CODES AND STANDARDS ENHANCEMENT INITIATIVE (CASE)

Final Measure Information Template – Laboratory Exhaust VAV and Reheat

2013 CaliforniaBuilding Energy Efficiency Standards

California Utilities Statewide Codes and Standards Team, October 2011

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Measure Information Template
Labs – VAV and Heat Recovery

2013 California Building Energy Efficiency Standards

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1. Overview

1.1 Measure Title
Energy recovery and variable volume labs

1.2 Description
This measure would apply to laboratory supply and exhaust air systems in California.

1.3 Type of Change
This measure would be a prescriptive requirement.

1.4 Energy Benefits
The 2008 Standard allows supply and exhaust in labs to be constant volume. Most labs are currently designed as 100% outside air to all spaces including non-laboratory support spaces like offices and conference rooms. There are currently no requirements for energy recovery for high ventilation spaces. This proposal addresses both of these measures: design for variable air volume and energy recovery.

Variable air volume systems save significant energy compared to constant volume systems. Labs typically have minimum ventilation requirements based on processes and the chemicals or other contaminants that are used in the lab. These minimums for dilution of contaminants typically range from 4 to 12 air changes per hour (ACH) and are typically set by the facilities environmental health and safety departments (EH&S).

Labs also have relatively high airflows at design conditions. This high airflow is primarily due to two factors: make-up air for the fume hoods, and the relatively high equipment loads in many labs (it is not uncommon to have as much as 30 w/ft2 in a laboratory equipment room). A constant volume laboratory supply and exhaust design provides the design airflow at all times. These typically range from 12 to 20 ACH. Functionally, this high air change rate is not required at all times;it is only required when the hoods are open or the loads are high. The remainder of the time, the airflow can be turned down to the minimum ventilation requirement for the space. Variable volume systems save fan energy by reducing the volume of air. They also save heating and cooling energy by reducing the intake of outside air and the amount of reheat when the hoods are open but the loads are low.

Energy recovery systems work on both constant volume and variable volume laboratory designs.They save energy by recovering heat (or coolth) from the exhaust and using it to preheat or precool the outside air. Energy recovery systems use air-to-air or air-to-water-to-air heat exchangers between the exhaust airstream and the ventilation air stream. During the winter, heat can be transferred from the exhaust air to the ventilation air. During the summer, heat can be transferred from the ventilation air to the exhaust air. The energy transferred can be sensible and latent or sensible only, depending upon the type of energy recovery system.

The most common methods of energy recoveryfor laboratory buildings are runaround coils. You will also find in some designs the use of air-to-air heat exchangers, enthalpy wheels, and heat pipes. Run-around coils are the most common as it is typical to have the OSA intakes spatially distant from the exhaust to prevent entrainment of the contaminants in the exhaust air. Although air-to-air heat exchangers, enthalpy wheels and heat pipes are more efficient than run-around loops, they present significant design challenges as they require the exhaust air and outdoor air streams to be located adjacent to each other. Runaround coils, on the other hand can be added to almost any design. It requires two air-to-water heat exchangers, a pump, and some piping. Therefore, the analysis was done assuming runaround coils rather than enthalpy wheels or heat pipes even though the effectiveness is lower.

Runaround coils only recover sensible heat. The capacity of the runaround coils are usually controlled by varying the flow of the water in the loop. You can reduce the fan energy penalty of these coils by including a coil bypass on the outdoor and exhaust air coils.

The energy analysis performed for both the VAV and energy recovery measures are described in detail below in Section 2 and results are given in Section 3. A summary of the results are shown below in Table 1 and Table 2. The prototype building used is a mix of lab and office space, totaling approximately 170,000 square feet. The majority of the lab spaces (~34,000 square feet) are served by a dedicated system with a minimum air change rate of 10 air changes per hour (ACH). The building is in climate zone 12. In the energy recovery case, the runaround loop has an effectiveness of 0.28.

	Electricity Savings (kwh/yr)	Demand Savings (kw)	Natural Gas Savings (Therms/yr)	TDV Electricity Savings	TDV Gas Savings
Per Prototype Building	1,377,172	285.2	45,134	$233,397	$60,308
Savings per square foot	41.3	0.0085	1.4	$7.00	$1.81

Table 1. Energy savings, VAV, Climate Zone 12

	Electricity Savings (kwh/yr)	Demand Savings (kw)	Natural Gas Savings (Therms/yr)	TDV Electricity Savings	TDV Gas Savings
Per Prototype Building	20,802	61.8	8,644	$2,269	$11,957
Savings per square foot	0.62	0.0019	0.26	$0.07	$0.36

Table 2. Energy savings, energy recovery, Climate Zone 12

1.5 Non-Energy Benefits

The non-energy benefits for a VAV system include the following: reduced acoustical noise; better comfort (due to the reduction of drafts); in general less disruption of the airflow at the face of the fume hood sashes (again due to reduction of drafts); reduced wear on the fan motors, belts and bearings; feedback and alarming for room air balances and sash velocities (these are an integral part of the VAV controls and are often not provided on CAV systems; and less disruption of airflow from retrofits in other spaces. This latter point is a critical advantage of the VAV design as described below. VAV systems also make it much easier to accommodate future changes to the zoning, typically this can be handled by simply reprogramming the supply and exhaust valves. VAV systems use standard off-the-shelf technologies.

VAV systems are intrinsically safer than constant volume systems when spaces are being modified. In a constant volume system, if a hood is added, removed, or modified (which is common over the course of a lab's life), the entire system must be rebalanced in order to maintain the correct airflows. However, due to the high cost of this, this is rarely ever done in practice. In a VAV system, the entire system does not need to be rebalanced every time the system is modified. The changes only need to take place at the zone level. The pressure independent valves on the exhaust and supply automatically accommodate the changes in the duct mains due to the remodel. Typical VAV zone dampers have a control range from several tenths of an inch of water column to greater than 10 inches of water column.

Energy recovery systems have no non-energy benefits.

1.6 Environmental Impact

There are no significant potential adverse environmental impacts of this measure.

1.7 Technology Measures

The VAV measure encourages the use of fast-acting air valves on both supply and exhaust. There are already several large manufacturers who make and sell these valves, including Phoenix Controls, TSI, Siemens, Triatek and Tek-Air.

The energy recovery measure uses off the shelf components that are widely available: coils, pipe, flow control valves, dampers, pump and piping appurtenances.

1.8 Performance Verification of the Proposed Measure

Both proposed measures require startup and commissioning.

1.9 Cost Effectiveness

A summary of the cost-effectiveness is given in Table 3 below. The prototype building isdescribed below in Section 2.2. For details on the results, see Section 3.1.3.

a	b	c	d	e	f	g

Measure Name	Measure Life (Years)	Additional Costs[1]– Current Measure Costs (Relative to Basecase) ($)		Additional Cost[2]– Post-Adoption Measure Costs (Relative to Basecase) ($)		PV ofAdditional[3] Maintenance Costs (Savings) (Relative to Basecase) (PV$)		PV of[4] Energy Cost Savings – Per Proto Building (PV$)	LCC Per Prototype Building ($)	
		Per design cfm	Per Proto Building	Per design cfm	Per Proto Building	Per design cfm	Per Proto Building		(c+e)-f Based on Current Costs	(d+e)-f Based on Post-Adoption Costs
VAV, 6 ACH, CZ03	15	$14.27	$1,423,608	$14.27	$1,423,608	$0.00	$0	$3,829,373	-$2,405,765	-$2,405,765
VAV, 12 ACH, CZ03	15	$14.27	$1,423,608	$14.27	$1,423,608	$0.00	$0	$2,493,770	-$1,070,162	-$1,070,162
VAV, 6 ACH, CZ12	15	$14.27	$1,423,608	$14.27	$1,423,608	$0.00	$0	$4,338,856	-$2,915,248	-$2,915,248
VAV, 12 ACH, CZ12	15	$14.27	$1,423,608	$14.27	$1,423,608	$0.00	$0	$2,866,318	-$1,442,710	-$1,442,710

Table 3. Cost effectiveness of VAV

1.10 Analysis Tools

Variable air volume and energy recovery can both be modeled in eQuest.

1.11 Relationship to Other Measures

The analysis and results of this measure also support ASHRAE 3: ERV for high minimum outside air.

2 Methodology

The energy savings for VAV and energy recovery were calculated using energy simulations. A calibrated energy model of an actual lab at Stanford University was used for the analysis. The model was originally built in order to decide what energy conservation measures would be implemented during a retrofit and to estimate the energy cost savings. The model was calibrated using three years of utility data. The original building had a constant volume reheat system, and was retrofitted to a variable-air volume reheat system.

2.1 Baseline model details

2.1.1 Building Description

The building contains 104,000 gross square feet of laboratory and office space. It is a 5-story building with a basement. The basement contains the mechanical equipment and serves as a living area for the research animals. The first floor has office and administrative spaces. The 2nd through 5th floors house offices and laboratory spaces.

2.1.2 Mechanical system

The mechanical system has five air handlers, which includes one main air handler that serves the majority of the spaces in the building, which are a mix of lab and office-type spaces. For the purpose of this analysis, the labs and office-type spaces were separated onto separate air handlers. The system serving the main labs is the only system taken into account for this analysis. There were no changes made during the analysis to the remaining systems.

In the baseline design the air handler serving the main labs is constant volume and runs on 100% outside air. The air handler runs 24/7. The air handler contains a pre-heat coil, chilled water coil, and hot water coil. In the actual lab, the chilled water and hot water are provided from the campus distribution system. However, for the purposes of this analysis, a plant to serve the building was made. The plant consisted of 2 identical water-cooled chillers and one 2-cell cooling tower. The design heating supply air temperature is 95°F. The design cooling supply air temperature is 55°F, and can be reset up to 65°F.

There are 16 total exhaust fans, out of which there are five main ones which are the ones that exhaust the spaces supplied by the main air handler. These five exhaust fans are the only ones accounted for in the analysis.

2.1.3 Zones

The space served by the main lab air handler contains 21 zones, totaling 34,000 square feet. The design peak airflows in these zones range from 6 to 18 ACH. The zones all have constant volume boxes with reheat coils. The labs are a mix of interior and perimeter zones. The labs have peak occupant densities between 100 and 200 square foot per person, peak lighting density of 1.5 watts per square foot, and peak plug loads ranging between 2 and 4 watts per square foot. The equipment rooms have 5 to 30 w/ft2.

2.2 Variable air volume energy model

In the baseline lab building model, the system is constant volume reheat. For this measure, a variable air volume system was modeled. Seven different cases were run on this building in each climate zone. In each case the lab minimum air change rate was set at 6 ACH. The lower the minimum air change rate, the higher the potential savings are for a variable air volume system. See section 3.1 for results.

2.3 Energy recovery energy model

In the baseline lab building model, the system is constant volume reheat with no energy recovery as described above. For this measure, energy recovery was modeled in a constant volume lab building as well as in a variable air volume lab building. The measure was also run at high and low air change rates. Higher air change rates have more energy leaving the lab that can be recovered, so savings will generally be higher. Also, constant volume systems have more energy leaving the lab that can be recovered, so savings will generally be higher than variable air volume systems.

Certain types of energy recovery, such as enthalpy wheels, have significant design challenges which may not make it possible to implement on every building. Runaround coils, however, can be added to almost any design without significant changes to the rest of the design. Two extreme cases of energy recovery were modeled, one in which small coils were selected to have a low effectiveness and one in which large coils were selected to have a high effectiveness. In both cases, the length by width dimensions of the coils were based on having a maximum coil face velocity of 500 ft/min. The fins/inch were specified on all coils to be 10, as that is the maximum allowable in Standard 62 that will allow for the most heat transfer. It is assumed that the air handler already has a 2-row preheat coil which can be used for energy recovery. The entering exhaust air temperature was assumed to be 75°F, and the entering outside air temperature was assumed to be the ASHRAE median of extremes winter design temperature, which was 32°F for CTZ 3 and 30°F for CTZ 12.

In one case, the goal was to get the maximum feasible effectiveness. In this case, both the supply side and exhaust side coils were selected to be the largest that they would feasibly be, that is 8-rows each.In a second case, a 2-row coil was selected for the supply side and a 4-row coil was selected for the exhaust side. This coil selection represents the lower threshold of effectiveness.In this case the 2-row coil on the supply side does not add any extra pressure drop in the system since the coil serves as a pre-heat coil, which would already be present whether or not there was energy recovery. The 8-row coil for the high effectiveness case has a total pressure drop of 0.75”, but is discounted by 0.19” to factor in a pre-heat coil that would otherwise be there. The water flowrate through the coil was determined based on existing runaround coil designs. It was determined that the average cfm/gpm was 300 and that the average water ΔT was 14.6°F. See Table 4 for details of the coil selections.The coil selection was made for a design airflow of 10,000 cfm, but the selection can be scaled to fit any design.

	Case 1		Case 2	
	Supply	Exhaust	Supply	Exhaust
Rows	8	8	2	4
Fins/inch	10	10	10	10
GPM	35	35	35	35
CFM	10,000	10,000	10,000	10,000
EDB (°F)	30	75	30	75
LDB (°F)	55.1	53.4	42.5	62.5
EWT (°F)	62.0	45.7	62.9	54.1
LWT (°F)	46.5	60.9	55.2	61.9
Airside dP (in H20)	0.75	0.86	0.19	0.37
Waterside dP (ft H20)	6.2	5.2	2	2.9
Effectiveness	0.48		0.28	

Table 4.Energy recovery coil selections

The effectiveness is equal to the difference between entering exhaust air temperature and leaving exhaust air temperature divided by the difference between entering exhaust air temperature and entering outside air temperature. It is noted above that even for coils as large as 8-rows, the effectiveness is still only 0.48.

Though the design outdoor air temperatures for many of these climates is below freezing, glycol was not modeled. This is a realistic assumption, as glycol generally reduces heat transfer and increases pumping head. Instead, in good practice, the sequences of operation would activate the system when the outdoor air temperature gets below a certain threshold and the system is in danger of freezing.

The following table gives all of the parameters entered into eQuest to represent the designabove.

	Input	Notes
Basic Specifications		
ERV Device Typ	Sensible HX	Runaround coils
HX Performance		
HX Configuration	Cross flow	
Effectiveness, sensible	0.28/0.48	0.28 for low-effectiveness case, 0.48 for high-effectiveness case
HX Air Film Resist, sensible	0.4	eQuest default
Air Film Resist Exp, sensible	0.2	eQuest default

Control Sequences		
Operation	OA Exhaust DT	Operates when HVAC fans are on and temperature diff btwn outdoor air temperature and exhaust air temperature is above the specified delta T.
Outside/Exhaust air delta T	8°F	
Operating mode	OA Heat/Cool	Recovery in heaing and cooling. Operates whenever the absolute temperature difference btween outdoor air temp and exhaust air temp is above the specified delta T.
Make-up Air Temp Control	Mixed Air Reset	
Capacity Control	Bypass OA	Outdoor air will be bypassed as required as to not overheat/overcool.
ERV Power		
HX Power	1.7 kw	(variable)
ERV Fans	HVAC Supply/Return	Pressure drops through the coils are added to the system fans.
ERV Fan Efficiency	0.6	eQuest default
Fan Motor Efficiency	Standard	
Delta P at Design flow, Make-up	0.19”/0.75”	0.19” for low-effectiveness case, 0.75” for high-effectiveness case
Delta P at Design flow, Exhaust	0.37”/0.86”	0.37” for low-effectiveness case, 0.86” for high-effectiveness case Calculated outside of eQuest

Table 5.Energy recovery details

Exhaust fan energy was calculated outside of eQuest.

3 Analysis and Results

An energy model was used for this measure. The energy model originated from an actual job of a lab at Stanford University. See Section 2 above for details of the model.

3.1 Variable air volume

3.1.1 Energy savings

Energy savings estimates for actual case studies are presented in Table 6 below. The energy savings per cfm and per square foot vary considerably by lab building, as discussed above. Savings vary considerably based on climate, minimum air change rate, and envelope and internal loads.

Source	Area (sqft)	Airflow (cfm)	Annual Energy Savings		
			Total	Per cfm	Per sqft
New Construction					
Labs 21 Case Study	71,347	71,347	$92,120	$1.29	$1.29
Retrofits					
Stanford Beckman	182,000	325,535	$987,001	$3.03	$5.42
Stanford Stauffer I	28,000	38,380	$110,258	$2.87	$3.94
Stanford Gilbert	75,000	134,000	$836,855	$6.25	$11.16

Table 6. Energy savings estimates from case studies

Energy savings were calculated using the methodology above. The results are presented in terms of kwh per design cfm and therms per design cfm in

	6 ACH		10 ACH		14 ACH	
Climate Zone	kwh/cfm	therms/cfm	kwh/cfm	therms/cfm	kwh/cfm	therms/cfm
3	14.52	0.68	12.27	0.42	7.85	0.23
4	15.38	0.64	13.08	0.41	8.36	0.23
6	12.60	0.88	10.30	0.72	5.55	0.60
7	12.70	0.98	10.44	0.87	5.70	0.78
8	11.84	0.92	9.50	0.79	4.46	0.68
9	16.91	0.42	14.51	0.27	9.32	0.15
12	16.18	0.70	13.80	0.45	8.82	0.25
13	10.55	0.70	8.13	0.48	2.83	0.30

Table 7 below for the 8 climate zones where the majority of the new construction is expected to happen. As expected, the savings are highest when the minimum air change rate is the lowest. Savings decrease as the minimum air change rate increases.

	6 ACH		10 ACH		14 ACH	
Climate Zone	kwh/cfm	therms/cfm	kwh/cfm	therms/cfm	kwh/cfm	therms/cfm
3	14.52	0.68	12.27	0.42	7.85	0.23
4	15.38	0.64	13.08	0.41	8.36	0.23
6	12.60	0.88	10.30	0.72	5.55	0.60
7	12.70	0.98	10.44	0.87	5.70	0.78
8	11.84	0.92	9.50	0.79	4.46	0.68
9	16.91	0.42	14.51	0.27	9.32	0.15
12	16.18	0.70	13.80	0.45	8.82	0.25
13	10.55	0.70	8.13	0.48	2.83	0.30

Table 7. Annual energy savings of variable air volume systems compared to constant volume systems per design cfm

The HVAC TDV savings are presented in Figure 1 and Figure 2for climate zones 3 and 12, respectively. The savings are broken out by equipment type, including fans, pumps, space cooling, and space heating. The savings by equipment type were estimated based on the percentage of each end use of the whole building annual energy use. As expected, the savings decrease as the minimum ACH rate is increased. The savings profiles seen in these two figures are typical of all climate zones.

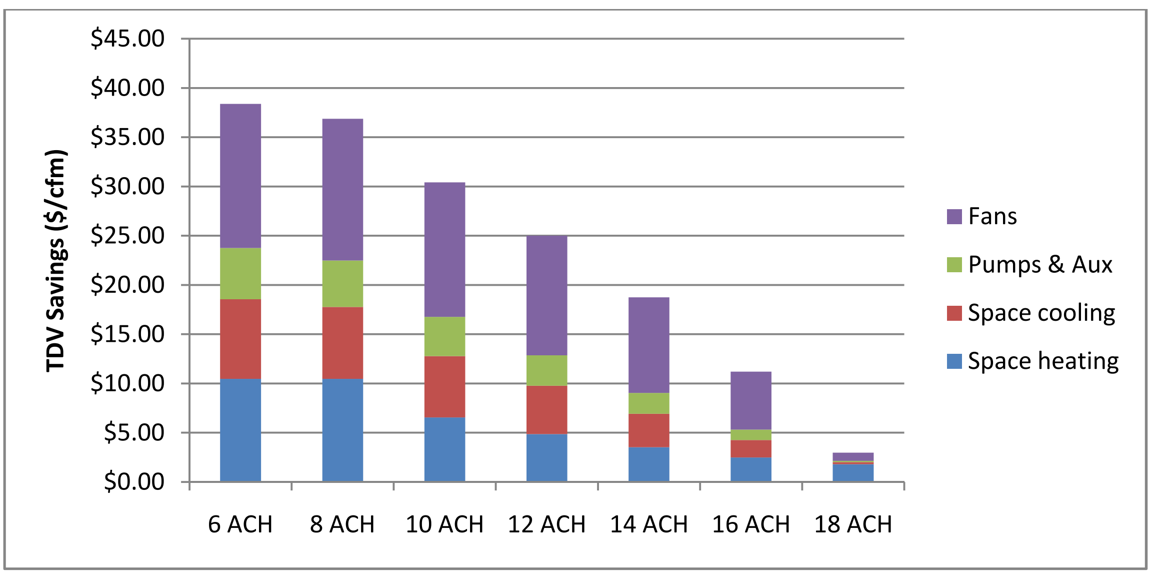


Figure 1. TDV Savings in Climate Zone 3

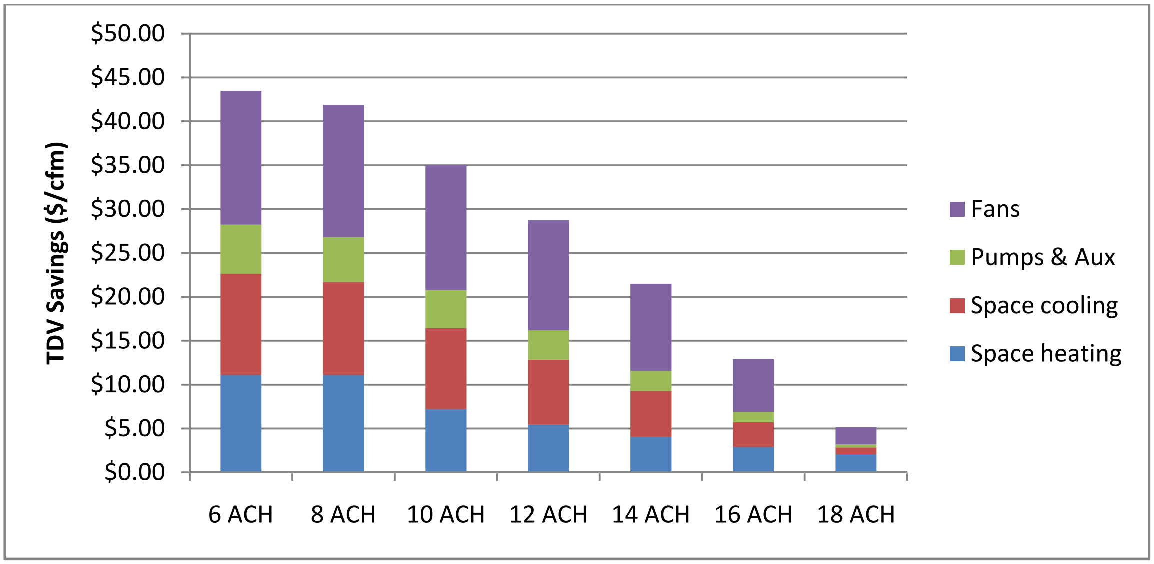


Figure 2. TDV Savings in Climate Zone 12

3.1.2 Costs

The main incremental cost of a variable air volume system over a constant air volume system in lab buildings is the cost of air valves that modulate supply, general exhaust, and fume exhaust air as conditions in the lab change, either due to changes in load or due to the opening or closing of a hood sash. Along with the cost of the valves comes the additional cost of controls and commissioning.

Determining the actual incremental cost of a VAV lab over a constant volume lab is difficult and has many variables. Instead of projecting what this cost would be for new construction, lab retrofit projects where a lab building was converted from VAV to constant volume were studied. Retrofits of this nature are significantly more expensive than the change would be in new construction, and therefore provide an extreme upper bound to the cost.

Four different lab buildings on a university campus were used as case studies for determining the incremental cost of going VAV. Each of the four labs was built between 19xx and 19xx, and underwent a major retrofit between 200x and 200x to implement energy-saving and maintenance measures. Each of the lab buildings was previously constant volume reheat, and was converted to VAV as part of the retrofit. The retrofit included other energy-savings measures as well, but the other measures were small in terms of cost and energy savings compared to VAV.

Cost data from each of these four lab retrofits are shown in Figure 3below. The blue markers indicate the total cost of the retrofit of each of the four buildings, and range from $15- $25 per design cfm. The costs listed are contractor costs from real bids, and include materials, installation, controls, and all other associated costs with the retrofit. These costs include the cost of converting the building to VAV, as well as other energy-saving measures. As expected, the cost in dollars per cfm goes down as the design cfm increases. For one of the lab buildings, a breakdown of the costs was studied. For this lab, bids from four different mechanical contractors were taken and the costs associated with just converting the system to VAV was broken out. This cost data is represented in the figure below with the red markers, and ranges from $10 - $18 per cfm with an average of ~$14.00.

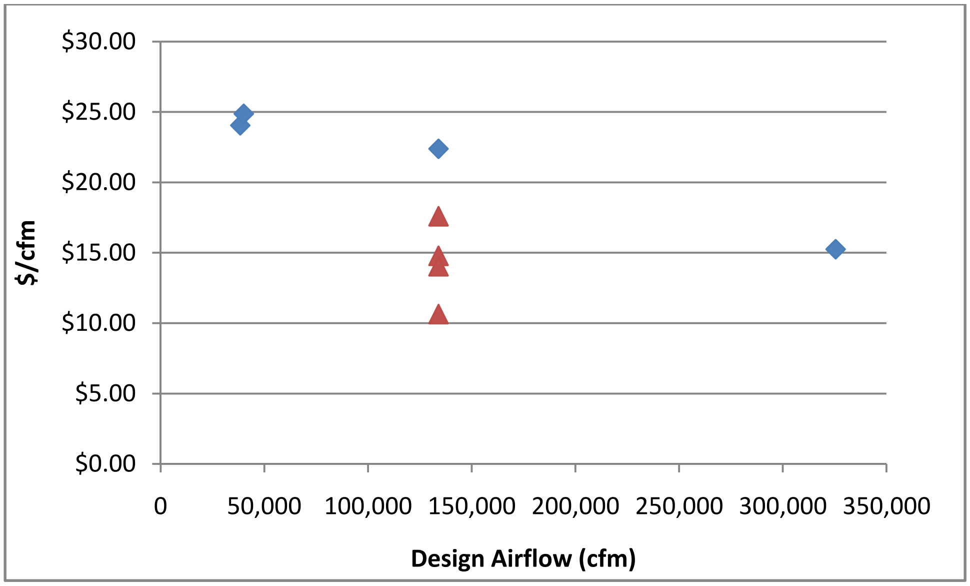


Figure 3. Cost data from four major lab retrofits

A case study done by Labs 21 determined that the incremental cost of implementing a VAV system instead of a constant volume system for new construction is roughly $4.20 per cfm(Laboratories for the 21st Century: National Renewable Energy Laboratory, Science and Technology Facility). The details of the assumptions that went into this cost estimate are not known, so this number was not used for the analysis, but simply presented as a benchmark.

3.1.3 Life-cycle cost calculations

The life-cycle cost of labs with constant volume and variable volume systems was calculated for each climate zone.The incremental first-cost of a VAV system is assumed to be the average of the VAV retrofit bids described above. The assumed incremental cost does not vary by climate zone or by lab size.

The results are presented in Figure 4 and Table 8 for different minimum air change rates. The life-cycle cost of a constant volume system is assumed to be $0. From the figure and table, it is clear that VAV systems have a lower life-cycle cost in all cases across all climate zones, except in climate zone 13 with a minimum air change rate of 14 ACH.

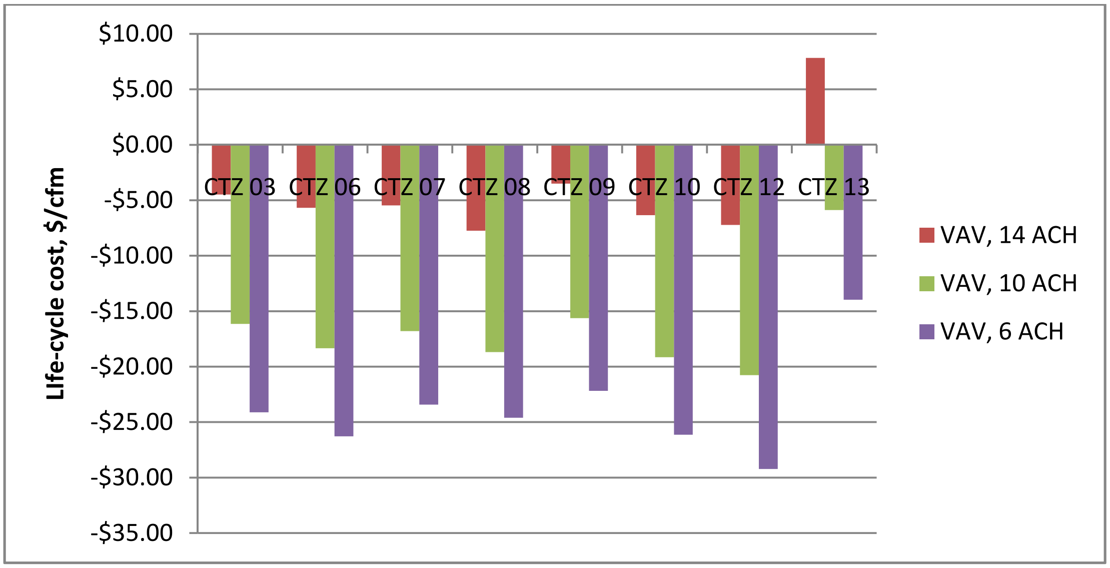


Figure 4. Life-cycle cost

			6ACH		14ACH	
Climate Zone	City	Incremental cost ($/cfm)	PV of energy cost savings	LCC of VAV	PV of energy cost savings	LCC of VAV
3	Oakland	$14.27	-$38.38	-$24.11	-$18.75	-$4.48
6	Torrance	$14.27	-$40.55	-$26.28	-$19.95	-$5.68
7	San Diego	$14.27	-$37.69	-$23.42	-$19.74	-$5.47
8	Fullerton	$14.27	-$38.88	-$24.61	-$22.01	-$7.74
9	Los Angeles	$14.27	-$36.46	-$22.19	-$17.77	-$3.50
10	Riverside	$14.27	-$40.41	-$26.14	-$20.61	-$6.34
12	Sacramento	$14.27	-$43.49	-$29.22	-$21.50	-$7.23
13	Fresno	$14.27	-$28.24	-$13.97	-$6.43	$7.84

Table 8. Life-cycle cost

3.2 Energy recovery

3.2.1 Energy savings

Energy savings estimates for actual case studies are presented in Table 9 below. The energy savings per cfm and per square foot vary considerably by climate zone and energy recovery design.

		Airflow (CFM)	Annual Energy Savings ($/cfm)
Labs 21 Case Study	NREL, Science and Technology Facility, Golden, CO	71,347	$0.51
Labs 21 Best Practices	City: Minneapolis	1	$0.91
Labs 21 Best Practices	City: Denver	1	$0.52
Labs 21 Best Practices	City: Seattle	1	$0.41
Labs 21 Best Practices	City: Atlanta	1	$0.32
Konvekta	Gilbert Hall sample quote	80,400	$0.71
KJWW	Rock Valley College, Northern IL	55,000	$0.06
KJWW	Wheaton College, Northern IL	105,000	$0.03
KJWW	Joliet Junior College, Northern IL	52,500	$0.22
Paul DuPont	OSU Linus Pauling in Corvallis, OR	180,000	$0.02

Table 9. Energy savings estimates from other studies

The energy model described above in Section 2.3 was run in multiple climate zones, multiple air change rates, multiple energy recovery effectivenesses, and for constant volume and variable volume. The results of these runs are presented below in

			Climate Zone 3		Climate Zone 12	
Airflow	ACH	Energy recovery effectiveness	kwh/cfm	therm/cfm	kwh/cfm	therm/cfm
CV	10	0.3	-0.942	0.095	-0.551	0.155
CV	10	0.5	-2.449	0.131	-1.976	0.209
VAV	10	0.3	-0.482	0.103	-0.141	0.156
VAV	10	0.5	-0.633	0.116	-0.164	0.181
CV	18	0.3	-1.610	0.159	-0.803	0.257
CV	18	0.5	-4.379	0.226	-3.272	0.351
VAV	18	0.3	-0.472	0.128	0.022	0.194
VAV	18	0.5	-0.671	0.133	0.005	0.203

Table 10. It is clear from the table that in most cases the building actually uses more electricity with energy recovery than without it. This is because of the increase in fan energy due to the pressure drop of the energy recovery coils. Though there is some electricity savings in cooling energy, in most cases it is not enough to make up for the increased fan energy in terms of kilowatt-hours. However, on a TDV rate for electricity, the energy recovery actually saves

money. This is because during peak cooling times, when TDV rates are the highest, energy recovery pre-cools the outdoor air, therefore requiring less chiller and pump energy. In all cases gas energy used for space heating is saved with energy recovery.

			Climate Zone 3		Climate Zone 12	
Airflow	ACH	Energy recovery effectiveness	kwh/cfm	therm/cfm	kwh/cfm	therm/cfm
CV	10	0.3	-0.942	0.095	-0.551	0.155
CV	10	0.5	-2.449	0.131	-1.976	0.209
VAV	10	0.3	-0.482	0.103	-0.141	0.156
VAV	10	0.5	-0.633	0.116	-0.164	0.181
CV	18	0.3	-1.610	0.159	-0.803	0.257
CV	18	0.5	-4.379	0.226	-3.272	0.351
VAV	18	0.3	-0.472	0.128	0.022	0.194
VAV	18	0.5	-0.671	0.133	0.005	0.203

Table 10. Energy savings results

Figure 5 through Figure 12 show the TDV energy savings by equipment type and the total energy savings in climate zones 3, 9, and 12. The TDV energy savings by equipment type were estimated based on the percentage of each end use of the whole building annual energy use. From these figures it is clear that fan energy costs a significant portion of the savings, and that the majority of the savings come from space heating (gas) and space cooling (electricity). The proportions and magnitudes of these savings vary considerably by climate zone, by minimum air change rate, and by energy recovery effectiveness. In climate zone 3, the TDV savings range from -\$2.00/cfm to \$1.00/cfm. In climate zone 12, the TDV savings range from -\$1.50/cfm to \$4.10/cfm.

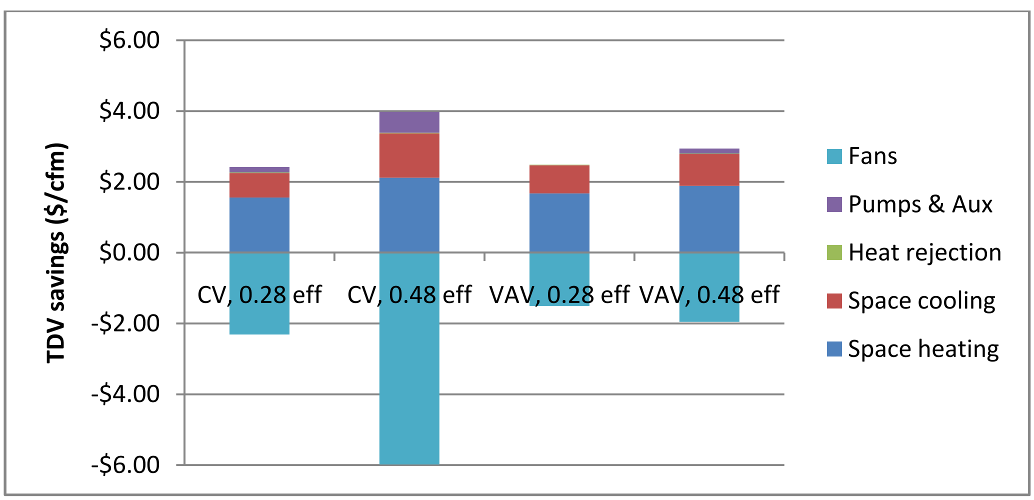


Figure 5. 15-year energy savings by end-use, CTZ 3, 10 ACH

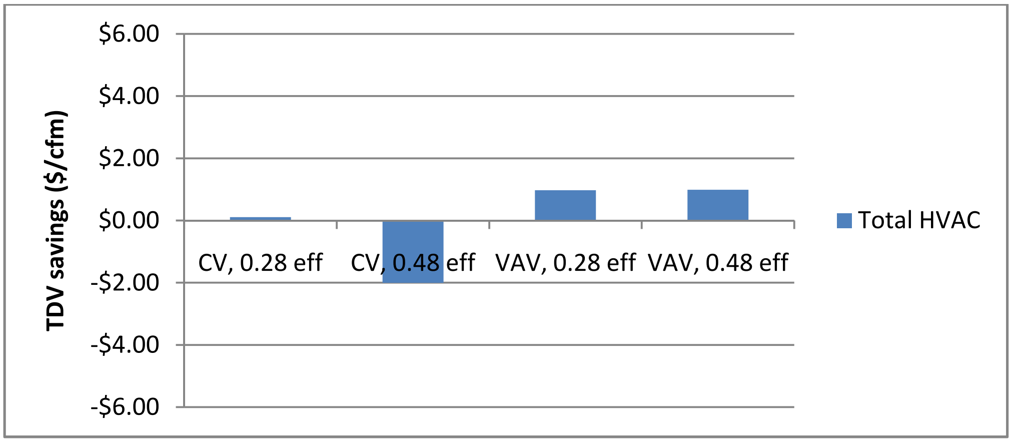


Figure 6. Total 15-year energy savings, CTZ 3, 10 ACH

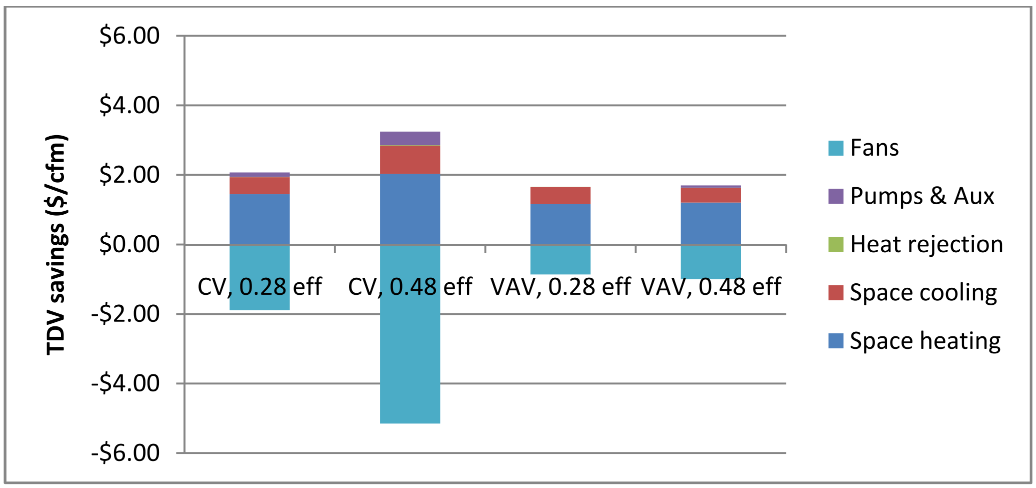


Figure 7.15-year energy savings by end-use, CTZ 3, 18 ACH

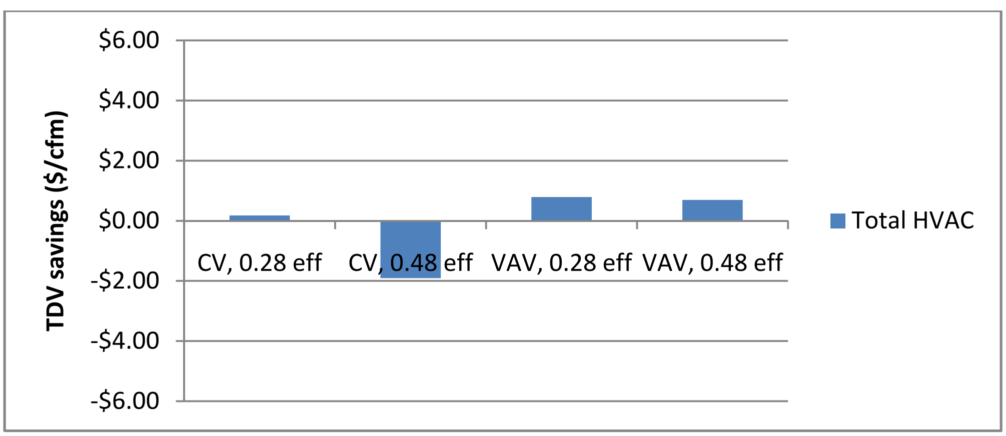


Figure 8. Total 15-year energy savings, CTZ 3, 18 ACH

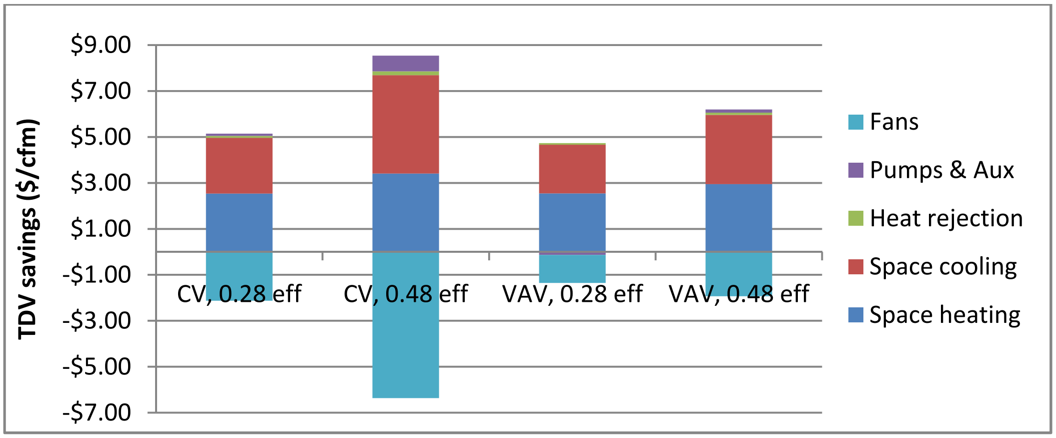


Figure 9.15-year energy savings by end-use, CTZ 12, 10 ACH

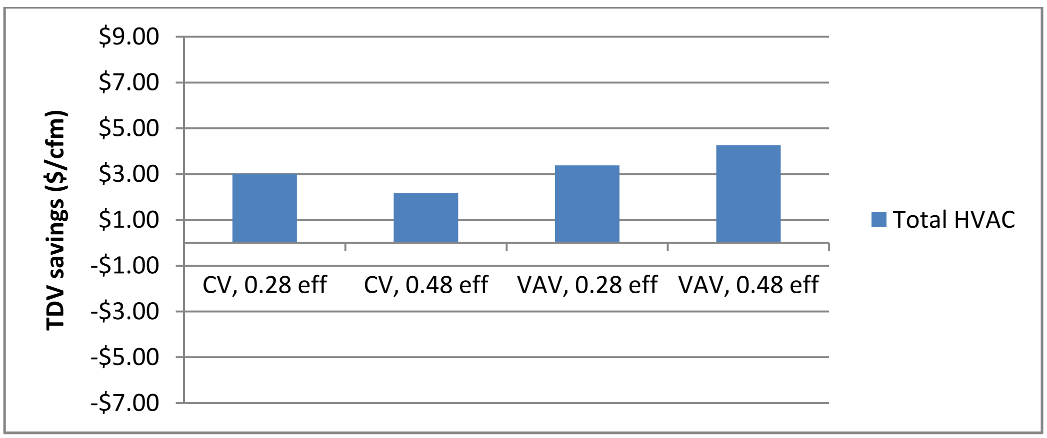


Figure 10.Total 15-year energy savings, CTZ 12, 10 ACH

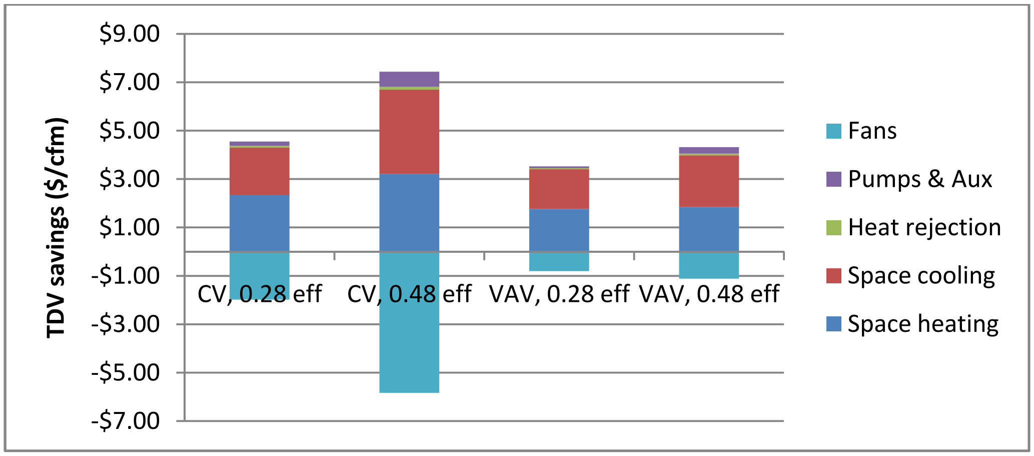


Figure 11.15-year energy savings by end-use, CTZ 12, 18 ACH

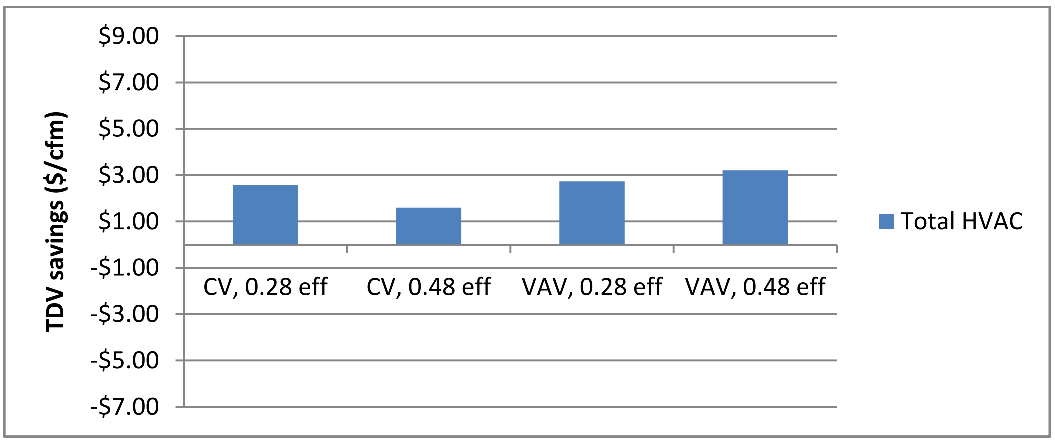


Figure 12.Total 15-year energy savings, CTZ 12, 18 ACH

A summary of the total TDV energy cost savings across several climate zones is given in Table 11.

	10 ACH				18 ACH			
	CV		VAV		CV		VAV	
	Eff = 0.30	Eff = 0.50	Eff = 0.30	Eff = 0.50	Eff = 0.30	Eff = 0.50	Eff = 0.30	Eff = 0.50
CTZ03	-$0.11	$2.00	-$0.98	-$0.99	-$0.18	$1.91	-$0.79	-$0.70
CTZ08	$0.41	$2.69	-$0.56	-$0.68	-$0.02	$1.98	-$0.60	-$0.69
CTZ09	-$1.02	$0.48	-$1.61	-$2.16	-$0.74	$0.84	-$1.42	-$1.82
CTZ12	-$3.02	-$2.17	-$3.38	-$4.26	-$2.57	-$1.60	-$2.73	-$3.20

Table 11. TDV energy cost savings of energy recovery ($/cfm)

3.2.2 Costs

From the case studies on energy recovery collected, the price data that was available was compiled and is presented in Table 12. In both cases presented, the cost of energy recovery is ~$1.15 per square foot, or between $0.65 and $1.12 per cfm.

			Incremental Cost		
	Area (sqft)	Airflow (CFM)	Total	Per cfm	Per sqft
NREL, Science and Technology Facility, Golden, CO	71,347	71,347	$80,000	$1.12	$1.12
OSU Linus Pauling in Corvallis, OR	100,000	180,000	$116,250	$0.65	$1.16

Table 12.Energy recovery costs received from projects

Cost estimates were received for individual components of aenergy recovery system and are presented below in Table 13. The costs include coils, piping, and pumps.

	$/cfm
Coils (low eff)	$0.40
Coil (high eff)	$1.01
Pumps	$0.12
Piping	$0.62
Total (high eff)	**$1.75**
Total (low eff)	**$1.14**

Table 13.Energy recovery costs

3.2.3 Life-cycle cost calculations

The life-cycle cost of the energy recovery was calculated for multiple climate zones. A summary of the results is given in the table below. In the basecase with no energy recovery, the life-cycle cost is assumed to be $0. From the table it is clear that whether or not energy recovery is cost-effective is largely a function of the climate zone. Energy recovery is never cost-effective in climate zones 3 or 8, but is almost always cost-effective in climate zone 12.

	10 ACH				18 ACH			
	CV		VAV		CV		VAV	
	Eff = 0.30	Eff = 0.50	Eff = 0.30	Eff = 0.50	Eff = 0.30	Eff = 0.50	Eff = 0.30	Eff = 0.50
CTZ03	$1.03	$3.75	$0.16	$0.76	$0.96	$3.66	$0.35	$1.05
CTZ08	$1.55	$4.44	$0.58	$1.07	$1.12	$3.73	$0.54	$1.06
CTZ09	$0.12	$2.23	-$0.47	-$0.41	$0.40	$2.59	-$0.28	-$0.07
CTZ12	-$1.88	-$0.42	-$2.24	-$2.51	-$1.43	$0.15	-$1.59	-$1.45

Table 14.Life-cycle cost of energy recovery ($/cfm)

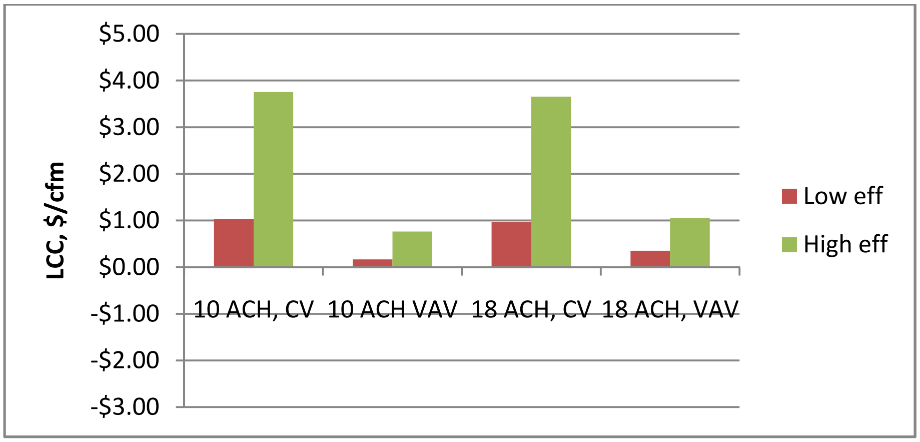


Figure 13. 15-year life-cycle cost, CTZ 3

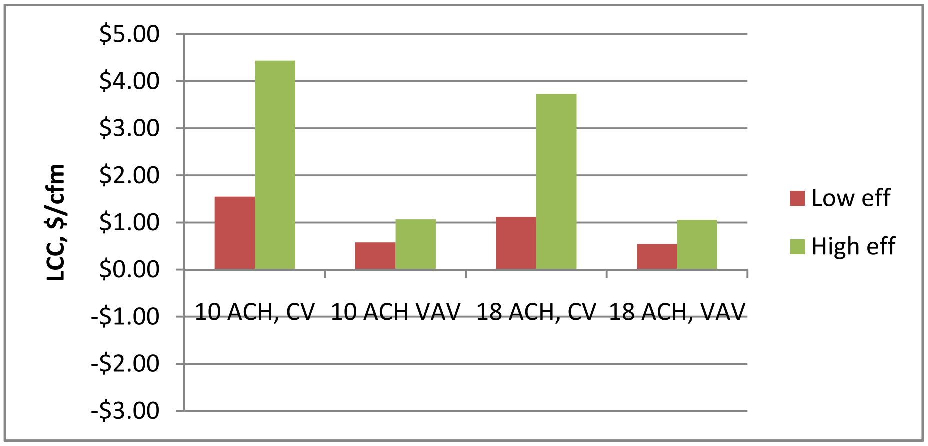


Figure 14. 15-year life-cycle cost, CZ 8

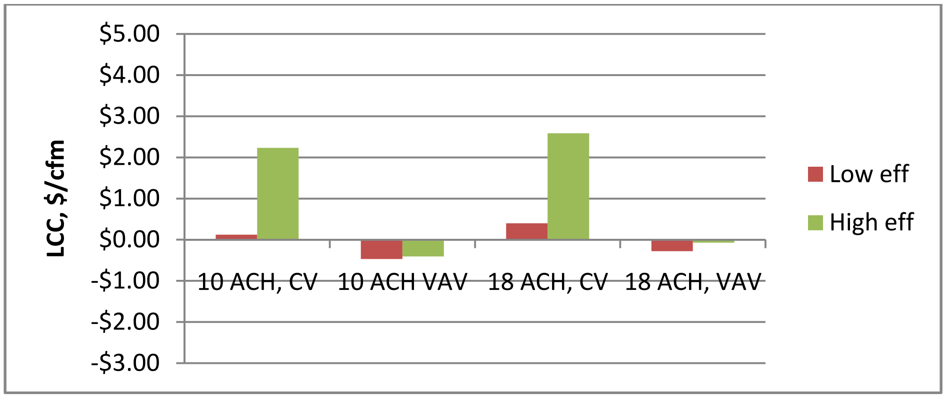


Figure 15. 15-year life-cycle cost, CZ 9

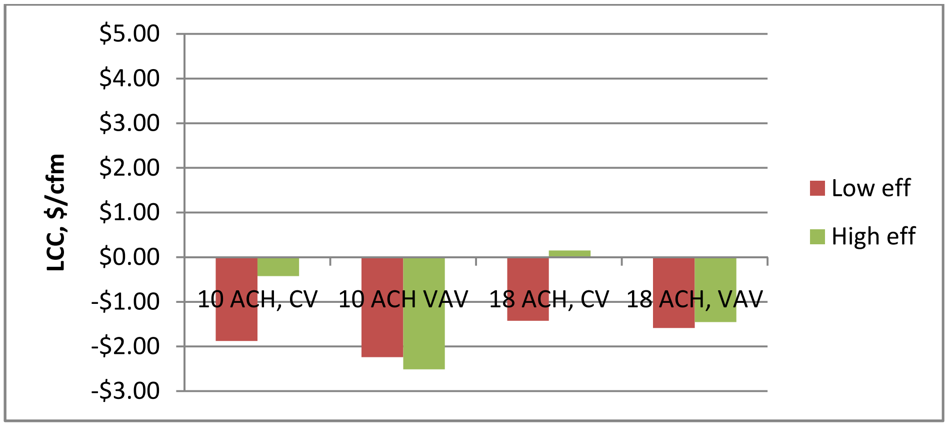


Figure 16. 15-year life-cycle cost, CTZ 12

3.2.4 Reach code analysis

The reach code analysis differs in that energy is valued higher than in the analysis done for the Standard. As seen in Section 3.2.1, energy recovery always saves heating and cooling energy, but always uses more fan energy. Because the reach code multipliers apply to both increases and decreases in energy use, in some cases the energy cost savings increased and in some cases it decreased compared to the Standard calculations.

The life-cycle cost of energy recovery using the reach code TDV multipliers was calculated for multiple climate zones. A summary of the results is given in the table below. In climate zone 12, where energy recovery is cost-effective by a wide margin according to the Standard calculations, the measure has an even lower life-cycle cost with the reach code multipliers. In the remaining climate zones, the measure still has a higher life-cycle cost than the basecase the majority of the time. See a summary of the results in Table 15.

	10 ACH				18 ACH			
	CV		VAV		CV		VAV	
	Eff = 0.30	Eff = 0.50	Eff = 0.30	Eff = 0.50	Eff = 0.30	Eff = 0.50	Eff = 0.30	Eff = 0.50
CTZ03	$0.81	$4.00	-$0.29	$0.28	$0.74	$3.89	$0.01	$0.73
CTZ08	$1.57	$5.01	$0.35	$0.80	$1.04	$4.12	$0.33	$0.82
CTZ09	-$0.25	$2.20	-$1.00	-$1.09	$0.10	$2.65	-$0.73	-$0.62
CTZ12	-$2.95	-$1.39	-$3.40	-$3.95	-$2.36	-$0.65	-$2.49	-$2.49

Table 15.Life-cycle cost of energy recovery using reach code multipliers ($/cfm)

3.3 Typical practice

Typically the minimum ventilation rate in labs is between 6 and 12 air changes per hour (ACH), and can even get above 20 ACH. Labs are often 100% outside air, constant volume reheat systems. As mandated by code, the stack velocity of the exhaust is typically 3,000 feet per minute, and the supply and exhaust fans typically run at 4 to 6 inches of pressure. Fume hoods are typically constant volume. Supply air temperature reset may or may not be used.

3.4 Safety

3.4.1 Laboratory VAV Controls

As described in Section 1.5 Non-Energy Benefits, VAV controls have many safety benefits due to the application of pressure independent air valves and the ability to track and alarm hood face velocity and room air balance.During one of our stakeholder meetings with personnel from CalOSHA and ARB they raised concerns about the speed of response for the zone controls. We surveyed three of the major manufacturers TSI, Siemens and Phoenix. Here are the responses that we have received to date:

3.4.1.1 TSI Response: Dan Schuster of Bayside Mechanical

The TSI gear responds very quickly.

- The end to end response is about 3 seconds
- The controllers responds to a change in input in 100mS (0.1 Second) and the controlled devices (actuators) go full stroke in 1.5 seconds.
- A fume hood exhaust damper actuator travel from about 1/4 damper position (sash closed) to ¾ damper position (sash full open).
- It takes a typical user about 1-2 seconds to fully open or close a hood.
- So the control system is as fast or faster than the user.
- Room pressure controllers have a similarly fast operation.

3.4.1.2 Phoenix Response: Rich Yardley of Newmatic Engineering

The Phoenix system has a total response time of about 0.60 seconds, and that even includes the air transport delay (the time it takes for the air to start moving after you open a damper).

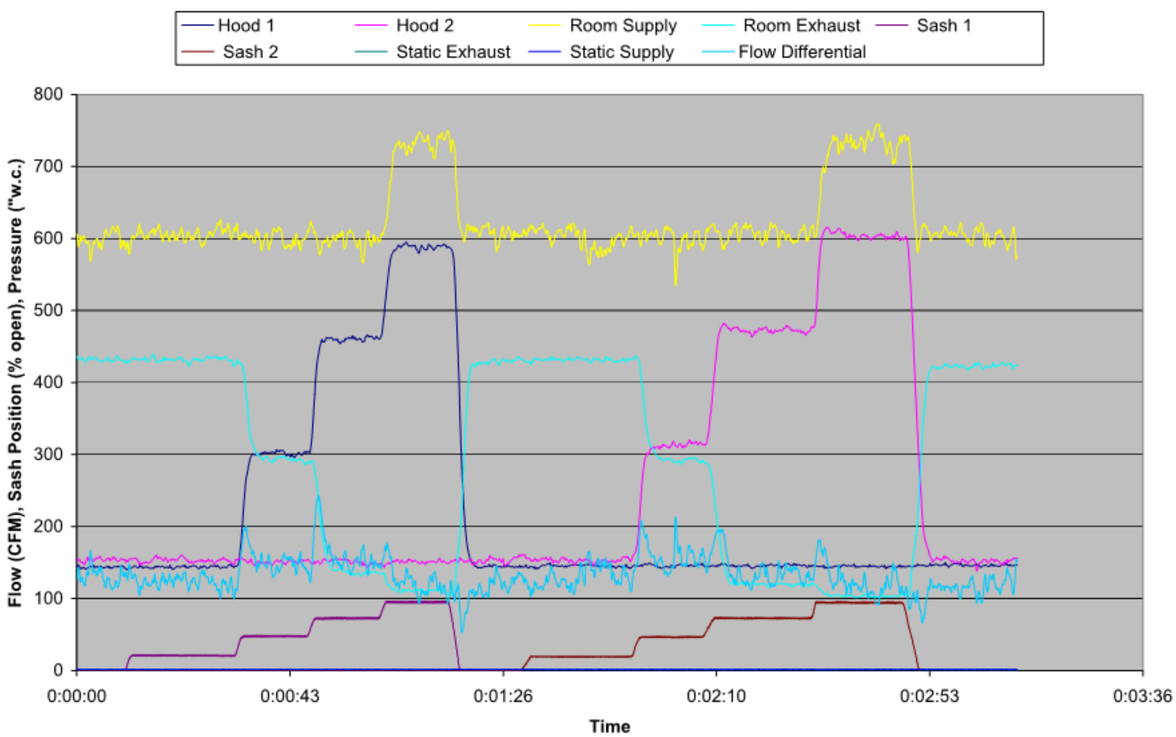
Many systems promise performance, but the reality is that they can't react that fast. A system's total response time needs to take multiple factors into account:

1. Hood sensing technology - Phoenix Controls has ALWAYS used sash sensing. As soon as a sash starts opening (whether horizontally, vertically, or a combination thereof), our system starts to respond. Immediately.
2. Systems that use sidewall sensing (sometimes erroneously referred to as face velocity sensing) are guaranteed to fail a speed test: they don't begin to respond until the system has already started to fail.
3. Valve actuation - Of course you need a high-speed actuator. It needn't travel full stroke in 1 second, but it needs to respond fast enough to accommodate a "worst case" scenario

(e.g., raising a vertical sash from minimum to maximum at a rate of 1.5 fps). That usually equates to 2 to 3 seconds for full stroke.

4. Pressure independence - Pressure changes occur both locally and at adjacent and downstream devices. The pressure independence of the system must respond in a fraction of a second (not normal in HVAC controls ... most respond in 60 to 90 seconds to maintain system stability).
5. Overall control system response - There are a lot of "moving parts" in a control system. All components must by synchronized to provide the proper response time with stability. This is easier said than done. Phoenix Controls does it so well that it seems easy. The only way I've seen "others" approach our speed of response with stability is by limiting the hood turndown to a very narrow range (2:1 or less).
6. "Other" devices - as you stated, the supply valve and general exhaust valve must be similarly responsive to the fume hood valve. Otherwise room pressure relationships will be compromised, and in some cases, fume hood containment will be compromised (when the room is "starved" of air).

Below is data from a 3rd party field test of a Phoenix system installed at the University of Cincinatti from March of 2005.



3.4.1.3 Siemens Response: Jim Coogan of Siemens

Below is a test report from the University of Alaska in Fairbanks on side by side labs with Phoenix and Siemens valves and controls.

On April 22, 2004, Ben Larue and Jeff Waters of Siemens Building Technologies conducted tests on the airflow in O'neil Building Laboratories 218 and 236. The results of the ASHRAE-110 face velocity and smoke tests on hoods 218N, 218S, and 236M are attached. The balancer's report on room airflow will be sent separately.

In summary, the two Siemens-controlled fume hoods in Lab 218, and the Phoenix-controlled fume hood in room 236 performed adequately and contained smoke at the design sash height of 18 inches.

The average of five ASHRAE-110 time response tests on the Siemens fume hoods was 2.2 seconds to the stop of sash movement, and 3.6 seconds to within 10% of setpoint. The Phoenix hood averaged 1.8 seconds to the stop of sash movement (controlled by the operator) and 3.0 seconds to within 10% of setpoint.

The most significant factor in the fume hoods' ability to contain smoke was the equipment placed in the hoods. At the start of testing, the face of hood 218 South was completely blocked on the right side with 3-gallon plastic bottles. Smoke escaped from the hood, which is the criteria for failure of the ASHRAE-110 test. We demonstrated this to the Lab Supervisor, Ruth Post, and several hood operators. They removed the plastic containers, and the hood passed the smoke test. Dead air was observed in front of large temperature baths in hood 218 North and at least one hood in Lab 236. Large objects should be elevated on two-inch blocks.

Both the Siemens controls in Lab 218 and the Phoenix controls in Lab 236 performed adequately to maintain negative pressurization and containment in the fume hoods.

3.4.2 Run Around Coils

In contrast to the use of air-to-air heat exchangers or wheels, run-around loops pose no risk for cross contamination between the exhaust and building supply air streams. The biggest concern for a run-around loop is the risk of the coil in the exhaust fouling or corroding from materials in the exhaust air stream.

4 Stakeholder Input

4.1 Concerns over safety

Concerns raised by CalOSHA and ARB on speed of response are addressed in the previous section.

5 Recommended Language for the Standards Document, ACM Manuals, and the Reference Appendices

Laboratory Exhaust Systems

5.1 Standards

5.1.1 Section 101 Definitions. Add new definition for Covered Process and Covered Process Load as follows

COVERED PROCESS includes the following:

- Datacom equipment
- Laboratory exhaust
- Garage exhasut
- Kitchen ventilation
- Refrigerated warehouses

COVERED PROCESS LOAD is a load resulting from a covered process

5.1.2 Section 101 Definitions. Modify existing definitions for Process and Process Load as follows

EXEMPT PROCESS is an activity or treatment that is not related to the space conditioning, lighting, service water heating, or ventilating of a building as it relates to human occupancyand is not listed as a Covered Process~~.~~

EXEMPT PROCESS LOAD is a load resulting from anExempt ~~process~~Process.

5.1.3 Modify Section 121(e) as follows

121(e) Design and Control Requirements for Quantities of Outdoor Air. All mechanical ventilation and space-conditioning systems shall be designed with and have installed ductwork, dampers, and controls to allow outside air rates to be operated at the larger of (1) the minimum levels specified in Section 121(b)1 or (2) the rate required for make-up of exhaust systems that are required for an Exempt~~process~~Process, for a Covered Process, for control of odors, or for the removal of contaminants within the space.

5.1.4 Deleteexception to Section 122(b) as follows

~~EXCEPTION to Section 122(b)[Criteria for Zonal Thermostatic Controls]: Systems serving zones that must have constant temperatures to prevent degradation of materials, a process, plants or animals.~~

Note: this exception is for a capability of the thermostat to have set-up, set-back and deadband. It is not an operational requirement. All thermostats have this capability today.

5.1.5 Modify exceptions to Section 123 as follows

EXCEPTION 3 to Section 123 [Pipe Insulation]: Piping that serves ~~process loads,~~ gas piping, cold domestic water piping, condensate drains, roof drains, vents, or waste piping.

Note: process piping typically runs at more extreme temperatures and for longer hours it make no sense to exempt it.

5.1.6 Modify Section 141(c) as follows

141(c) [Calculation of Budget and Energy Use] 3. Energy excluded. The following energy shall be excluded:

A. ~~Process~~ Exempt Process loads;

Note: this change allows trade-offs for systems serving Covered Processes loads.

5.1.7 Modify Section 144(c) as follows

144(c) Power Consumption of Fans. Each fan system used for comfort space conditioning shall meet the requirements of Item 1 or 2 below, as applicable. Total fan system power demand equals the sum of the power demand of all fans in the system that are required to operate at design conditions in order to supply air from the heating or cooling source to the conditioned space, and to return it back to the source or to exhaust it to the outdoors; however, total fan system power demand need not include the additional power demand caused solely by air treatment or filtering systems with final pressure drops more than 245 pascals or one-inch water column (only the energy accounted for by the amount of pressure drop that is over 1 inch may be excluded), or fan system power caused solely by Exempt ~~process~~ Process loads.

Note: fan energy for covered process loads should be included.

5.1.8 Modify Section 144(d) as follows

EXCEPTION 4 to Section 144(d) [Reheat/Recool Minimums]: Zones in which specific humidity levels are required to satisfy Exemptor Covered~~process~~ Process~~needs~~loads.Computer Rooms or other spaces with only IT Equipment may not use this exception.

Note: The published IT guidelines have broadened their humidity limits and recent research suggests that no humidity control is necessary. The NEBs standard for telecommunication Central Office Facilities has no lower humidity limit.

5.1.9 Delete Exception 4 to Section 144(e)1 as follows

~~EXCEPTION 4 to Section 144(e)1 [Economizers]: Where it can be shown to the satisfaction of the enforcing agency that the use of outdoor air is detrimental to equipment or materials in a space or room served by a dedicated space conditioning system, such as a computer room or telecommunications equipment room.~~

Note: This is no evidence that this is necessary. See Data Center CASE report.

5.1.10 Modify Exception 3 to Section 144(f) as follows

EXCEPTION 3 to Section 144(f) [SAT reset]: Zones in which specific humidity levels are required to satisfy Exempt or Covered~~process~~ Process ~~needs~~loads. Computer Rooms or other spaces with only IT Equipment may not use this exception.

Note: This is no evidence that this is necessary. See Data Center CASE report.

5.1.11 Add new requirement to 144 as follows

144(TBD) Buildings with laboratory exhaust systems where the minimum circulation rate to comply with code or accreditation standards is ≤10 ACH or less than the design exhaust airflow shall be capable of reducing zone exhaust and makeup airflow rates to the regulated minimum circulation values, or the minimum required to maintain pressurization relationship requirements whichever is larger.

EXCEPTION TO 144(TBD) Exhaust and supply serving zones where constant volume is required by the AHJ, facility EH&S department or code.

5.1.12 Add a new laboratory HVAC system to the ACM

VAV AHU with 100% OSA supply with preheat coil and cooling coil

CV Exhaust Modeled as a plug load in an unconditioned space equal to the scheduled MHP of the exhaust fans.

VAV zone controls with the airflow minimums to match those mandated by the AHJ for each lab space occupancy.

6 Bibliography and Other Research

6.1 Laboratories for the 21st Century: Case Studies: National Renewable Energy Laboratory, Science and Technology Facility, Golden, Colorado.

7 Appendices

7.1 90.1 Addendum AS

6.5.7 Exhaust ~~Hoods~~ Systems

6.5.7.2 ~~Fume Hoods~~Laboratory Exhaust Systems. Buildings with ~~fume hood~~laboratory exhaust systems having a total exhaust rate greater than ~~15,000~~5,000 cfm shall include at least one of the following features:

~~a. VAV hood exhaust and room supply systems capable of reducing exhaust and makeup air flow rates to 50% or less of design values.~~

a. VAV laboratory exhaust and room supply system capable of reducing exhaust and makeup air flow rates and/orincorporate aenergy recovery system to precondition makeup air from laboratory exhaust that shall meet the following:

A + B ≥ 50%

Where:

A = Percentage that the exhaust and makeup air flow rates can be reduced from design conditions.

B = Percentage *sensible recovery effectiveness*.

b. VAV laboratory exhaust and room supply systems that are required to have minimum circulation rates to comply with code or accreditation standards shall be capable of reducing *zone* exhaust and makeup air flow rates to the regulated minimum circulation values,or the minimum required to maintain pressurization relationship requirements.Non regulated *zones* shall be capable of reducing exhaust and makeup air flow rates to 50% of the zone design values, or the minimum required to maintain pressurization relationship requirements.

~~b~~c. Direct makeup (auxiliary) air supply equal to at least 75% of the exhaust air flow rate, heated no warmer than 2°F belowroom set point, cooled to no cooler than 3°F above room setpoint, no humidification added, and no simultaneous heating and cooling used for dehumidification control.

~~c. Energy recovery systems to precondition makeup air from fume hood laboratory exhaust in accordance with Section 6.5.6.1, Exhaust Air Energy Recovery, without using~~

~~any exception.~~

6.5.7 Exhaust Systems

6.5.7.2 Laboratory Exhaust Systems. Buildings with laboratory exhaust systems having a total exhaust rate greater than 5,000 cfm shall include at least one of the following features:

a. VAV laboratory exhaust and room supply system capable of reducing exhaust and makeup air flow rates and/or incorporate aenergy recovery system to precondition makeup air from laboratory exhaust that shall meet the following:

$$A + B \geq 50\%$$

Where:

A = Percentage that the exhaust and makeup air flow rates can be reduced from design conditions.

B = Percentage *sensible recovery effectiveness*.

b. VAV laboratory exhaust and room supply systems that are required to have minimum circulation rates to comply with code or accreditation standards shall be capable of reducing *zone* exhaust and makeup air flow rates to the regulated minimum circulation values, or the minimum required to maintain pressurization relationship requirements. Non regulated *zones* shall be capable of reducing exhaust and makeup air flow rates to 50% of the zone design values, or the minimum required to maintain pressurization relationship requirements.

c. Direct makeup (auxiliary) air supply equal to at least 75% of the exhaust air flow rate, heated no warmer than 2°F below room set point, cooled to no cooler than 3°F above room set point, no humidification added, and no simultaneous heating and cooling used for dehumidification control.