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WHITE PAPER - A Real-World Analysis of Chilled Beams vs. VAV

A review of the ASHRAE Journal article "VAV Reheat versus Active Chilled Beams with DOAS"

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What you should know and why ...

The May 2013 issue of the ASHRAE Journal features an article titled *VAV Reheat versus Active Chilled Beams & DOAS* which compares a "streamlined" VAV system to inefficiently designed active beam/DOAS system. The subject of the article is a classroom/office building at the University of California at Davis. The facility is a three story building with 56,500 ft2 of conditioned space. It has a very low occupant density (275 ft^2 per person) and representative space sensible cooling loads (averaging 19.5 Btu/h-ft^2). The article compares first costs and energy usage while assessing other claimed benefits of ACB systems.

Stacking the deck

The ACB system as described in the article has a number of shortcomings that lead to many of the authors' erroneous conclusions, which include:

- A DOAS air handler without any energy recovery continuously delivers 100% outside air.
- Primary air is delivered at 63˚F with an insufficiently depressed (54˚F) dew point temperature.
- The beams described all have four pipe coils to enable heating and cooling throughout the building. They employ unusually low primary airflow rates (about 8 CFM/LF) and their water side cooling efficiencies (20 to 25 Btu/h-CFM of primary air) are very low due to the high ACB system primary airflow rates required for space latent cooling.

On the other hand the VAV system described is an unconventional design which employees a number of control capabilities that aren't afforded the ACB system and provide significant advantage against it:

- Interior terminals have no reheat provisions, thus no interior hot water piping is required.
- Terminals are capable of full shut off when CO^2 sensors allow. The ACB system includes no set back provisions.
- The VAV system is designed for duct velocities of 2,000 FPM, the ACB system 900 FPM.
- The VAV system is designed with 55˚F primary air which can be reset to 63˚F when space load and outdoor air conditions allow. The ACB system's 63˚F primary temperature cannot be reset.

Playing the cards that were dealt

The abnormally high primary airflow rate (0.53 CFM/ft^2) of the ACB system is a result of the high dew point temperature of the primary air. Based on the space latent load of one person per 275 ft^2, the primary air humidity ratio entering the room would be only 2.7 grains lower than the room design. The VAV system's 52˚F dew point primary air provides a 7.9 grain differential between the room and supply air. As such, the ACB system primary airflow for space moisture removal is almost three times that of the VAV system. The ACB system performance and cost was subsequently skewed by this exorbitantly high primary airflow rate.

The authors state that the VAV system at full design space loads operates at 0.88 CFM/ft^2. They further suggest that average space sensible loads are only 40% of design, thus the VAV system's average airflow delivery is only 0.35 CFM/ft2. The ACB system operates at 0.53 CFM/ft2 at all times resulting in six times the fan energy consumption of the VAV system according to the authors.

The authors suggest that the ACB system requires 40% more cooling and 3.4 times the heating energy than the VAV system. This is again attributable to the inefficient DOAS that is employed. The VAV system operating at design needs about 0.15 CFM/ft^2 of outside air. If that is the case, why use a DOAS (much less one without heat recovery) when the outside air requirement is only about 30% of the required primary airflow? The fact that so much outdoor air is being continually conditioned and exhausted would explain the cooling and heating energy differences.

Table 1 summarizes the design primary airflow requirements for the VAV systems described by the authors. The primary airflow rate for the ACB system is determined by the space latent cooling requirement results and primary airflow rates 3.5 times the space ventilation requirements. Had a similar primary air dry bulb and dew point temperature been applied for both systems, the ACB system primary airflow rate could have been reduced to below 0.2 CFM/ft²!

First cost claims stated by the authors include:

- The ACB first cost ($62/ft²) is 2.5 times that of the VAV system
- The ACB system uses 30 times as much chilled water piping and 5 times the hot water piping
- ACB system subcontractor costs are 2.5 times higher due to more water valves, humidity sensors and higher test and balance costs for all of the extra control points.

	VAV System 55° DB/54° DP ΔW = 7.9 grains		ACB System 63° DB/54° DP ΔW = 2.7 grains	
	Interior Space	Perimeter Space	Interior Space	Perimeter Space
Ventilation	0.15	0.15	0.15	0.15
Dehumidification	0.18	0.18	0.53	0.53
Sensible Cooling	0.60	1.75	0.22	0.53
Resultant Airflow	0.60	1.75	0.53	0.53

Table 1: System airflow requirements as described in article

Leveling the playing field

Data revealed within the article allows review of the comparison at three distinct operating conditions, sensible and latent outdoor design (for which equipment capacity requirements are derived) with design space loads and shoulder season operation when the space sensible cooling load is 50% of design.

Properly designed ACB systems rely on primary air that is conditioned at the air handling unit to a dew point temperature similar to that provided by conventional all air systems, in this case 52˚F. If the ACB system had employed primary air delivered to the space at the same conditions as the VAV system, the design and energy performance story would have been drastically different. Table 2 illustrates the predicted energy performance of the systems at three defined operational scenarios as described followed by that which would have been predicted for properly designed ACB systems. Note that the fan energy of the modified ACB systems remains the same in all cases but the cooling energy requirement is significantly less when a mixing type AHU replaces the DOA unit.

Sensible design conditions (100% sensible & latent space loads)	SAT F°	OA CFM	RA CFM	OA %	AHU cooling kW	Beam cooling kW	Fan BHP	Pumps BHP	Total Energy kW	
VAVR system as described	55	8,475	41,525	17%	49.9	0.0	45.6	5.5	**88.0**	Similar
ACB system as described	63	30,000	0	100%	40.5	23.5	26.0	8.7	**89.9**	
Modified ACB system										
Primary air @ 55° F DB, 53°F DP	**55**	16,667	0	100%	28.7	24.4	4.5	7.6	**62.1**	30% less than VAVR
Mixing at air handling unit	**55**	**8,475**	**8,192**	51%	21.6	24.4	4.5	6.8	**54.5**	38% less than VAVR

Latent design conditions (75% sensible & 90% latent space loads)	SAT F°	OA CFM	RA CFM	OA %	AHU cooling kW	Beam cooling kW	Fan BHP	Pumps BHP	Total Energy kW	
VAVR system as described	55	8,475	29,025	23%	26.6	0.0	26.0	2.9	**48.2**	
ACB system as described	63	30,000	0	100%	40.2	14.3	26.0	7.0	**79.2**	64% more than VAVR
Modified ACB system										
Primary air @ 55° F DB, 53°F DP	**55**	16,667	0	100%	28.5	15.3	4.5	5.9	**51.5**	6% more than VAVR
Mixing at air handling unit	**55**	**8,475**	**8,192**	51%	10.3	15.3	4.5	3.9	**31.8**	34% less than VAVR

Shoulder season operation (50% sensible, 80% latent space loads)	SAT F°	OA CFM	RA CFM	OA %	AHU cooling kW	Beam cooling kW	Fan BHP	Pumps BHP	Total Energy kW	
VAVR system as described	55	8,475	16,125	34%	13.0	0.0	6.6	2.0	**19.4**	
ACB system as described	63	30,000	0	100%	15.6	5.1	26.0	1.4	**41.2**	112% higher than VAVR
Modified ACB system										
Primary air @ 55° F DB, 53°F DP	**55**	16,667	0	100%	8.4	6.1	4.5	2.0	**19.3**	Same as VAVR
Mixing at air handling unit	**55**	16,667	0	100%	8.4	6.1	4.5	2.0	**19.3**	

Table 2: System energy comparisons

The first cost estimates would be significantly reduced with a properly designed ACB system. Equipment costs associated with the significant reduction in the primary airflow rates (and resultant cooling requirements) will be much lower. Substitution of beams with more realistic primary airflow (16 CFM/LF) and water side cooling capacities (44 Btu/h per CFM) combine with the increased sensible cooling contribution of the cooler primary air to allow the number of beams to be reduced by 35 to 40% (see table 3 below), proportionally reducing the associated piping and balancing costs as well.

	VAV system as described			ACB system as described			Modified ACB system		
	Interior zones	Perimeter zones	All zones	Interior zones	Perimeter zones	All zones	Interior zones	Perimeter zones	All zones
Area served ft^2	38,922	17,578	56,500	38,922	17,578	56,500	38,922	17,578	56,500
BTU/H-ft^2	13.4	33.0	19.5	13.4	33.0	19.5	13.4	33.0	19.5
CFM_{PA}	23,634	26,367	50,000	13,123	16,877	30,000	7,878	8,789	16,667
CFM_{PA}/ft^2	0.61	1.50	0.88	0.34	0.96	0.53	0.20	0.50	0.29
BTU/H-CFM_{PA}	22.0	22.0	22.0	39.6	34.4	38.0	66.0	66.0	66.0
ACB Btu/h-LF				305	550	398	1,056	1,056	1,056
ACB CFM_{PA}/LF				7.7	16.0	10.9	16.0	16.0	16.0
Linear feet of ACB required				***1,706***	***1,055***	***2,761***	***492***	***549***	***1,043***

Table 3: Beam selection for various system designs

	VAV system as designed		ACB system as designed		Modified ACB system
	Cost or Qty.	Cost/CFM	Cost or Qty.	Cost/CFM	Cost or Qty.
Material cost ($)	$215,179	$4.30	$576,496	$19.22	$320,282
Labor cost ($)	$584,058	$11.68	$1,509,349	$50.31	$838,544
Equipment cost ($)	$319,695	$6.39	$608,349	$20.28	$337,978
Subcontractors ($)	$252,067	$5.04	$647,037	$21.57	$359,472
Lbs. of ductwork (lbs)	38,000		28,612		15,896
Chilled Water Piping (LF)	310		10,244		5,691
Hot Water Piping (LF)	2,085		9,630		5,350
Total HVAC cost ($)	$1,370,999		$3,341,231		$1,856,277
HVAC cost ($/$ft^2$)	$25	$27	$62	*$111*	$33

Table 4: Installed cost comparison for the systems described

Modified ACB system cost assumes the cost per CFM for the ACB system remains constant and thus system costs are proportional to the reduced primary airflow requirements. These costs also do not include any possible reheat piping reduction opportunities discussed earlier.

Table 4 illustrates the authors' installed cost estimates as well as predicted costs for the modified ACB system. The costs for the modified system assume that the cost per CFM of primary air for the ACB systems remains proportional to its primary airflow requirement. While this will undoubtedly vary, other factors such as the discrepancy in duct design velocities, the absence of interior space heating provisions in the VAV system and the obvious differences in thermal zoning of the two systems will tend to drive the cost of the modified chilled beams closer to that of the VAV system. The cost premium illustrated is representative of that encountered (10 to 15%) on other completed chilled beam projects.

Moving forward

The clear intent of the article was to discredit preferences given to chilled beam systems. The authors compared the VAV system to an ACB system whose performance and first cost destiny was doomed by its air handling unit strategy. On the other hand, the described VAV system design and performance are so streamlined that its compliance to ventilation codes is marginal and its ability to achieve the authors' energy usage claims is questionable.

Despite its obvious bias, the article does highlight some very useful information. It does a wonderful job of educating its readers how not to apply an ACB system by illustrating do's and don'ts in ACB system design, such as:

- DOAS systems are not always required with ACB systems. This example illustrates a case where the DOAS strategy significantly impaired system energy performance. Certain applications such as laboratories and healthcare require 100% outside air and others such as classrooms may require only slightly more primary air than the ventilation requirement but use of 100% OA when it constitutes a small fraction of the primary air is not advisable.
- When DOAS strategies are applied, total energy recovery should be employed.
- ACB systems should employ primary air at conditions (temperature and moisture content) that are similar to that used with conventional all air systems.

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