.

Conclusion

While no system is best for every application, this observer gets the sense that two straw men systems were set up to be knocked off by VAVR. A different approach as described above would have substantially reduced fan energy, cooling and heating energy, and equipment first cost. Engineers are hired to solve unique problems using first principles. If the two alternates to VAVR were not straw men, it shows the urgency for ASHRAE to put a DOAS design guide into the hands of its membership to facilitate good problem solving and design. It is unfortunate that the ASHRAE Journal Editor did not solicit and publish opposing views.