CODES AND STANDARDS ENHANCEMENT INITIATIVE (CASE)

Draft Measure Information Template – Data Centers

2013 California Building Energy Efficiency Standards

California Utilities Statewide Codes and Standards Team, March 2011

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Measure Information Template
Data Centers

2013 California Building Energy Efficiency Standards

March 30, 2011

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1. Overview

1.1 Measure Title
Data Centers

1.2 Description
Prior to 2013 computer rooms were generally considered exempt from Title 24 due to the process exemption. This interpretation is not necessarily supported by the Standard but is nevertheless a common interpretation. This measure makes it clear that computer rooms, i.e. data centers, are not exempt from Title 24. It also establishes a number of new prescriptive requirements that are specific to computer rooms, including:

- Requiring economizers in small computer rooms in buildings that have economizers
- Exempting some computer room expansions from the economizer requirement
- Exempting some new computer rooms in existing buildings from the economizer requirement
- Prohibiting reheat in computer rooms
- Prohibiting non-adiabatic humidification in computer rooms
- Limiting power of fan systems serving computer rooms to 27 watts/kBtuh of net sensible cooling capacity
- Requiring variable speed controls on all chilled water fan systems and all direct expansion (DX) systems over 5 tons serving computer rooms.
- Requiring containment in large, high density data centers with air-cooled computers

A computer room as defined herein is any room with more than 20 W/ft^2 of computer equipment. So the measures in this proposal apply to computer rooms ranging from small computer closets in office buildings to large stand-alone data centers. This measure also defines modeling rules for computer room performance compliance.

1.3 Type of Change
The measure includes new prescriptive requirements and new modeling rules. One could argue that it also includes new mandatory requirements since it makes it clear that the existing mandatory requirements apply to computer rooms, which have commonly been interpreted as exempt from Title 24.

1.4 Energy Benefits
There are no current requirements for computer rooms (per common interpretation) so it is difficult to define the baseline. There are also a number of system types that meet the requirements of this measure and computer rooms vary dramatically in their load density so it is also difficult to define the proposed case. Furthermore, typical practice in data centers is

changing rapidly. Data centers being built today are generally much more efficient than data centers built even a couple years ago. We have run a series of energy simulations of typical baseline and typical proposed case data centers to arrive at the following typical energy benefits of this measure:

	Electricity Savings (kwh/yr)	Demand Savings (kw)	Natural Gas Savings (Therms/yr)	TDV Electricity Savings	TDV Gas Savings
Per Prototype Building*	5,500,000	160	0	$10,500,000	0
Savings per square foot	280	0.008	0	$520	0

* Prototype building: 20,000 ft^2, 100 W/ft^2 design IT load

1.5 Non-Energy Benefits

Non-energy benefits include:

- Improved IT performance – containment and cold aisle temperature monitoring can reduce hot spots that affect computer performance
- Reduced noise – variable speed fans reduce noise
- Improved comfort – without containment it was often necessary to overcool the space to avoid hot spots. With containment cold aisle temperature can be raised improving comfort for occupants. It is common, for example, to make the computer room a 70oF "cold" plenum, with hot air limited to relatively small hot enclosures (e.g. chimney racks).
- Improved monitoring – when variable speed fans are added, cold aisle temperature sensors are often also added. Variable speed fans also provide additional feedback that constant speed fans do not (e.g. current, power, etc.)
- Redundancy – economizers provide backup cooling capacity most of the time should a compressor fail. Adding a VAV box to a small computer room that is already served by a split DX system, for example, provides redundancy should the split DX system fail.

1.6 Environmental Impact

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

1.7 Technology Measures

1.7.1 Measure Availability:

This measure encourages the increased use of the following technologies. All of these technologies are in widespread use in data centers today and are available from multiple manufacturers, some of which are listed below.

- Variable speed drives or EC fan motors – ABB, Siemens, GE, ebm-Papst

- CRAC or AC units with multiple stages of compression or variable capacity DX compressors – Liebert, Stulz, DataAire, APC, Carrier, Aaon, Daikin, Mitsubishi, etc.
- Strip curtains, blanking panels and other containment products – APC, Wrightline, Chatsworth, SubZero Engineering, etc.
- Direct evaporative cooling media or ultrasonic humidifiers – Munters, Stulz,

1.7.2 Useful Life, Persistence, and Maintenance:

Energy savings from this measure will persist for the life of the system. Commissioning is required to achieve and maintain the full savings potential. For example, if variable speed fans are used and fan speed is modulated to maintain the worst case cold aisle temperature sensor at setpoint then personnel must track which sensors are driving the fan speed and investigate consistently unsatisfied sensors. Data centers are typically equipped with sophisticated digital control systems that allow the operators to easily monitor and optimize system performance.

1.8 Performance Verification of the Proposed Measure

Commissioning is required to achieve and maintain the full savings potential. The existing acceptance tests for economizers and supply fan variable flow controls would apply where those technologies are employed. No additional acceptance testing is necessary. Note that data centers are typically extensively commissioned to insure reliability and energy efficiency.

1.9 Cost Effectiveness

Life cycle cost (LCC) per unit and per prototype building were calculated using the Energy Commission Life Cycle Costing Methodology posted on the 2013 Standards website for each proposed measure. Results of the analysis are summarized in the following table. Details of the analysis, including results for different climate zones, are included in Section 3.

a	c		d		e		f	g	
Measure Name	Additional Costs[1]– Current Measure Costs (Relative to Basecase) ($)		Additional Cost[2]– Post-Adoption Measure Costs (Relative to Basecase) ($)		PV of Additional[3] Maintenance Costs (Savings) (Relative to Basecase) (PV$)		PV of[4] Energy Cost Savings – Per Proto Building - 15 yr measure life (PV$)	LCC Per Prototype Building ($)	
	Per Unit	Per Proto Building	Per Unit	Per Proto Building	Per Unit	Per Proto Building		(c+e)-f Based on Current Costs	(d+e)-f Based on Post-Adoption Costs
Economizer – Small Stand-Alone Computer Room	$1,500	$1,500	$1,500	$1,500	$1,194	$1,194	$14,100	-$11,406	-$11,406
Economizer – Small Stand-Alone Computer Room – Air-Air Heat Exchanger	$5,205	$5,205	$5,205	$5,205	$2,388	$2,388	$13,529	-$5,936	-$5,936
Small Computer Room in Office Building – VAV Box	$3,129	$31,290	$3,129	$31,290	$597	$5,970	$94,780	-$57,530	-$57,530
Large Data Center – Water Economizer	$204	$116,042	$204	$116,042	$80	$45,507	$845,286	-$684,307	-$684,307
VAV Fan Control – DX Systems	$3,000	$30,000	$3,000	$30,000	$2,380	$23,800	$102,610	-$78,810	-$78,810
VAV Fan Control – CHW Systems	$282	$2,820	$282	$2,820	$0	$0	$19,300	-$16,480	-$16,480
Containment	$0	$0	$0	$0	$0	$0	$3,079,689	-$3,079,689	-$3,079,689

1.10 Analysis Tools

Currently available simulation programs such as eQuest are capable of modeling the technologies encouraged by this measure. Containment for example, can be modeled by increasing the airside ΔT and the return air temperature to the air handler. Other features that can easily be modeled include: variable volume single zone systems and air and waterside economizers.

1.11 Relationship to Other Measures

Variable speed single zone – Variable speed single zone control is already required in Title 24 and would apply to data center (e.g. CRAC units) once the scope exception for data centers is removed. This measure also would expand the coverage of the variable speed single zone requirement as it applies to data centers. Taylor Engineering is proposing another measure that would expand the coverage of variable speed single zone control for all applications, including data centers. These two measures are not in conflict.

2 Methodology

Computer rooms range from tiny IDF closets to huge stand-alone data centers. The methodology for evaluating the cost effectiveness of this measure has been to break down the measure into individual measures and develop cost and energy models of basecase and proposed case for how each measure affects each type of data center. In addition to prototype cost and energy models we also identified examples of existing data centers and available products that meet the proposed case designs.

Each individual measure and the associated analysis are described in more detail in the next section.

3 Analysis and Results

3.1 Economizer Requirements

Different types of data centers would meet the proposed economizer requirements in different ways. Four different data center scenarios were evaluated in order to reasonably cover the range of data center types. Scenarios:

1. Small Stand-Alone Computer Room – Air Economizer
2. Small Stand-Alone Computer Room - Air-to-Air Heat Exchanger
3. Small Computer Room in Office Building – VAV Box
4. Large Data Center – Water Economizer

3.1.1 Small Stand-Alone Computer Room – Air Economizer

Title 24-2008 requires economizers for "Each individual cooling fan system that has a design supply capacity over 2,500 cfm and a total mechanical cooling capacity over 75,000 Btu/hr", i.e. for any system over 6.25 tons. This measure would lower that threshold to 5 tons and above. DOE-2 was used to evaluate energy savings of an airside economizer on a data center with a 5 ton cooling system.

3.1.1.1 Energy Analysis

3.1.1.1.1 DOE-2 Version

eQuest version 3.63b, build 6510 was used to perform the simulation runs. DOE-2.2 is the calculation engine.

3.1.1.1.2 Building Envelope

- Single story, 225 ft^2 square building. Floor to ceiling height is 9 feet, plenum height is 3 feet.
- No windows or skylights.
- Single zone for entire building.
- Exterior wall construction is R-19. Roof is R-30.

3.1.1.1.3 Climate

The simulation was initially run in Los Angeles (Zone 6). The weather files that were used in the simulations came from the California Energy Commission (CEC) and were developed for Title 24 – 2013 for use with the TDV rates. Because the measure is overwhelmingly cost effective in this scenario, it was determined that analysis in other climate zones was not necessary. Zone 6 is one of the hotter climates with fewer economizer hours than many other zones (e.g. Zone 3, 4).

3.1.1.1.4 Internal Loads

- Lighting power density: 1.0 W/ft^2

- Equipment power density: 100.0 W/ft^2
- Occupancy density: 500 ft^2/person

3.1.1.1.5 Load Schedules

In order to investigate varying fractions of equipment loads in a computer room over the life of the computer room, three seasons were created. Each season contains four values of a fraction of the equipment power density load from 0.25 to 1.0, in 0.25 increments. See Figure 1 below for the annual equipment power density load profile. This is the load profile that was recently adopted by ASHRAE 90.1 for performance compliance and is proposed below for new ACM rules for data centers.

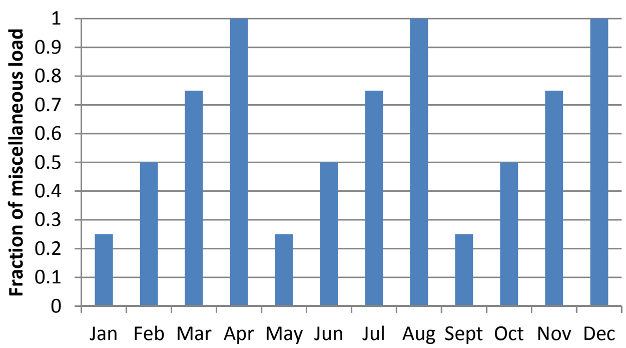


Figure 1: Annual Equipment Power Density Load Profile

The lighting and occupancy schedules follow a standard office schedule. Both peak at 0.9 during occupied hours and drop to 0.05 and 0.0, respectively, during unoccupied hours. Occupied hours run from 8am to 5pm, Monday through Friday, excluding holidays.

3.1.1.1.6 Fan Schedule

24 hours of operation, 7 days a week.

3.1.1.1.7 Temperature setpoints

60ºF design supply air temperature and 80°F return air temperature setpoint

3.1.1.1.8 Zone Properties

- OA-FLOW: Minimum outdoor air was calculated based on the larger of 15 cfm/person and 0.15 cfm/sqft, per Title 24. In this case, the outdoor air required based on 0.15 cfm/sf controls since the occupant density is 500 sf/person. Note that minimum outdoor air is maintained at all times for space pressurization.
- DESIGN-COOL-T: 80F

- TYPE-ZONE
- Ground Floor Single Zone: CONDITIONED
- Plenum Zone: PLENUM

3.1.1.1.9 System Properties for all cases

- SYSTEM-TYPE: Packaged Single Zone with DX cooling and no heating.
- RETURN-AIR-PATH: PLENUM-ZONES.
- Supply fan:
- SUPPLY-STATIC: 1.25”
- SUPPLY-EFF: 53%
- FAN-CONTROL: CONSTANT-VOLUME
- RECOVER-EXHAUST: NO. This indicates that there is no heat recovery.
- Cooling:
- DX cooling
- MIN-SUPPLY-T: 60°F
- COOLING-EIR: 0.2310. Converted from the minimum efficiency of 12.1 EER for a Unitary AC that is less than 65,000 Btu/h from Title 24 and ASHRAE Standard 90.1 (Table 6.8.1A).
- CONDENSER-TYPE: AIR-COOLED.
- HEAT-SOURCE: None.

3.1.1.2 Parametric Runs

3.1.1.2.1 Conceptual Explanation

The baseline is a Packaged Single Zone system without an economizer. Economizers in small packaged units are often not truly integrated due to discrete compressor capacity steps. Therefore, three parametric runs were set-up to investigate full and partial economization:

1. A fully integrated economizer with a differential drybulb high limit switch
2. A partially-integrated economizer per DOE-2
3. A non-integrated economizer with a *low* fixed drybulb high limit switch of 60F

The first run is a fully integrated economizer that can run simultaneously with the compressor when it cannot provide all the cooling itself, as long as the outside air is cooler than the return air. However, a 5 ton unitary cooling system will not be truly integrated since such a small unit will most likely have one compressor without unloading. The compressor has a minimum run time. The compressor can over cool the supply air before it meets its minimum run time which can cause the economizer to temporarily close (false load the compressor) to maintain supply air setpoint. Therefore, the second parametric run was set-up to more realistically simulate how an economizer would behave in a 5 ton unit.

The second parametric run is a partially-integrated economizer. It has a fixed drybulb high limit of 70F. It cannot operate simultaneously with the compressor. However, unlike the non-integrated economizer modeled in the third parametric run, the partially-integrated economizer can operate when the outdoor temperature is between 60F and 70F, as long as it can provide enough cooling to handle the entire load in a given hour (DOE-2 uses hourly time steps). This happens often enough to differentiate the partially-integrated economizer from the non-integrated economizer, since the setpoint in the space is high, at 80F. Thus, the partially-integrated economizer will run when the outside air is below 70F and can often provide enough cooling for the entire space.

The third parametric run is a truly non-integrated economizer that will disable the economizer whenever the outside air temperature is above the design supply air temperature of 60F. This is an unrealistic and inefficient system for a 5 ton unit. This run is just for our reference internally to verify that the economizers are behaving as expected.

The first and second parametric runs were averaged to comprise the "proposed" case. This method was used to most accurately represent a partially-integrated economizer in a 5 ton unitary cooling system.

3.1.1.2.2 Modeling Explanation

In the first case, a differential drybulb high limit switch is specified, which compares the return air to the outside air drybulb temperature to determine when it is beneficial to bring in more outside air than return air. This is modeled in eQuest by utilizing the "DUAL-TEMP" outside air control option. A differential limit is used instead of a fixed limit since the setpoint in the space is higher than a typical building, at 80F. A differential limit allows for more "free cooling" than a fixed limit. The economizer in this case is fully integrated, which is modeled in eQuest by setting "ECONO-LOCKOUT = NO". This indicates that the economizer and the compressor can run simultaneously. If the economizer cannot provide all the cooling that is necessary, it will remain on and the remainder of the cooling will be mechanically provided by the compressor.

In the second case, the economizer is partially-integrated, which is modeled by setting "ECONO-LOCKOUT = YES". This means that the economizer and compressor cannot operate simultaneously. DOE-2 uses an hourly timestep when performing simulations. Therefore, if the economizer cannot meet the entire cooling load in a given hour then it does not run at all that hour. The drybulb limit is fixed at 70F, as opposed to allowing a differential drybulb limit, which means that the economizer will not operate if the outside air is above 70F. This is indicated in eQuest by setting "OA-CONTROL = OA-TEMP".

The third case is modeled as a true non-integrated economizer would behave, with a low, fixed drybulb limit of 60F. If all the cooling for the space cannot be provided by the economizer when the outside air is 60F or lower, than it shuts off completely. This case is modeled the same as the second case, except with a different DRYBULB-LIMIT. For example: if the outside air temperature is 63F, the economizer in this case cannot operate and all of the cooling will be mechanically provided by the compressor. However, the partially-integrated economizer in the second case could operate with 63F outside air, provided it can meet the entire load with 63F air.

3.1.1.2.2.1 Baseline: PSZ without economizer

1. OA-CONTROL: FIXED fraction. A fixed amount of outside air will be brought in whenever the fans are running, which is 24/7. It is not based on Thus, this simulates the absence of an economizer
2. MAX-OA-FRACTION: 1.0
3. DRYBULB-LIMIT: n/a
4. ECONO-LOCKOUT: NO

3.1.1.2.2.2 Parametric Run 1: PSZ with an integrated economizer with a differential drybulb high limit switch

1. OA-CONTROL: DUAL-TEMP. The economizer is enabled when the outside air temperature is below the return air temperature. This input indicates that the economizer uses a differential drybulb limit, as opposed to a fixed drybulb limit, to determine how much outside air to bring in for “free” cooling.
2. MAX-OA-FRACTION: 1.0
3. DRYBULB-LIMIT: n/a
4. ECONO-LOCKOUT: NO. This indicates that the economizer is integrated.

3.1.1.2.2.3 Parametric Run 2: PSZ with a non-integrated economizer with a fixed drybulb high limit switch

1. OA-CONTROL: OA-TEMP. The economizer is enabled when the outside air temperature is below the DRYBULB-LIMIT, the maximum allowed outside air temperature.
2. MAX-OA-FRACTION: 1.0
3. DRYBULB-LIMIT: 70°F
4. ECONO-LOCKOUT: YES. This indicates that the economizer is non-integrated.

3.1.1.2.2.4 Parametric Run 3: PSZ with a non-integrated economizer with a low fixed drybulb high limit switch

1. OA-CONTROL: OA-TEMP. The economizer is enabled when the outside air temperature is below the DRYBULB-LIMIT, the maximum allowed temperature.
2. MAX-OA-FRACTION: 1.0
3. DRYBULB-LIMIT: 60°F
4. ECONO-LOCKOUT: YES. This indicates that the economizer is non-integrated.

3.1.1.2.3 Cooling System Sizing

The auto-sizing feature in DOE-2 is not reliable. Therefore, the model was run iteratively: first it was run to determine the peak cooling load then the equipment was manually sized at 125% of the peak load. The baseline and parametric runs were run with the manually-entered cooling equipment capacities. These numbers were normalized to a 5 ton unit for the energy savings and economic analysis.

3.1.1.3 Sample eQuest Output: 36 hour study

Output for a 36 hour period was looked at for each of the runs to verify that the economizers in each run were behaving as expected. The 36 hour period from midnight on May 21 to noon on May 22 was selected due to the large swing in outside air temperature, peaking at 89°F on the afternoon of the 21st and dropping to 44°F in the early hours of the 22nd. Thus there are times when the economizer is running and when it is not running. The output for the runs is shown below, in Figures 2 through 5.

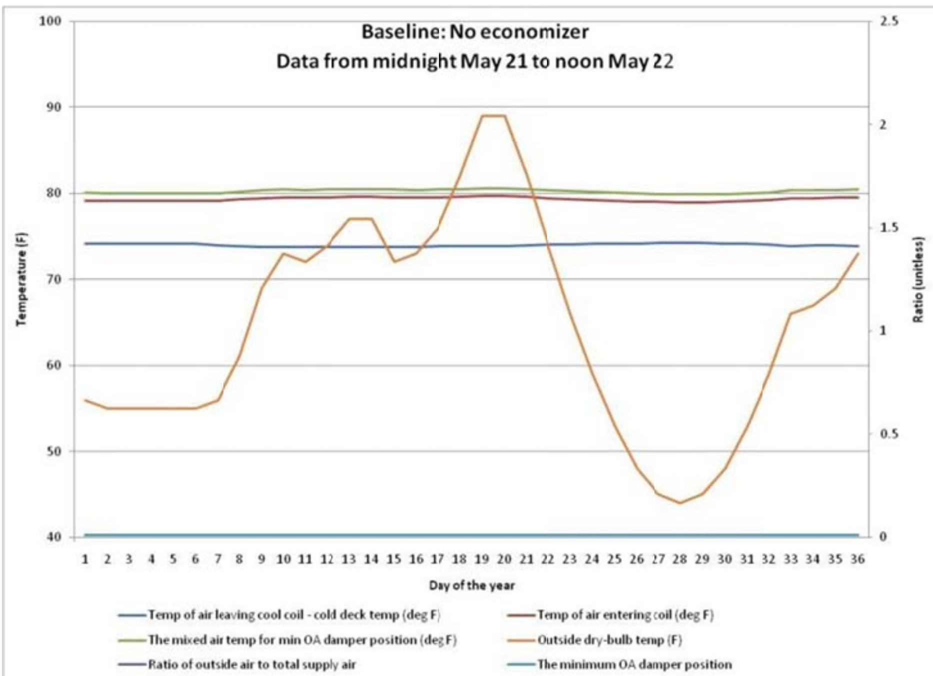


Figure 2: Baseline output for 36 hour study

In the Baseline, where there is no economizer, the ratio of outside air to total supply air always equals the minimum outside air, as shown in Figure 2. In the first parametric run, as shown in Figure 3, the economizer is fully-integrated and operates whenever the outside air (orange line) is cooler than the return air (red line). When the economizer is operating, the ratio of outside air to total supply air (purple line) exceeds the minimum outside air ratio (bright blue line). The fully-integrated economizer operates most hours, except when the outside air gets very warm. In this 36 hour period, the economizer only shuts off from hours 18 to 21 when the outside air reaches the 80s °F.

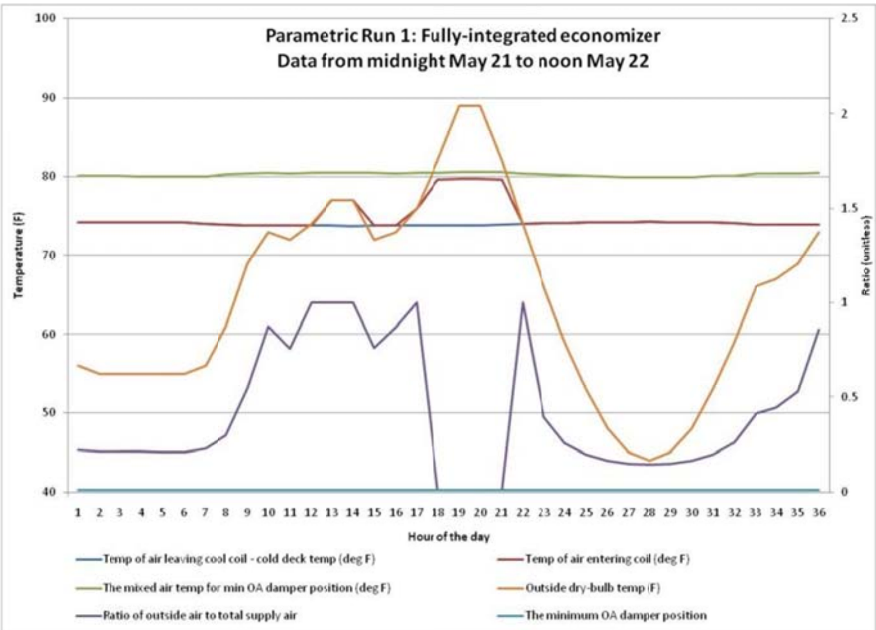


Figure 3: Fully-integrated economizer output for 36 hour study

As shown in Figure 4, the partially-integrated economizer in parametric run 2 operates fewer hours than the fully-integrated economizer, as expected. It shuts off, which means the ratio of outside air to total supply air drops to zero, whenever the outside air temperature exceeds 70F. It also only operates when it can satisfy the entire load. There are hours when the outside air exceeds 70F but it still cooler than the return air. From hour 10 to 18 and hour 21 to 22, the fully-integrated economizer operates but the partially-integrated economizer does not.

The non-integrated economizer behaves similarly to the partially-integrated economizer. However, there are hours when the partially-integrated economizer is on but the ratio of outside air to total supply air drops to the minimum for the non-integrated economizer. This occurs from hours 8 to 10 and 22 to 23, as shown in Figures 4 and 5.

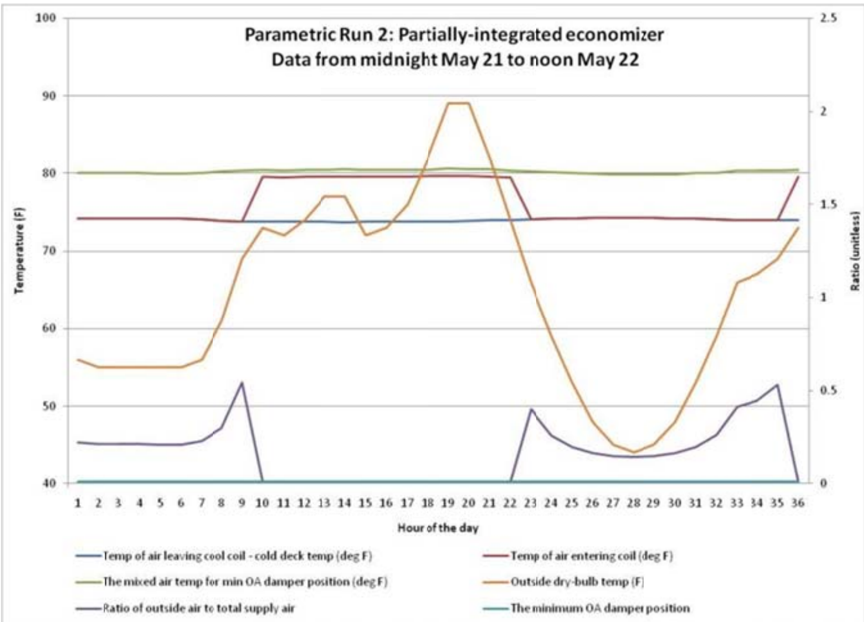


Figure 4:Partially-integrated economizer output for 36 hour study

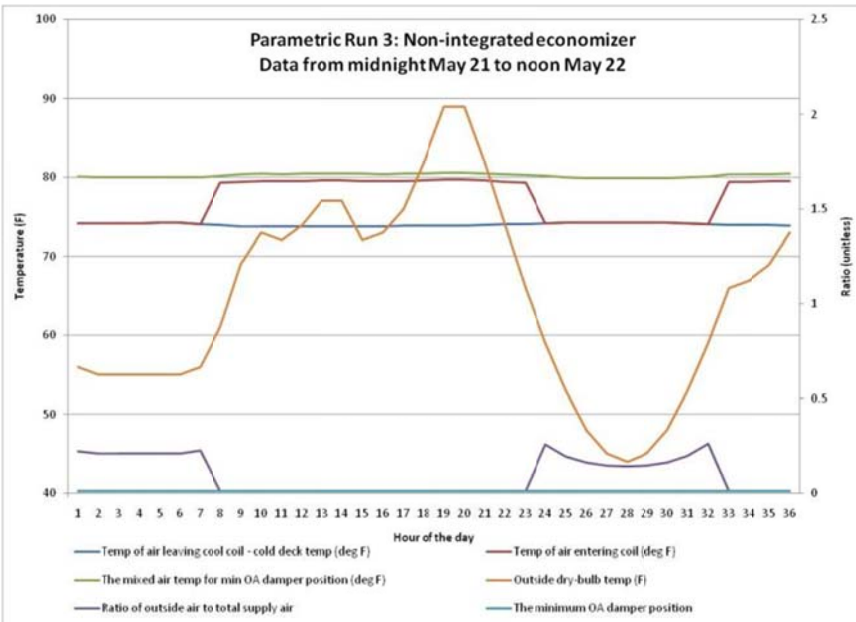


Figure 5: Non-integrated economizer output for 36 hour study

3.1.1.4 Energy Results

The energy savings were normalized from the 8.31 ton unitary cooling system in the eQuest model to a 5 ton cooling system. As expected, the fully integrated economizer saves the most energy, and the non-integrated economizer with the low drybulb limit saves the least amount of energy, as shown in Table 1. The additional pump and auxiliary energy in the runs when there is an economizer is from 0.05 kW of crankcase heat for the compressor that occurs when the outside air temperature drops below 50°F. Overall, adding an economizer saves between one-third and two-thirds of the total HVAC energy when compared to a unitary cooling system without an economizer.

Table 1. Annual End-Use Energy Consumption for a 5 ton unit for Climate Zone 06 [kWh]

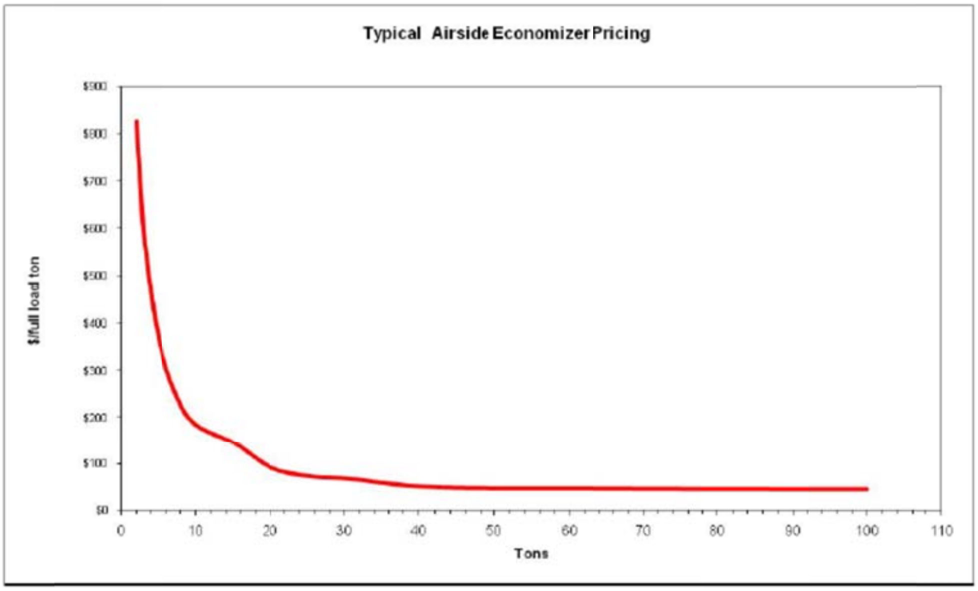
Case		Space Cooling	Pumps & Aux	Vent Fans	HVAC Total	% HVAC Savings from Baseline
Base	Basecase	14,111	0	5,049	11,528	*n/a*
Run 1	Add fully-integrated economizer	2,152	29	5,049	4,350	*62%*
Run 2	Add partially-integrated economizer	5,155	29	5,049	6,157	*47%*
Run 3	Add non-integrated economizer	7,966	29	5,049	7,848	*32%*

TDV energy cost savings are shown below with the lifecycle cost results.

3.1.1.5 Incremental Installed Cost

The incremental economizer cost used in this analysis was $1,500 for a 5 ton unit per Figure 6 ($300/ton for 5 tons). This data was provided by Richard Lord of Carrier Corporation. It was collected from AHRI in January 2010 for use in the analysis that used to justify expanding the economizer requirements in ASHRAE Standard 90.1-2010. This is an actual installed cost, including general contractor markup and commissioning.

Figure 6. Incremental Economizer Cost per Ton. Source: Carrier Corp., January 2010.



3.1.1.6 Maintenance Costs

Incremental maintenance cost data was provided by a Bay Area mechanical contractor and service contractor. It is based on approximately 30 minutes of service time per economizer twice a year at a labor rate of $100/hr.

3.1.1.7 Lifecycle Cost Results

As shown in Table 2, the measure is highly cost effective, even under to most conservative assumption (non-integrated economizer).

Table 2. Lifecycle Cost Results for 5 ton unit with Economizer (CZ06)

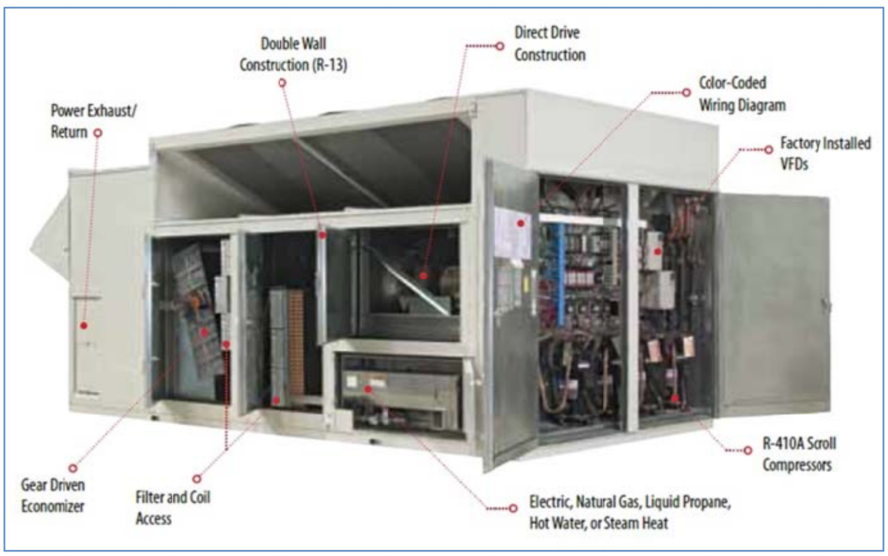
	Run 1 (fully integrated)	Run 2 (partially integrated)	Run 3 (non-integrated)

	Run 1 (fully integrated)	Run 2 (partially integrated)	Run 3 (non-integrated)
Incremental Installed Cost	$ 1,500	$ 1,500	$ 1,500
Incremental Annual Maint.	$ 100	$ 100	$ 100
NPV of Annual Maint.	$ 1,194	$ 1,194	$ 1,194
Total Incremental Cost	$ 2,694	$ 2,694	$ 2,694
NPV of Energy Savings	$ 19,900	$ 14,100	$ 9,500
Lifecycle cost savings	$ 17,206	$ 11,406	$ 6,806
1st yr energy savings	$ 1,672	$ 1,185	$ 798
Simple payback (yrs)	1.6	2.3	3.4

3.1.1.8 Example Product

Figure 7 is an example of packaged unit with an economizer that is currently operating at a data center in Oakland CA.

Figure 7. 5 ton Packaged Unit with Integrated Economizer



3.1.2 Small Stand-Alone Computer Room - Air-to-Air Heat Exchanger

Opponents of the recent ASHRAE 90.1 data center economizer requirements cited gaseous contaminants and particulates as reasons for why airside economizing should not be required in data centers. They pointed to a recent ASHRAE TC9.9 whitepaper that discussed a recent increase in data center hardware failures due to gaseous contaminants (see Bibliography). Researchers at LBNL, UC Berkeley and Taylor Engineering have recently completed a review of this whitepaper and all available data on the subject (see Bibliography). These researchers concluded that the whitepaper was highly misleading, that there was no evidence of hardware failures due to gaseous contaminants or particulates in the US and that there was no evidence that airside economizing increased any such risks. It turns out that the only hardware failures in data centers described in the white paper where in data centers in heavily polluted areas in India and China and none of these had airside economizers. A recent study by Lawrence Berkeley National Laboratory found that particulate concentrations in data centers with and without airside economizers were functionally equal.

Nevertheless, in support of this measure, an analysis was performed to show that airside economizing is cost effective in a small data center even if an air-to-air heat exchanger is required, i.e. airside economizing is cost effective without bringing any outside air into the data center.

3.1.2.1 Energy Analysis

The same model that was used for the Small Stand-Alone Computer Room – Air Economizer analysis (described above in section 3.1.1) was used for this analysis. Deviations from that analysis are described below.

3.1.2.2 Equest model differences

SUPPLY-STATIC: 1.25” in the baseline and 1.67” in the proposed case. The additional 0.42” is the additional static pressure in the heat exchanger that the supply fan must overcome. This figure came from Richard Lord, from Carrier Corporation. Richard has recently worked extensively with the Air-Conditioning, Heating, and Refrigeration Institute (AHRI) to develop typical performance and cost data on air-air heat exchangers as part of a new requirement for energy recovery ventilators in ASHRAE Standard 90.1.

3.1.2.3 Assumptions for Excel Calculations

An air-to-air heat exchanger in this configuration cannot be explicitly modeled in eQuest. Therefore, it was modeled manually using Excel based on the hourly output from the previously described eQuest parametric run.

In the baseline, there is no heat exchanger. In the proposed case, the return air and outside air are run through a counterflow heat exchanger. The two airstreams do not mix, but the outside air is used as a heat sink for the return air.

3.1.2.3.1 Heat Exchanger Assumptions

The output from the eQuest baseline was used as the basecase without any modifications. The proposed case was based on the output from the parametric run with an increased static pressure in the supply fan. The increased static pressure was added to model the additional static pressure from the heat exchanger that the supply fan must overcome.

1. Air-to-air heat exchanger:

   a. Assume heat exchanger flow is equal to supply and return flow, as calculated by eQuest.
   b. Sensible effectiveness = 59%. Source: Richard Lord.
   c. Total static pressure = 0.42” (added to supply fan). Source: Richard Lord.
2. Scavenger fan:
   a. Total static pressure = 0.42”. Source: Richard Lord.
   b. Fan efficiency = 53%. Based on typical fan efficiencies of what is commercially available.
   c. Fan power = 0.32 kW. Fan power was calculated using the assumed fan efficiency and static pressure, and the air flow rate as calculated by eQuest.

The sensible effectiveness for the heat exchanger and the additional static pressures were provided by Richard Lord. Since an air-to-air heat exchanger for a 5 or 10 ton system do not yet exist on a commercial scale, Richard Lord provided placeholder information based on Energy Recovery Ventilators (ERVs), which have similar components to an air-to-air heat exchanger. Richard Lord used theAHRI 1060 directory for ERVs that he developed during the last comment period for ASHRAE Standard 90.1. He sorted the directory by plates and found the average pressure drop and effectiveness at the 1060 standard rating condition, which were 0.42 inches of water and 59%, respectively.

3.1.2.3.2 Operation Modes

There are three modes of operation for this system as shown in Table 2. The most common is Operation Mode 1. In this situation, the heat exchanger is providing as much cooling as it is capable of providing, yet still not satisfying the entire cooling load. Therefore, the compressor runs at a fraction of its full power to provide the rest of the cooling. In Operation Mode 2, the heat exchanger can provide all of the required cooling. The scavenger fan, which has a Variable Frequency Drive (VFD), operates at a fraction of full power and the cooling compressor is off. The least common mode, Operation Mode 3, occurs when the outside air temperature is higher than the return air temperature, thus the outside air cannot provide any free cooling. Therefore, the scavenger fan is turned off and the cooling compressor does all of the cooling. A bypass damper could be added to allow the supply air to bypass the HX in Mode 3 (non-economizer mode) but with the high return air temperatures that are common in data centers there are relatively few hours when the system is not in economizer mode. Therefore, a bypass damper is probably not cost effective and was not modeled.

Table 2: Operation Modes to Satisfy Cooling Load

Operation Mode	Operation	Economizer Cooling	Cooling Compressor	Number of Hours in each Operation Mode for CTZ06

				Annually
1	Mixed mode	*Full* (100%)	*Partial*	5336
2	Economizer mode	*Partial*	*Off* (0%)	3298
3	Non-economizer mode	*Off* (0%)	*Full* (100%)	126

3.1.2.3.3 Temperature Calculations to Determine Operation Mode

In order to determine the operation mode for each hour of the year, the outside air temperature was compared with the return air temperature to determine if the heat exchanger was operating or not. Both the outside air and return air temperatures were taken directly from the output from the eQuest runs. Next, the supply air temperature required to satisfy the cooling load was compared to the coldest supply air temperature that could be provided by the heat exchanger if it were running at 100%. These temperatures were calculated based on the eQuest output, as described in Table 3.

Table 3: Temperature Calculations

Temperature	**Source or calculation**
Return air (RAT)	eQuest output "Temp of air entering coil"
Supply air required to meet load (Desired SAT)	(eQuest output "Temp of air leaving cool coil") + (eQuest output "air stream temperature rise across the supply fan")
Outside air (OAT)	eQuest output "Outside dry-bulb"
ΔT that can be provided by the heat exchanger (ΔT_{HX})	(RAT – OAT) x (Effectiveness of heat exchanger)
Lowest supply air temperature possible using heat exchanger only (Lowest $SAT_{HX\ only}$)	(RAT) – (ΔT_{HX}) + (eQuest output "air stream temperature rise across the supply fan")

The calculated temperatures described in Table 3 were used to determine the mode of operation, as shown below in Table 4. The heat exchanger only operates if the outside air temperature is lower than the return air temperature. When the heat exchanger is operating and the supply air temperature that can be provided by the heat exchanger (Lowest $SAT_{HX\ only}$) is lower than the supply air temperature to satisfy the cooling load (Desired SAT), then the cooling compressor is off and the heat exchanger provides all of the cooling. Otherwise, the cooling compressor runs at partial power to provide the rest of the cooling.

Table 4: Temperature Checks for Modes of Operation

Operation Mode	Heat Exchanger	Cooling Compressor	Temperature Check(s)
1	*Full* (100%)	*Partial*	OAT < RAT Desired SAT < Lowest $SAT_{HX\ only}$
2	*Partial*	*Off* (0%)	OAT < RAT Desired SAT ≥ Lowest $SAT_{HX\ only}$
3	*Off* (0%)	*Full* (100%)	5. OAT ≥ RAT

3.1.2.3.4 Partial Power Operation for Cooling Compressor

The partial power of the cooling compressor during Operation Mode 2 is the directly proportional to the ratio of the amount of cooling that it must provide for a given hour to the amount of cooling that it could provide for that hour if operating at 100%. In other words, it is the ratio of cooling that must be provided by the cooling coil after the heat exchanger provides as much cooling as it can to the amount of cooling that the compressor could provide if there was no heat exchanger. To determine the partial power of the cooling compressor, the full power of the cooling compressor for each hour is multiplied by:

$$Ratio\ of Partial\ Power_{DX} = \left(\frac{\Delta \mathrm{T}\ provided\ by\ cooling\ coil\ with\ HX}{\Delta \mathrm{T}\ provided\ by\ cooling\ coil\ without\ HX}\right)$$

$$Ratio\ of Partial\ Power_{DX} = \left(\frac{RAT - \Delta \mathrm{T_{HX}} - \text{Temperature of Air leaving Cooling Coil}}{RAT - \text{Temperature of Air leaving Cooling Coil}}\right)$$

The resulting power, in kilowatts, multiplied by 1 hour for each hour is the cooling end-use energy for that hour.

3.1.2.3.5 Partial Power Operation for Scavenger Fan

According to the ideal fan laws, power is proportional to the cube of fan speed. To be conservative in this analysis, power was assumed to be proportional to fan speed to the power to 2.6. The partial power of the scavenger fan during Operation Mode 1 is directly proportional to the ratio of the amount of cooling that the scavenger fan needs to provide for a given hour to the amount of cooling that the scavenger fan operating at 100% could provide that hour, raised to the power of 2.6. To determine the partial power of the scavenger fan on an hourly basis, the full power of the scavenger fan is multiplied by:

$$Ratio\ of Partial\ Power_{HX} = \left(\frac{\Delta T\ provided\ by\ HX\ to\ satisfy\ cooling\ load}{\Delta T\ provided\ by\ HX\ operating\ at\ 100\%}\right)^{2.6}$$

$$Ratio\ of Partial\ Power_{HX} = \left(\frac{Return\ air\ setpoint - Desired\ air\ temparature\ leaving\ HX}{Return\ air\ setpoint - (RAT - \Delta T_{HX})}\right)^{2.6}$$

The resulting power, in kilowatts, multiplied by 1 hour is the scavenger fan energy for that hour. It is added to the supply fan energy to determine the end-use fan energy for each hour.

	Occupancy Type	Area (Square Feet)	Number of Stories	Other Notes
Prototype	Computer Room	225	1	No windows to represent an interior computer room

3.1.2.4 Energy Results

The energy savings were normalized from the 8.31 ton unitary cooling system in the eQuest model to a 10 ton unit, as shown in Table 5, and a 5 ton unit, as shown in Table 6. Overall, the fan energy increases, the cooling energy decreases, and total HVAC energy decreases by about two-fifths. The fan energy increases due to the addition of a scavenger fan, which draws the outside air through the heat exchanger. It also increases due to the additional static pressure from the heat exchanger that the supply fan must overcome. Energy for space cooling, however, decreases dramatically due to the free cooling that the heat exchanger provides. Overall, the magnitude of the space cooling savings offsets the increased fan energy and results in significant HVAC total energy savings. Thus, adding a small amount of additional scavenger fan energy to draw air through a heat exchanger creates massive cooling energy savings.

Table 5: Annual End-Use Energy Consumption for Climate Zone 06 for a 10 ton PSZ system [kWh/year]

	Space cooling [kWh]	Fans [kWh]	HVAC total [kWh]
Basecase – 10 ton unit	30,800	10,100	40,900
Proposed Case: Add an air-to-air HX	9,900	14,000	23,900

% Savings from Baseline	*68%*	*-39%*	*42%*

Table 6: Annual End-Use Energy Consumption for Climate Zone 06 for a 5 ton PSZ system [kWh/year]

	Space cooling [kWh]	**Fans [kWh]**	**HVAC total [kWh]**
Basecase – 5 ton unit	15,400	5,000	20,500
Proposed Case: Add an air-to-air HX	5,000	7,000	11,900
% Savings from Baseline	*68%*	*-40%*	*42%*

3.1.2.5 Incremental Installed Cost

Cost data for an Energy Recovery Ventilator (ERV) was used a reasonable conservative estimation for the cost of an air-to-air heat exchanger. The cost per CFM per ton for ERVs, shown below in Figure 1, was provided by Richard Lord of Carrier Corporation from his February 2010 study for ASHRAE Standard 90.1 updates. The data in Figure 1 reflects actual costs, including GC markups and commissioning.

ERVs have similar components to air-to-air heat exchangers, with a few additional components. The ERV data includes costs for the ERV heat exchanger, bypass damper, makeup air fan, exhaust fan, cabinet and connection sheet metal. The ERV deals with both sensible and latent heat, while the air-to-air heat exchanger only deals with sensible heat. Also, the heat exchanger in this study does not have a bypass damper. Therefore, assuming the heat exchangers were mass produced in a manner similar to ERVs, their price per ton would be lower than the ERV cost. Using the ERV data is a reasonable and conservative estimation.

From Figure 1 and conservatively using the 80% outside air cost curve, the incremental cost for adding an air-to-air heat exchanger used in this analysis was $8,300 for a 10 ton unit and $5,200 for a 5 ton unit (assuming 400 CFM/ton).

Figure 1: Cost per CFM per ton for ERVs – Used as reasonable estimation for air-to-air heat exchanger cost data

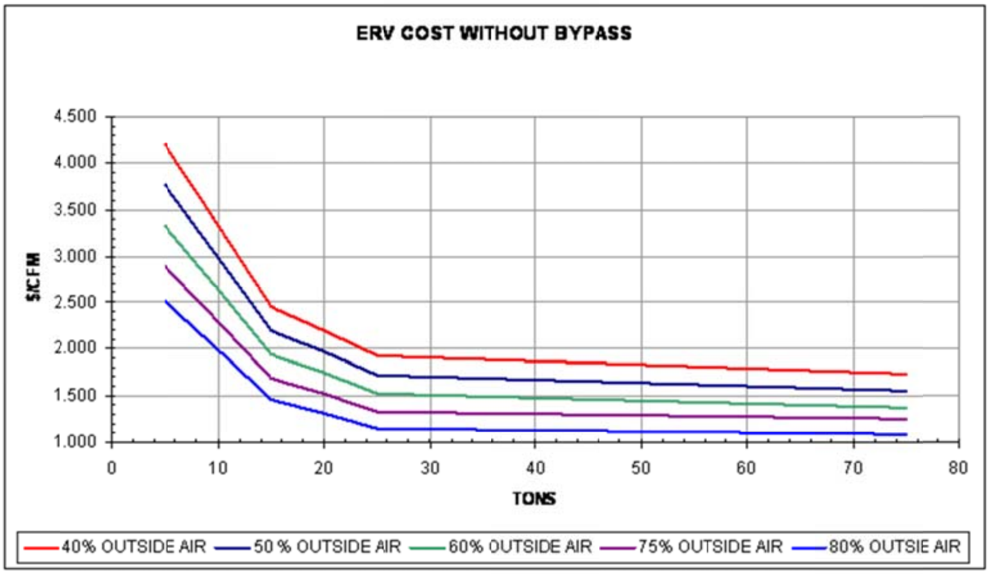


Figure 8: Incremental Energy Recovery Ventilator (ERV) Cost per Ton. Source: Richard Lord, February 2010.

3.1.2.6 Maintenance Cost

Incremental maintenance cost data was provided by a Bay Area mechanical contractor and service contractor. It is based on approximately one hour of service time per unit twice a year at a labor rate of $100/hr.

3.1.2.7 Lifecycle Cost Results

As shown in Figure 3, the measure is highly cost effective, even for a 5 ton unit.

Table 3. Lifecycle Cost Results for 5 and 10 ton unit with Air-Air Heat Exchanger (CZ06)

	10 TON	5 TON
Incremental Installed Cost	$ 8,328	$ 5,205
Incremental Annual Maint.	$ 100	$ 100
NPV of Annual Maint.	$ 2,388	$ 2,388
Total Incremental Cost	$ 10,716	$ 7,593
NPV of Energy Savings	$ 27,057	$ 13,529

Lifecycle cost savings	$ 16,341	$ 5,936
1st yr energy savings	$ 2,274	$ 1,137
Simple payback (yrs)	4.7	6.7

3.1.2.8 Example Product

Figure 9, Figure 10, and Figure 11are examples of air-air heat exchanger products that are currently used in data centers around the world.

Figure 9. Air-Air Heat Exchanger with Indirect Evaporative Cooling by Munters

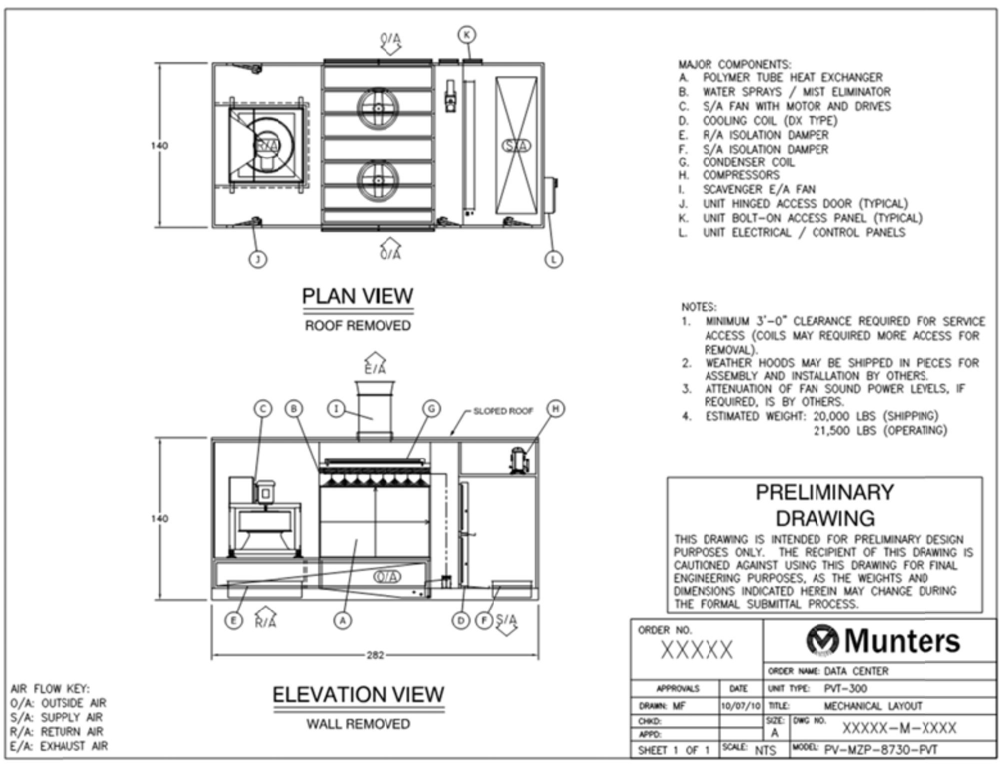
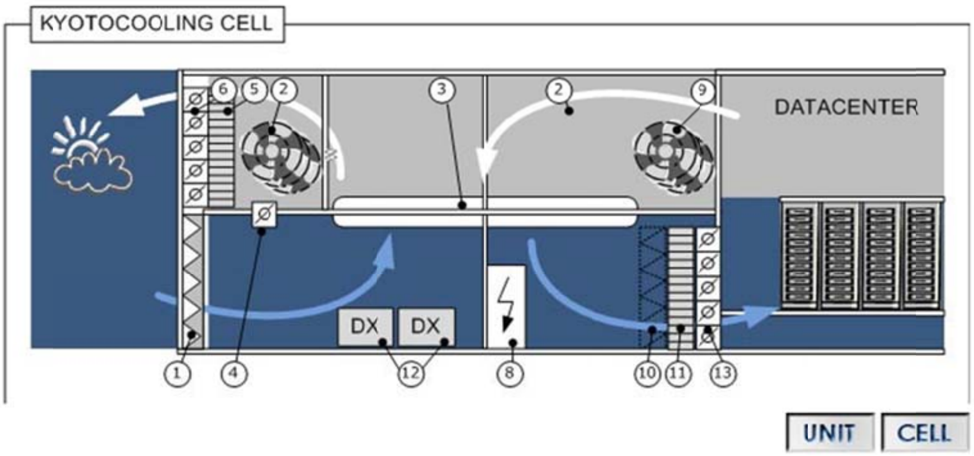


Figure 10. Kyoto Wheel Air-Air Heat Exchanger



To maintain the cooling conditions of the data room the following equipment has been installed:

- Filters outside air (1);
- Fans (2) to maintain the flow of open air over the rotary heat exchanger;
- A rotary heat exchanger (3) to cool the recirculation air to the open air;
- Bypass damper and air duct (4) to preheat the outside air;
- Condenser (5) and blocking damper (6) to build up resistance to preheat outside air;
- Return air damper (7) to block in the unit;
- Electrical cabinet (8) with controls and power distribution;
- Fans (9) to maintain the recirculation flow in the data room;
- Filter recirculation air (10);
- Mechanical cooling by DX (direct expansion) coolers (12), an evaporator (11) and a condenser (5);
- Supply air damper (13) to block in the unit / cell.

Figure 11. EcoBreeze Air-Air HX with Indirect Evaporative Cooling by APC/Schneider.

3.1.3 Small Computer Room in Office Building – VAV Box

Office buildings often have one or several small computer rooms that contain servers and other electronic equipment. While the office space cycles between occupied and unoccupied time, the servers run 24/7. Thus, computer rooms require cooling at all times while systems serving office spaces only have to meet setback temperatures during unoccupied hours. Office areas are typically served by VAV systems with airside economizers while computer rooms are served by DX split systems to satisfy their 24/7 load. It has not been considered cost effective to serve the computer rooms from the VAV system since the computer rooms require cooling at night when the VAV system is in unoccupied mode. With modern DDC systems, however, it is quite easy to shut off conditioning to office spaces at night and just serve computer rooms from the central VAV system. This analysis investigates a computer room that is served by a large VAV system running in economizer mode in order to investigate the lifecycle cost effectiveness of taking advantage of free cooling from the economizer by serving a computer room off of a large VAV system.

The basecase is a small computer room in an office building served by a split DX system. In the proposed case the computer room is served by both a split DX system and by a cooling only VAV box off of the central VAV system. Thus the incremental cost of the proposed case is the entire first cost of the VAV box and associated controls. This is very conservative since the DX system could be eliminated from the proposed case. However, to eliminate the DX system the air handler and ductwork for the VAV system would have to be increased in size to serve the computer room at air handler peak load. The cost to increase the AHU size and ductwork was not included in the analysis. Thus the VAV box serving the computer only operates when the AHU has spare capacity (e.g. at night, winter, weekends, etc.). When the AHU does not have spare capacity (e.g. hot summer days) then the VAV box is shut off and the DX split system serves the computer room. The recommended language for the Standards document was carefully crafted to not require that the central VAV system be sized to serve the computer rooms as long as it is capable of serving the computer rooms most of the time.

3.1.3.1 Energy Analysis

A model for the baseline was set-up in eQuest. It is a single zone computer room that is served by a packaged split system with DX cooling and no heating component. The zone has a high process load that varies throughout the year to investigate varying server loads throughout the lifetime of the computer room.

The proposed cases come from two parametric runs that were set-up in the eQuest model. Both of the proposed cases include a fully integrated economizer and lower fan power. When a large central VAV system operates to serve just a small computer it effectively operates as a constant volume system (with very low fan power) with supply air temperature reset. Thus in the first parametric run the system is modeled as a constant volume single zone DX system with an airside economizer.

Comment [JS1]: Dierdre, this is not clear. Can you explain.

The results from the eQuest runs were post-processed in spreadsheets by filtering for the hours considered in the three proposed cases, which are:

1. Savings during unoccupied hours only
2. Savings during unoccupied hours when the economizer is operating
3. Savings during all unoccupied hours plus occupied hours when the economizer is operating

In the second proposed case, the cooling for the computer room would be provided by free cooling from the VAV economizer and supplemented by the PSZ split DX. Only the savings during unoccupied hours were investigated in the first two cases. Thus, these runs are quite conservative since they do not account for savings from serving the computer rooms with VAV boxes during occupied hours. These hours are taken into consideration in the third case.

Comment [JS2]: Above you say 2 proposed cases, here you say 3??

See section 3.1.1.1 for a description of the model assumptions, including building envelope, climate, and computer room loads and load schedules

The office building hours of operation are assumed to be 6am to 6pm, Monday through Saturday

3.1.3.1.1 Temperature Setpoints

60°F design supply air temperature and 80°F return air temperature setpoint.

3.1.3.1.2 Basecase Zone Properties

1. OA-FLOW: Minimum outdoor air was calculated based on the larger of 15 cfm/person and 0.15 cfm/sqft, per Title 24. In this case, the outdoor air required based on 0.15 cfm/sf controls since the occupant density is 500 sf/person. Note that minimum outdoor air is maintained at all times for space pressurization.
2. DESIGN-COOL-T: 80F
3. TYPE-ZONE
   a. Ground Floor Single Zone: CONDITIONED
   b. Plenum Zone: PLENUM

3.1.3.1.3 Basecase System Properties

1. SYSTEM-TYPE: Packaged Single Zone with DX cooling and no heating.
2. OA-CONTROL: FIXED, which means there is no economizer.
3. RETURN-AIR-PATH: PLENUM-ZONES.
4. Cooling:
   a. DX cooling
   b. MIN-SUPPLY-T: 60°F
   c. COOLING-EIR: 0.2310. Converted from the minimum efficiency of 12.1 EER for a Unitary AC that is less than 65,000 Btu/h from Title 24 and ASHRAE Standard 90.1 (Table 6.8.1A).
   d. CONDENSER-TYPE: AIR-COOLED
   e. MIN-UNLOAD-RATIO: 1.0. The model does not allow for hot gas bypass or cycling. It is always in unloading operation.
5. Supply fan:
   a. SUPPLY-STATIC: 1.0"
   b. SUPPLY-EFF: 40%
   c. FAN-CONTROL: CONSTANT-VOLUME
6. HEAT-SOURCE: None. Due to the high process loads and high return air setpoint, the space requires cooling only.

3.1.3.1.4 Cooling System Sizing

The auto-sizing feature in DOE-2 is not reliable. Therefore, the model was run iteratively: first it was run to determine the peak cooling load then the equipment was manually sized at 150% of the peak load. The baseline and parametric runs were run with the manually-entered cooling equipment capacities. These values are listed below in Table 1. The proposed case was normalized to a 5 kW (~1.5 tons) unit for the lifecycle cost analyses.

Table 4: Cooling and Flow Capacity for Climate Zone 06

	eQuest peak loads	**Basecase and Proposed Cases: 150% oversized**
Total Cooling Capacity	77,500 Btu/h	116,300 Btu/h
Total Cooling Capacity	6.5 tons	9.7 tons
Total flow	3,460 CFM	3,460 CFM

3.1.3.2 System Properties for Parametric Runs

Two parametric runs were set up in eQuest. The hourly reports from these runs were used to model the three proposed cases. The first parametric run modeled the system during unoccupied mode. The second run modeled the system during occupied mode.

The proposed runs use the same cooling capacity and CFM as basecase, which are shown in Table 1. In reality, the cooling capacity of the proposed case would be higher since it is a VAV unit that serves the whole building. However, the compressor efficiency is assumed to be constant, meaning it cycles to meet the load, thus there is no need to oversize it in the model.

3.1.3.2.1 Parametric Run 1: Unoccupied Mode

1. OA-CONTROL: DUAL-TEMP. Adds an economizer that operates whenever the outside air temperature is lower than the return air temperature.
2. ECONO-LOCKOUT: NO. The economizer is fully integrated, thus it can operate at the same time as the compressor.
3. SUPPLY-KW/FLOW: 0.00004 kW/CFM. A typical VAV unit might be designed for 30,000 CFM, 4” total static and 60% fan efficiency. This comes out to about 0.8 W/CFM. If the computer rooms are 10% of the total CFM then following the ideal fan laws the fan power when serving computer rooms is only 0.1% of the design power. However, this assumes perfect reset of static pressure setpoint which is not realistic. Also motor and variable speed drive efficiencies decrease at very low load. A conservative estimate of fan power at 10% speed is 5% of design power or about 0.04 W/CFM.
4. SUPPLY-DELTA-T: 0.1236°F, which is 3090 times the supply kW/CFM, per eQuest’s Dictionary entry for SUPPLY-DELTA-T.

3.1.3.2.2 Parametric Run 2: Occupied Mode

1. OA-CONTROL: DUAL-TEMP. Adds an economizer that operates whenever the outside air temperature is lower than the
2. ECONO-LOCKOUT: NO. The economizer is fully integrated, thus it can operate at the same time as the compressor.

3. SUPPLY-KW/FLOW: 0.00024 kW/CFM. To determine this value, it is assumed that the average fan speed is 60% and the average fan power is 30% of the peak power. The peak power occurs when the fan is 60% efficient and has 4" w.c. of static pressure.
4. SUPPLY-DELTA-T: 0.7416°F, which is 3090 times the supply kW/CFM, per eQuest's Dictionary entry for SUPPLY-DELTA-T.

3.1.3.3 Spreadsheet Post-Processing of eQuest Results

Ideally, a small computer room within an office building would be served by a central system, such as a packaged VAV system, when it has available capacity and by a separate system, such as a split DX, when the central system does not have available capacity. However, DOE-2 cannot model a controls sequence that would dictate this kind of operation. In order to simplify the analysis and avoid attempting to "fake" this controls sequence, only after-hours savings and occupied economizer hours were considered.

The hourly output data from the parametric run were filtered for certain hours to create the three proposed cases. The basecase data was filtered for the same hours for each case. Note that hours for all cases were filtered according to daylight saving time as well

3.1.3.3.1.1 Proposed Case 1

Parametric Run 1 (Unoccupied Mode) was filtered to include unoccupied hours only.

3.1.3.3.1.2 Proposed Case 2

Parametric Run 1 (Unoccupied Mode) was filtered to include unoccupied hours only, then further filtered to only include hours when the economizer is operating, which occurs when the outside air temperature is lower than the return air temperature.

3.1.3.3.1.3 Proposed Case 3

Parametric Run 2 (Occupied Mode) was filtered to include occupied hours only, then further filtered to only include hours when the economizer is operating, which occurs when the outside air temperature is lower than 75°F. This was added to the unoccupied hours output from Proposed Case 1 to create a comprehensive look at daytime and nighttime energy savings. The assumption for this case is that the VAV system serving the office spaces will have sufficient capacity to also serve the computer room when the outside air temperature is low and the load is low.

3.1.3.4 Energy Results

The energy savings were normalized from the 6.5 ton peak cooling load as calculated in the eQuest model to a 5 kW (~1.5 tons) unit. This is a reasonable sized unit to serve an individual computer room. The energy savings between the basecase and the proposed cases are summarized in Tables 2, 3 and 4. The space cooling savings are 100% in Proposed Case 2 because the cooling load is satisfied by the economizer.

Table 2: Annual End-Use Energy Consumption for Proposed Case 1, normalized to 5 kW (~1.5 ton) unit [kWh/year]

	Space cooling [kWh/yr]	**Fans [kWh/yr]**	**HVAC total [kWh/yr]**
Basecase	2,400	1,000	3,400
Proposed Case 1	210	140	350
% Savings vs basecase	*91%*	*86%*	*90%*

Table 3: Annual End-Use Energy Consumption for Proposed Case 2, normalized to 5 kW (~1.5 ton) unit [kWh/year]

	Space cooling [kWh/yr]	**Fans [kWh/yr]**	**HVAC total [kWh/yr]**
Basecase	1,700	800	2,500
Proposed Case 2	0	100	100
% Savings vs basecase	*100%*	*86%*	*95%*

Table 4: Annual End-Use Energy Consumption for Proposed Case 3, normalized to 5 kW (~1.5 ton) unit [kWh/year]

	Space cooling [kWh/yr]	**Fans [kWh/yr]**	**HVAC total [kWh/yr]**
Basecase	5,200	2,000	7,200
Proposed Case 3	680	980	1,700
% Savings vs basecase	*87%*	*51%*	*77%*

3.1.3.5 Incremental Installed Cost

The incremental cost used for the addition of one VAV terminal to serve a computer room is $3,129. This incremental cost is the full cost of the VAV box and associated controls. This cost was determined by using real unit prices from 7 HVAC and controls contractors covering recently bid projects in the Bay Area and Los Angeles. The unit price breakdown is shown in Table 5.

This lifecycle cost analysis does not include maintenance costs, since the lifetime of the unit is 15 years, which is equal to the time period for the analysis.

Table 5: Unit pricing breakdown for Incremental Cost of VAV box and controls – based on seven recently bid projects

Unit Pricing from Mechanical Contractor:	
Installed cost for cooling only VAV box, including connection to main duct and outlet sound plenum	$ 1,021
Installed cost for above-ceiling supply air outlet, including volume damper and flexible ductwork to terminal box	$ 339
Unit Pricing from Controls contractor:	
Controls for cooling only VAV box	$ 1,065
Estimated EMCS cost for central system programming	$5,000
Estimated number of zones over which central programming is spread	10
Total cost to General Contractor	$ 2,925
GC markup	7%
Total cost to owner	**$ 3,129**

3.1.3.6 Maintenance Cost

Incremental maintenance cost data was provided by a Bay Area mechanical contractor and service contractor. It is based on approximately ½ hour of service time per unit each year at a labor rate of $100/hr.

3.1.3.7 Lifecycle Cost Results

As shown in Table 5, the measure is highly cost effective.

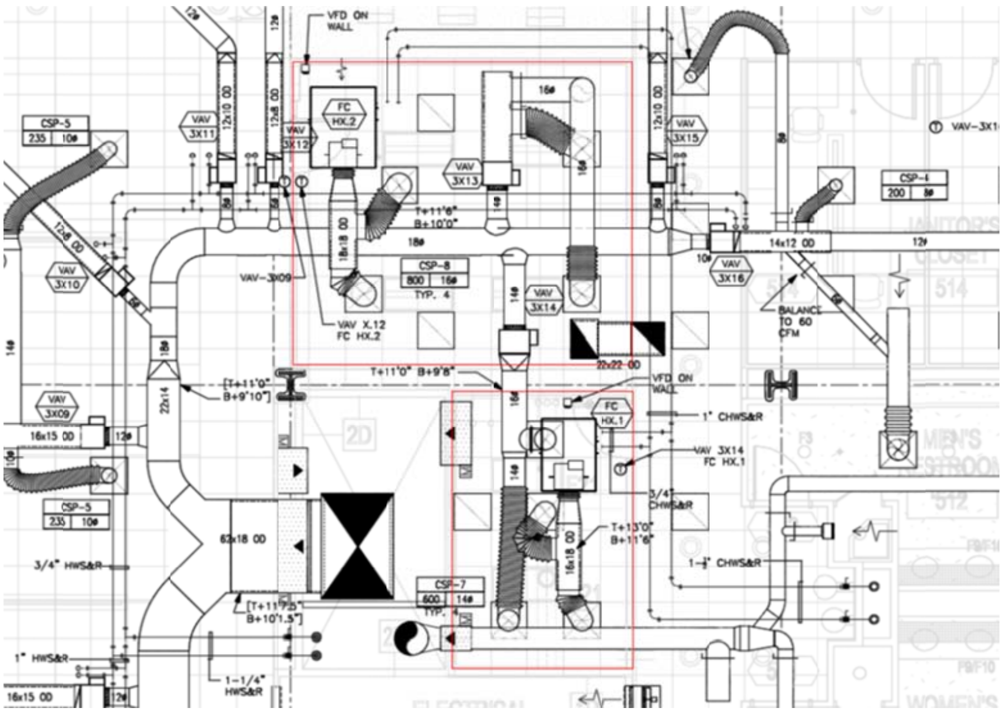
Table 5. Lifecycle Cost Results to Add VAV Box to Small Computer Room (CZ06)

Incremental Installed Cost	$ 3,129
Incremental Annual Maint.	$ 50
NPV of Annual Maint.	$ 597
Total Incremental Cost	$ 3,726
NPV of Energy Savings	$ 9,478
Lifecycle cost savings	$ 5,753
1st yr energy savings	$ 797
Simple payback (yrs)	4.7

3.1.3.8 Example Installation

Figure 12. is a section of a design drawing for a recently completed office building in Pleasanton, CA. Highlighted are two computer rooms that are served by chilled water fan coils and by VAV boxes.

Figure 12. Two Computer Rooms Served by Fan Coils and VAV Boxes



3.1.4 Large Data Center – Water Economizer

For very large data centers waterside economizing is likely to be lower first cost than airside economizing. Therefore an analysis was performed comparing a large data center without economizing to one with a waterside economizer. Note that airside economizing may well have a lower lifecycle cost than waterside economizing for large data centers but as long as waterside economizing is cost effective then the measure is justified.

While it is expected that chilled water air handlers (CRAHs) will be required to be variable volume it is possible that the variable volume fan requirement will not be adopted into Title 24. Therefore, the waterside economizer analysis was performed with both constant speed and variable speed CRAH units.

3.1.4.1 Energy Analysis

A 10,000 square feet single-zone datacenter building was modeled using the eQuest Design Day version to evaluate annual energy performance of waterside economizer.

The eQuest model has a 100 feet by 100 feet floor plan, with a floor to ceiling height of 12 feet and a 3 feet plenum space above the ceiling. The building's envelope was modeled to have R-10 wall with no windows and/or doors, adiabatic roof and floor. The space was modeled to have zero occupancy and 0.5 w/sf uniform lighting load. The envelope and non-IT cooling load was simplified in the energy model because its values are small enough to be negligible comparing to the IT load.

The space IT equipment load was modeled to be 100 w/sf. The IT load part-load schedule was modeled to be 24 x 7 each week with the load being constant during each month but varying from month to month. The following table listed the IT load schedule for each month.

Table 6 IT load schedule

Month	IT Load Fraction
Jan, May, Sep	25%
Feb, Jun, Oct	50%
Mar, Jul, Nov	75%
Apr, Aug, Dec	100%

Two basecase models, i.e. basecase A and basecase B, were established. In the basecase A model, a constant volume air system was modeled. In the basecase B model, a variable volume air system with a minimum air flow rate of 50% of design flow was modeled.

The system air flowrate, cooling coil size, chiller and tower capacities were calculated based on the assumed static space peak IT loads. The system and water loop temperature differences were assumed to be the following typical values: for the air system, the temperature difference was assumed to be 20 ^{o}F, chilled water loop 18 ^{o}F and CW loop 11.5 ^{o}F. Fan, pump, chiller and tower efficiencies were also assumed to be typical values listed in the table below.

To study the effect of the waterside economizer (WSE), a waterside economizer was added to each of the basecase model. The waterside economizer capacity was assumed to be the total capacity of the two chillers. The waterside economizer was assumed to have 3 ^{o}F approach.

Detailed system, zone, and plant assumptions for basecase A are summarized in the following tables. Inputs for basecase B and WSE models that are different from basecase A are noted at the end of each table.

Table 7 Zone assumptions in the datacenter energy model

ZONE

TYPE	CONDITIONED
FLOW/AREA	0
OA-FLOW/PER	0
MIN-FLOW/AREA	0
DESIGN-HEAT-T	55
DESIGN-COOL-T	80
THERMOSTAT-TYPE	REVERSE-ACTION
THROTTLING-RANGE	0.5
SIZING-OPTION	ADJUST-LOADS

Table 8 System assumptions in the datacenter energy model

SYSTEM	
TYPE	VAVS
HEAT-SOURCE	NONE
BASEBOARD-SOURCE	NONE
ZONE-HEAT-SOURCE	NONE
SIZING-RATIO	1
SUPPLY-FLOW	155,900
COOLING-CAPACITY	3,562,600
MIN-SUPPLY-T	60
COOL-SET-T	60
COOL-CONTROL	WARMEST
COOL-MIN-RESET-T	60
RESET-PRIORITY	SIMULTANEOUS
COOL-MAX-RESET-T	75
MIN-RESET-FLOW [1]	1.0
MIN-OUTSIDE-AIR	0
OA-CONTROL	FIXED
FAN-CONTROL [3]	CONSTANT-VOLUME
SUPPLY-STATIC	1.25
SUPPLY-EFF	0.585
MOTOR-PLACEMENT	IN-AIRFLOW
FAN-PLACEMENT	DRAW-THROUGH
NIGHT-CYCLE-CTRL	CYCLE-ON-ANY
MIN-FLOW-RATIO [2]	1
CHW-COIL-HEAD	15
CHW-VALVE-TYPE	TWO-WAY
CHW-LOOP	CHW Loop
COOL-CTRL-RANGE	0.1

6.
7. Note:
   1. For basecase B and WSE+B, MIN-RESET-FLOW = 0.5

2. For basecase B and WSE+B, MIN-FLOW-RATIO = 0.5
3. For basecase B and WSE+B, FAN-CONTROL = SPEED

Table 9 Pump assumptions in the datacenter energy model

	CHWP	CWP
FLOW	198	356
HEAD [1]	85	48
NUMBER	2	2
MOTOR-CLASS	PREMIUM	PREMIUM
CAP-CTRL	VAR-SPEED-PUMP	VAR-SPEED-PUMP
MIN-SPEED	0.2	0.2
HEAD-RATIO	1	1

8. Note:

1. In the WSE models, the CHWP head was increased to be 95'

Table 10 Water loop assumptions in the datacenter energy model

	CHW LOOP	CW Loop
LOOP-DESIGN-DT	18	11.5
LOOP-OPERATION	DEMAND	DEMAND
SIZING-OPTION	SECONDARY	SECONDARY
DESIGN-COOL-T	45	73
COOL-SETPT-CTRL	LOAD-RESET	LOAD-RESET
LOOP-SETPT-RNG	0.1	0.1
MAX-RESET-T	65	73
MIN-RESET-T	45	55
PIPE-HEAD	43	10
START-WSE-WB [1]		Not used in basecase
WSE-SETPT [2]		Not used in basecase

9. Note:

1. In WSE+A and WSE+B models, START-WSE-WB = 57;
2. In WSE+A and WSE+B models, WSE-SETPT = 50

Table 11 Chiller assumptions in the datacenter energy model

CHILLER	
TYPE	ELEC-HERM-CENT
RATED-CHW-T	45
RATED-COND-T	73
RATED-CW-FLOW	2.38
RATED-CHW-FLOW	1.32
SPECIFIED-AT	RATED-CONDITIONS
CAPACITY	1.7957

MIN-RATIO	0.2
HGB-RATIO	0.15
VARIABLE-SPEED	YES
ELEC-INPUT-RATIO	0.139
CHW-LOOP	CHW Loop
CHW-HEAD	17
CHW-MAX-FLOW	1.3
CONDENSER-TYPE	WATER-COOLED
CW-LOOP	CW Loop
CW-HEAD	17
CW-FLOW-CTRL	VARIABLE-FLOW
CW-MIN-FLOW	0.3
MAX-COND-T	77

Table 12 Tower assumptions in the datacenter energy model

TOWER	
TYPE	OPEN-TWR
CAPACITY	3.873
ELEC-INPUT-RATIO	0.0045
NUMBER-OF-CELLS	2
CAPACITY-CTRL	VARIABLE-SPEED-FAN
CELL-CTRL	MAX-CELLS
RATED-RANGE	11.5
RATED-APPROACH	5
RATED-WETBULB	68
MAX-FLOW/CELL	2
MIN-FLOW/CELL	0.5
MIN-VFD-SPEED	0.1
CW-LOOP	CW Loop
CW-HEAD	10
CW-STATIC-HEAD	10

10.

11. For the WSE models, a WSE is defined as following in the energy model.

Table 13 WSE assumptions in the datacenter energy model

WSE	
TYPE	WATER-ECONOMIZER
CAPACITY	3.5914
RATED-CHW-FLOW	1.33
RATED-CHW-DT	18

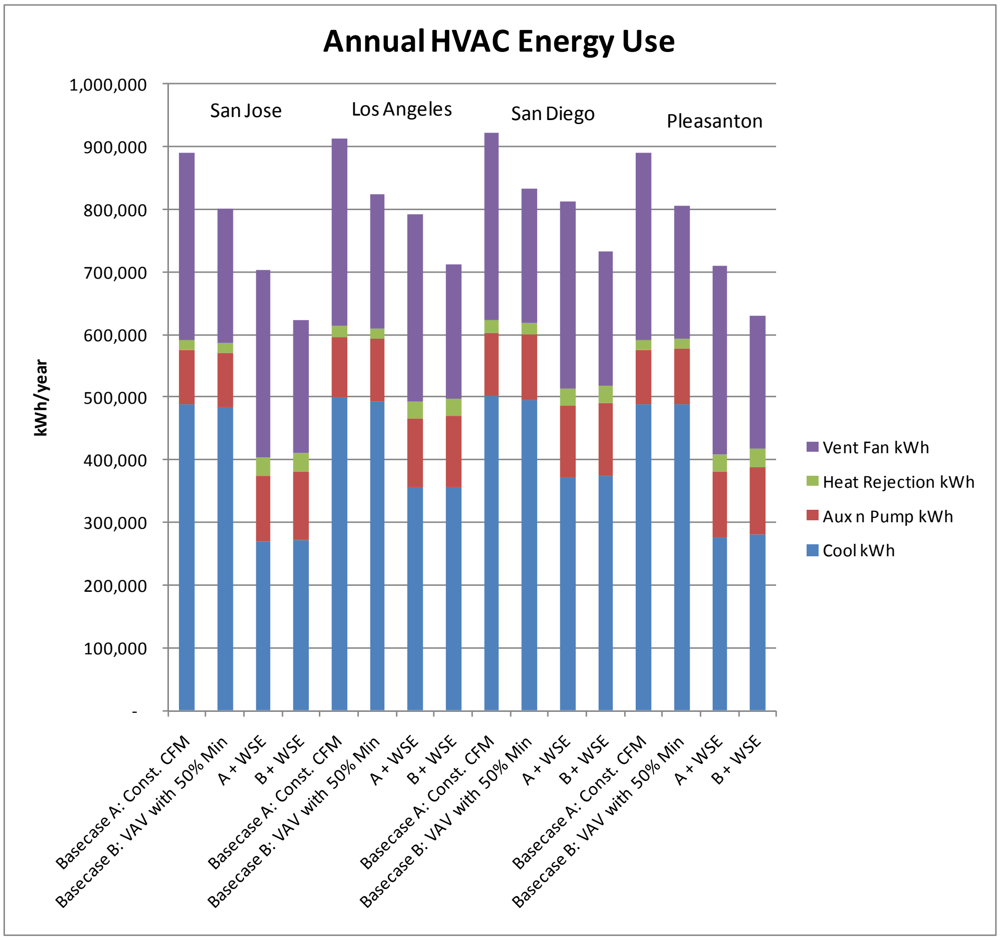
RATED-CW-FLOW	2.09
RATED-CW-DT	11.5
CHW-LOOP	CHW Loop
CHW-HEAD	13.9
CHW-FLOW-CTRL	VARIABLE-FLOW
CW-LOOP	CW Loop
CW-DT	11.5
CW-HEAD	13.9
CW-FLOW-CTRL	VARIABLE-FLOW
RATED-WSE-TD	21
MIN-WSE-TD	3

3.1.4.2 Energy Results

	Per Ton of Plant Capacity				
	Cool kWh	Aux n Pump kWh	Heat Rejection kWh	Vent Fan kWh	Bldg Total kWh
CZ04-San Jose					
Basecase A: Const. CFM	1,630	287	54	998	21,395
Basecase B: VAV with 50% Min	1,611	294	53	711	21,095
A + WSE	897	351	96	998	20,768
B + WSE	909	363	97	711	20,506
CZ06-Los Angeles					
Basecase A: Const. CFM	1,663	323	59	998	21,470
Basecase B: VAV with 50% Min	1,644	331	59	712	21,171
A + WSE	1,185	368	87	998	21,064
B + WSE	1,190	379	88	712	20,796
CZ07-San Diego					
Basecase A: Const. CFM	1,673	338	65	998	21,500
Basecase B: VAV with 50% Min	1,653	345	64	713	21,202
A + WSE	1,243	376	92	998	21,136
B + WSE	1,249	388	93	713	20,869
CZ12-Pleasanton/Sacramento					
Basecase A: Const. CFM	1,630	287	54	998	21,395
Basecase B: VAV with 50% Min	1,629	293	53	712	21,113
A + WSE	922	348	94	998	20,788

B + WSE	938	358	95	712	20,529

Figure 13. Annual HVAC Energy Use for 300 ton system with and without VAV and with and without waterside economizer in 4 climate zones.



3.1.4.3 Incremental Installed Cost

Incremental cost data was provided by 7 mechanical and controls contractors who provided alternate pricing on waterside economizers at two recent data center projects where the waterside economizer was bid as an add alternate. In both cases the central plant served both office and data center spaces thus the economizer was not sized for the full capacity of the plant. These costs include installation and commissioning.

	Pleasanton	Los Angeles	Average
data center tons	110	2000	
Incremental Costs:			
HVAC	$ 36,200	$ 53,000	
controls	$18,485	$6,000	
GC costs and markup	$4,000	$5,000	
Total Incremental Cost	$ 58,685	$ 64,000	
tons of HX capacity	190	660	425
$/ton of HX capacity	$ 310	$ 97	$ 203

Note that no incremental cost was included to increase cooling tower size beyond what would normally have been selected without a waterside economizer.

The recommended language for the Standard includes new wording for sizing waterside economizers serving data centers. Currently the Standard requires a water economizer to be capable of meeting 100% of the expected cooling load at 50°F drybulb / 45°F wetbulb. This is no problem for an office building where the expected load at these low ambient conditions is a small fraction of the design load. For a data center, however, the expected cooling load is dominated by IT load and therefore can be quite high at low ambient conditions. Therefore the recommendation language relaxes the ambient conditions to 40°F drybulb / 35°F wetbulb for a data center. (Note that the heat exchanger and cooling tower sizing assumptions in the simulation analysis are consistent with these criteria.)

A number of heat exchanger and tower selections were evaluated to insure that these are reasonable sizing criteria. The sizing criteria can be met with an air-air HX with a 3 degree approach and does not require increasing the tower size. With decent airflow management design supply air temperature can easily be above 60°F and design CHW supply temperature can easily be above 45°F. Accounting for reductions in ventilation and envelope load and accounting for redundancy in air handlers the CHWST could easily be 46-48F degrees to meet the expected cooling load at 35F WB. A reasonable heat exchanger selection will have a 2 to 3 degree approach and a reasonable tower selection could easily have a 7 to 11F degree approach at 35F. Note that the tower load for economizer sizing is less than the design cooling load due to reduced ventilation and envelope load, load diversity, and elimination of chiller heat. Chiller heat alone is typically 15-18% of design tower load. A tower selected for a 4-5F approach and 10F range at 75-78F WB can achieve a 7-10F approach at 35F WB at 80-85% of design load and 10-13F range.

3.1.4.4 Maintenance Cost

Incremental maintenance cost data was provided by a Bay Area mechanical contractor and service contractor. It is conservatively estimated to be 20 hours per year at a labor rate of $100/hr or about $2,000 per year.

3.1.4.5 Lifecycle Cost Results

	Per Ton of Plant Capacity					
	TDV Energy Savings $	**Incremental Cost**	**NPV of Incremental Maint.**	**Total Incremental Cost**	**Lifecycle Cost Savings**	**Simple Payback (yrs)**
CZ04-San Jose						
Basecase A: Const. CFM						
Basecase B: VAV with 50% Min						
A + WSE	$ 978	$ 203	$ 80	$ 283	$ 696	3.4
B + WSE	$ 1,486	$ 204	$ 80	$ 284	$ 1,203	2.3
CZ06-Los Angeles						
Basecase A: Const. CFM						
Basecase B: VAV with 50% Min						
A + WSE	$ 633	$ 203	$ 80	$ 283	$ 350	5.3
B + WSE	$ 1,130	$ 204	$ 80	$ 284	$ 846	3.0
CZ07-San Diego						
Basecase A: Const. CFM						
Basecase B: VAV with 50% Min						
A + WSE	$ 595	$ 203	$ 80	$ 283	$ 313	5.6
B + WSE	$ 1,101	$ 204	$ 80	$ 284	$ 817	3.1
CZ12-Pleasanton/Sacramento						
Basecase A: Const. CFM						
Basecase B: VAV with 50% Min						
A + WSE	$ 915	$ 203	$ 80	$ 283	$ 633	3.7
B + WSE	$ 1,392	$ 204	$ 80	$ 284	$ 1,109	2.4

3.1.4.6 Example Installation

Figure 14. Sample Waterside Economizer Plan View – Pleasanton Site

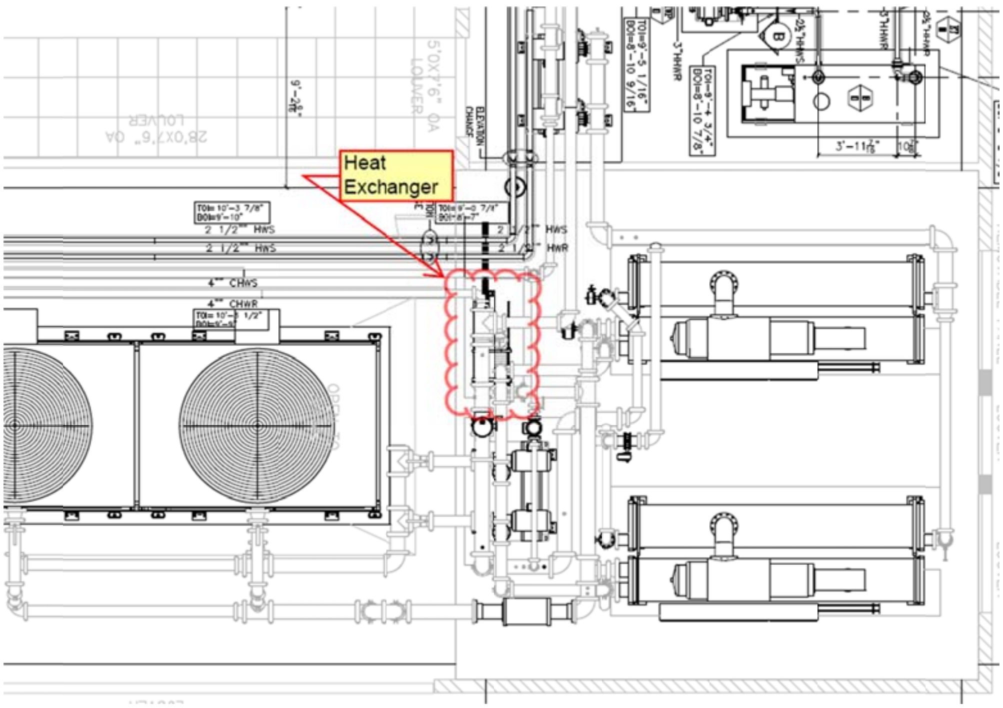


Figure 15. Sample Waterside Economizer Installation - Pleasanton

Figure 16. Sample Waterside Economizer Piping Schematic – Los Angeles Site

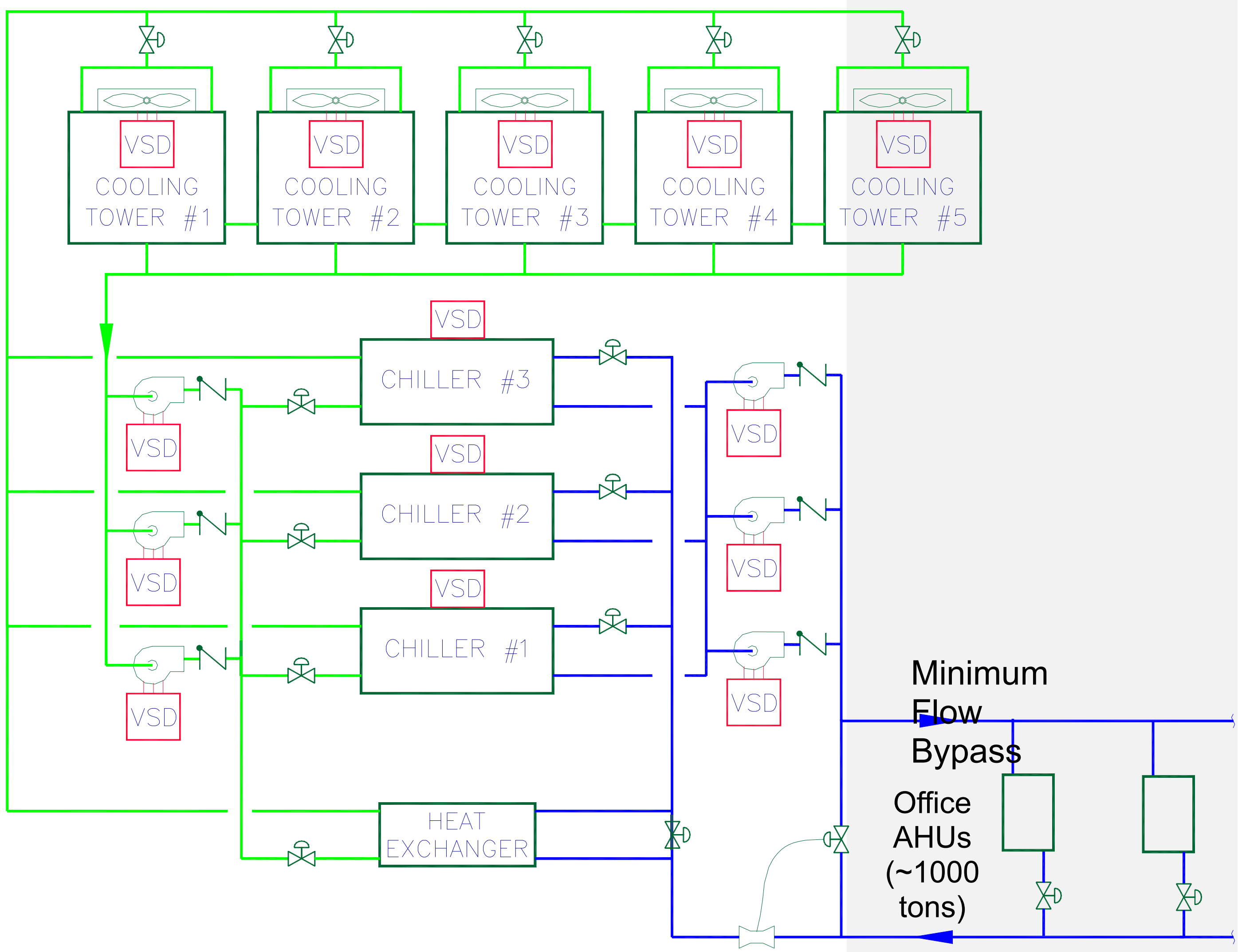


Figure 17. Waterside Economizer – Los Angeles Site

3.2 Humidity Control Limitations

As recently as 3 years ago it was quite common to include humidification (e.g. steam humidifiers) and dehumidification (e.g. electric reheat coils) in data center designs. One of the reasons for humidity control was the ASHRAE Technical Committee (TC) 9.9 recommendation in 2004 that data centers be maintained between 40 and 55% relative humidity. (Ironically it is impossible to maintain relative humidity in this range because relative humidity is relative to the temperature which varies considerably – e.g. if the cold aisle is 65^{o}F and 50%RH and the hot aisle is 85^{o}F then the hot aisle will be 25% RH.). In 2008 TC 9.9 expanded their recommended range to 41.9^{o}F dewpoint at the low end and 60% RH and 59^{o}F dewpoint at the high end. There is no published research supporting the need for humidity control in data centers.

Today most new data centers in California do not have any humidity controls. Some new data centers still have non-adiabatic humidifiers but almost no data centers have reheat for dehumidification.

The common explanation for humidification was electrostatic static discharge. It is true that the voltage of electrostatic discharge from people is higher at lower humidity levels (anyone who has walked across a carpet on a cold winter day knows this). However, it is also true that any CE-rated computer is immune from any charge level that a person can generate during normal activity. CE is the European Union testing standard (Standard IEC61000-4-2) that tests for ESD immunity. Essentially all computers today are CE-rated.

According to the Electrostatic Discharge Association, it is also true that while relative humidity will affect the charge level a person can generate, there is basically no humidity level at which a person will NOT create a charge that can damage circuit boards or components, i.e. if you open the CE-rated enclosure then you will damage the components even at high humidity. The only way to prevent damage to components is with personnel grounding practices (e.g. wrist straps). At high humidity levels and low charge levels a person will not be able to feel it when they create an electrostatic discharge but it is there nevertheless. Thus humidity control alone cannot reduce the risk of damage to equipment but it can create a false sense of security that might lead to lax grounding practices and therefore humidity control could increase the risk of damage from ESD. For this reason ANSI/ESD S20.20-2007 (Protection of Electrical and Electronic Parts, Assemblies and Equipment) does not allow humidification as a primary control of ESD. This standard provides administrative and technical requirements for establishing, implementing, and maintaining an ESD Control Program to protect electrical or electronic parts, assemblies, and equipment susceptible to ESD damage from Human Body Model (HBM) discharges greater than or equal to 100 volts.

The explanation for dehumidification from TC 9.9: "… conductive anodic filament (CAF) growth is strongly related to relative humidity. As humidity increases, time to failure rapidly decreases. Extended periods of relative humidity exceeding 60% can result in failures…" The guideline goes on to say "For short periods of time it should be acceptable to operate outside this recommended envelope and approach the extremes of the allowable envelope."

With reasonable airflow management circuit board temperatures in data centers should exceed 85^{o}F. To exceed 60% RH at 85^{o}F the dew point must be above 69^{o}F. This rarely occurs in any California climate and will not occur inside any data center that has mechanical cooling and more than 20 W/ft^2 of IT load (computer rooms are defined as having over 20 W/ft^2 of load). The mechanical cooling automatically reduces the inside dewpoint. Typical new data center IT loads range from 100-500 W/ft^2.

Furthermore, a survey of computer server specifications reveals that no manufacturers require relative humidity levels below 80%. The Cisco UCS 5108 Server, for example, lists 5-93% RH and the Dell PowerEdge M605 Blade Server, lists 8-80% RH.

Direct evaporative humidification is a well established technique for maintaining data center humidity. Figure 18 is from a large co-lo facility that has been operating with direct evaporative humidification for over 10 years without any significant problems.

Figure 18. Data Center with Direct Evaporative Humidification

3.3 Fan Power Limitation

Title 24 §144(c) has fan-power limitations (in watts/cfm) based on built-up ducted overhead systems with terminal units. Computer rooms typically have less pressure drop (more close coupled) and operate longer hours. Therefore, lower maximum fan power can be justified.

The proposed fan power limitation for computer rooms is stated in watts per Btuh, rather than in watts/cfm because this is a more accurate measure of fan system efficiency. The goal of the system is to deliver cooling in Btuh, not CFM. With most comfort cooling applications the design ΔT is around 20°F. With data centers, however, the design ΔT can range from 10°F to 40°F. A system could have a good W/cfm but be very inefficient because it uses a very low ΔT. Conversely, a system could have a poor W/cfm but be very efficient becaust it uses a high ΔT. One of the most important factors for determining the ΔT is the degree of containment. A system with good airflow management can be designed for a higher ΔT than one without good airflow management.

The proposed fan power limit is 27 watts per kBtuh of net sensible cooling capacity. It is based on the following conservative estimates: 20ºF ΔT, 2.5" total pressure, 55% fan efficiency and 90% motor/drive efficiency. It can also be met at 3" total pressure and 65% fan efficiency and many other combinations of ΔT, total pressure and efficiency.

Taylor Engineering has surveyed a number of actual data centers or data center designs and found that all but one met the proposed limitation. See Table 14. The one that did not meet it was a water-cooled DX unit with a condenser water economizer coil in series with the DX coil. Without the economizer coil the same unit complies. Note that a waterside economizer coil in a DX unit is very different from a plant waterside economizer. A plant economizer is a heat exchanger in the plant. It does not add any extra pressure drop to the fan system. Since it has already been shown above that airside economizers and plant waterside economizer coils are cost effective it can reasonably be argued that there is no real incremental cost for the proposed fan power limitation. The value of the limitation is to prevent designers from simply adding an economizer coil to water-cooled DX units in order to meet the economizer requirement, rather than properly designing the system with an airside economizer or a plant waterside economizer.

Table 14. Fan Power in W/kBtuh for a sample of actual data centers

	BHP	kW	CFM	TSP	fan effic	dT	sens. Btuh	W/kBtuh	w/cfm	Complies?
Liebert DH380A	10.0	7.45	15,200	2.55	61%	20.0	334,400	22	0.490	YES
Liebert DH380A with econo coil	10.0	7.45	14,250	2.78	62%	20.0	313,500	24	0.522	YES
Liebert DH380A with econo coil	13.8	10.29	15,200	3.45	60%	20.0	334,400	31	0.677	NO
Liebert DH267W	4.7	3.53	10,200	1.63	55%	20.0	224,400	16	0.346	YES
Liebert DH267W with econo-coil	7.5	5.59	10,200	2.5	54%	20.0	224,400	25	0.548	YES
Liebert CHW - FC fan	11.4	8.52	17,100	2.46	58%	20.0	376,200	23	0.498	YES
Liebert CHW - EC fan	9.7	7.20	17,100	2.46	69%	20.9	393,881	18	0.421	YES
Liebert XDV-10		0.18	1,000			24.3	26,730	7	0.180	YES
Huntair CHW	8.3	6.16	16,700	1.8	57%	22.6	414,244	15	0.369	YES
Stulz CHW	7.6	5.66	18,000	1.38	51%	21.8	432,036	13	0.315	YES
APC CHW - InRow		0.92	2,900				60,000	15	0.317	YES
Energy Labs - CHW ducted upflow	1.3	0.94	4,425	1.23	68%	21.1	102,850	9	0.212	YES
Energy Labs - CHW downflow	7.4	5.53	17,700	1.63	61%	21.1	411,401	13	0.312	YES
Team Air - InRow	0.8	0.63	6,150	0.6	69%		139,919	4	0.102	YES

The User's Manual will explain that the fan power limitation calculation can account for redundancy. It is common to have redundant fan coils or CRACs with variable speed drives (see section 3.4 for variable speed fan requirements). With variable speed fans all units can operate in normal mode at design conditions which reduces the fan speed and total static of each fan system. For example, a CRAC unit may be designed for 30 W/kBtuh but will operate at 20 W/kBtuh at peak load because it will not be running at full speed.

3.4 Fan Control

Two speed or variable speed control of all air conditioning units greater than or equal to 10 tons is required in Title 24-2008 with an effective date of 1/1/2012. This applies to non-computer applications and computer room applications. Computer rooms typically operate 24/7 and often have redundant fan systems. Therefore, two speed or variable speed fan control is generally much more cost effective in computer room applications and can be justified for smaller equipment. The proposal is to lower the threshold for DX equipment to systems > 5 tons and to all CHW fan systems.

3.4.1 DX Systems

Most of the major computer room air conditioner (CRAC) manufacturers already have variable speed fans either standard or optional on their CRAC units in this size range. And most also have either variable capacity compressors or multiple compressors in this range as well. See Table 15. In addition to the traditional CRAC unit manufacturers, several conventional air conditioner manufacturers also offer variable speed fans either standard or as an option. See Table 15.

Table 15. Currently available DX units with variable speed fans

Make/Model	Size range	Fan	Compressors	Comment
Liebert/DS	8t – 30t	Forward curve (beta testing electrically commutated (EC) fan)	Semi-hermetic with four-step or digital scrolls	$1800 for factory VFD.
Stulz CyberTwo	6t – 30t	EC fan optional (~$1700 add)	(2) scrolls	2 speed fan control?
Data Aire gforce	6t – 30t	EC fan standard	(2) scrolls (face split coil, not row split) (investigating variable speed	Currently accepts external speed signal (default min =

			compressor)	80%)
APC InRoom	5t – 20t	EC	(2) compressors above 8 tons	
Aaon	3t+	EC	Digital scroll standard; turbocor optional from 45-230t	EC fans and digital scroll are standard, no added cost
Carrier standard unit	5t+	Optional VFD	Multiple stages	~$2500 incremental cost for VFD on 5t units
Carrier Centurion 48PD	4 – 5t	VFD std	Digital scroll	No incremental cost for variable speed
Daikin	0.5 to 5t	EC	Variable speed scroll	
Mitsubishi	0.5 to 5t	EC	Variable speed scroll	
LG	0.5 to 5t	EC	Variable speed scroll	

In the analysis for DX units a conservative incremental cost of $3,000 per unit was used. This is based on the $1800 incremental cost for the Liebert/DS unit plus markup and incremental start-up and commissioning costs. This cost is conservative because the incremental cost is already at or close to zero for other manufacturers and will drop in the future for Liebert and other manufacturers due to mass production.

Incremental maintenance is conservatively estimated at $200/yr per unit. Again, this cost should be close to zero as variable speed units become more common.

Table 16 shows the lifecycle cost analysis for DX equipment. It is extremely conservative in that it assumes 2 speed fan control rather than variable speed fan control. It further assumes that the fan operates at 100% speed half of the time and only turns down to 60% speed and 30% fan power the rest of the time. Again, this is very conservative. For the given load profile a variable speed fan would provide significantly more savings. The computer room load is assumed to follow the computer room load profile proposed for ASHRAE Standard 90.1 and Title 24 performance compliance calculations (see section 5.3). The analysis is also conservative in that it only accounts for fan energy savings and does not account for fan heat savings which translates into compressor or chiller plant savings. Despite all these conservative assumptions the analysis shows that the measure is easily cost effective.

Table 16. Lifecycle Cost Analysis for DX Fan Control

DX CRAC 2 Speed Fan Savings			
tons	6	8	10
CFM/ton	500	500	500
CFM	3000	4000	5000
TSP	2.5	2.5	2.5
fan effic	50%	50%	50%
BHP	2.36	3.15	3.94
design kW	1.76	2.35	2.94
switch to low speed at % load	50%	50%	50%
low fan speed	60%	60%	60%
% fan power at low speed	30%	30%	30%
HVAC sizing ratio (incl. redundancy)	110%	110%	110%
hrs at 100% load	2190	2190	2190
fan speed at 100% load	100%	100%	100%
kW at 100% load	1.76	2.35	2.94
hrs at 75% load	2190	2190	2190
fan speed at 75% load	100%	100%	100%
kW at 75% load	1.76	2.35	2.94
hrs at 50% load or less	4380	4380	4380
fan speed at 50% load or less	60%	60%	60%
kW at 50% load or less	0.53	0.70	0.88
2speed annual fan energy (kwh/yr)	10,030	13,373	16,717
basecase annual fan energy	15,431	20,574	25,718
energy savings (kwh/yr)	5,401	7,201	9,001
peak power savings (kW)	-	-	-
avg TDV rate for 15yr life ($/kwh)	1.9	1.9	1.9
lifecycle energy savings ($)	$ 10,261	$ 13,682	$ 17,102
incremental maintenance ($/yr)	$ 200	$ 200	$ 200
NPV of incremental maintenance ($)	$ 2,380	$ 2,380	$ 2,380
lifecycle savings ($)	$ 7,881	$ 11,302	$ 14,722
Incremental first cost	$ 3,000	$ 3,000	$ 3,000
Cost Effective?	YES	YES	YES
simple payback (yrs)	4.5	3.2	2.4

DX CRAC 2 Speed Fan Savings			
tons	6	8	10
CFM/ton	500	500	500
CFM	3000	4000	5000
TSP	2.5	2.5	2.5
fan effic	50%	50%	50%
BHP	2.36	3.15	3.94
design kW	1.76	2.35	2.94
switch to low speed at % load	50%	50%	50%
low fan speed	60%	60%	60%
% fan power at low speed	30%	30%	30%
HVAC sizing ratio (incl. redundancy)	110%	110%	110%
hrs at 100% load	2190	2190	2190
fan speed at 100% load	100%	100%	100%
kW at 100% load	1.76	2.35	2.94
hrs at 75% load	2190	2190	2190
fan speed at 75% load	100%	100%	100%
kW at 75% load	1.76	2.35	2.94
hrs at 50% load or less	4380	4380	4380
fan speed at 50% load or less	60%	60%	60%
kW at 50% load or less	0.53	0.70	0.88
2speed annual fan energy (kwh/yr)	10,030	13,373	16,717
basecase annual fan energy	15,431	20,574	25,718
energy savings (kwh/yr)	5,401	7,201	9,001
peak power savings (kW)	-	-	-
avg TDV rate for 15yr life ($/kwh)	1.9	1.9	1.9
lifecycle energy savings ($)	$ 10,261	$ 13,682	$ 17,102
incremental maintenance ($/yr)	$ 200	$ 200	$ 200
lifecycle savings ($)	$ 7,881	$ 11,302	$ 14,722
Incremental first cost	$ 3,000	$ 3,000	$ 3,000
Cost Effective?	YES	YES	YES
simple payback (yrs)	4.5	3.2	2.4

3.4.2 CHW Systems

All computer room air handler (CRAH) manufacturers offer EC fans or variable speed drives as standard or standard options on their CRAH units. Since the proposed measure extends to all sizes of CHW units the cost effectiveness hurdle will be hardest to overcome at the very smallest sizes. Therefore, the analysis focuses on small fan coils that might be used to condition a small IDF closet. Small fan coil manufacturers include McQuay, Trane, Carrier, JCI/York, Williams and MagicAire. ECM motors are now a standard option from some fractional horsepower fan coil manufacturers. Furthermore, it is quite common and easy to add fractional horsepower VFD to fan coils in the field.

Incremental cost data was provided by two Bay Area suppliers of fan coil units with optional ECM motors. PSC motors are standard on these units. Incremental costs include incremental startup/commissioning costs. There is no incremental annual maintenance once the units have been commissioned.

Table 17 shows the lifecycle cost analysis results for CHW units. It shows that the measure is easily cost effective down for units below 1/12 horsepower.

Table 17. Lifecycle Cost Analysis for CHW Fan Control

MHP	1/12	1/8	1/4
BHP % of MHP	85%	85%	85%
BHP	0.07	0.11	0.21
design kW	0.05	0.08	0.16
min fan speed	50%	50%	50%
HVAC sizing ratio	120%	120%	120%
hrs at 100% load	2190	2190	2190
fan speed at 100% load	83%	83%	83%
kW at 100% load	0.03	0.05	0.09
hrs at 75% load	2190	2190	2190
fan speed at 75% load	63%	63%	63%
kW at 75% load	0.01	0.02	0.04
hrs at 50% load or less	4380	4380	4380
fan speed at 50% load or less	50%	50%	50%
kW at 50% load or less	0.01	0.01	0.02
proposed annual fan energy (kwh/yr)	124	186	372
basecase annual fan energy	463	694	1,388
energy savings (kwh/yr)	339	508	1,016
peak power savings (kW)	0.02	0.03	0.07
avg TDV rate for 15yr life ($/kwh)	1.9	1.9	1.9
lifecycle energy savings ($)	$ 643	$ 965	$ 1,930
Incremental motor cost	$ 185	$ 185	$ 140
contractor markup	30%	30%	30%
Add for start-up/commissioning	$ 100	$ 100	$ 100
Total incremental cost	$ 341	$ 341	$ 282
Cost Effective?	YES	YES	YES
simple payback (yrs)	6.3	4.2	1.7

3.5 Containment

Without containment a significant fraction of the HVAC supply air can bypass the computer servers and return to the HVAC unit – resulting in wasted fan energy and compressor energy. At the same time a significant fraction of hot air discharged from a server can recirculate back to the server inlet. This is referred to as a “hot spot”. High server inlet temperatures can increase server fan energy and cause servers to shut down. Hot spots do not mean that the HVAC system has insufficient cooling or airflow capacity but that airflow management is not adequate. When using air to cool high density data centers containment is often necessary to prevent hot spots.

Until fairly recently containment was not common. Now it is standard practice for new high density data centers. There are many forms of containment and many products available, including hot aisle containment, cold aisle containment, strip curtains, blanking panels, plastic panels, chimney racks, etc. (see Figure 19, Figure 20, Figure 21, Figure 22, Figure 23). The one thing all forms of containment have in common is that they largely prevent bypass and recirculation.

One of the main reasons that containment has become common is because it saves both energy and first cost for a new data center. Yes the curtains or other containment devices increase the first cost but the savings in reduced air distribution more than makes up for the added cost. Without containment it is necessary to duct supply air very close to the racks. Raised floor supply is often used without containment to effectively duct supply air to each rack. With containment, however, supply air does not need to be ducted to each rack. One of the most common systems now is to contain the hot aisles and simply provide supply air on one side of the room or in a couple locations if the room is large. Hot aisles are connected to a ceiling return plenum. Thus supply ductwork is eliminated and the raised floor can be eliminated.

Figure 19. Example Containment System

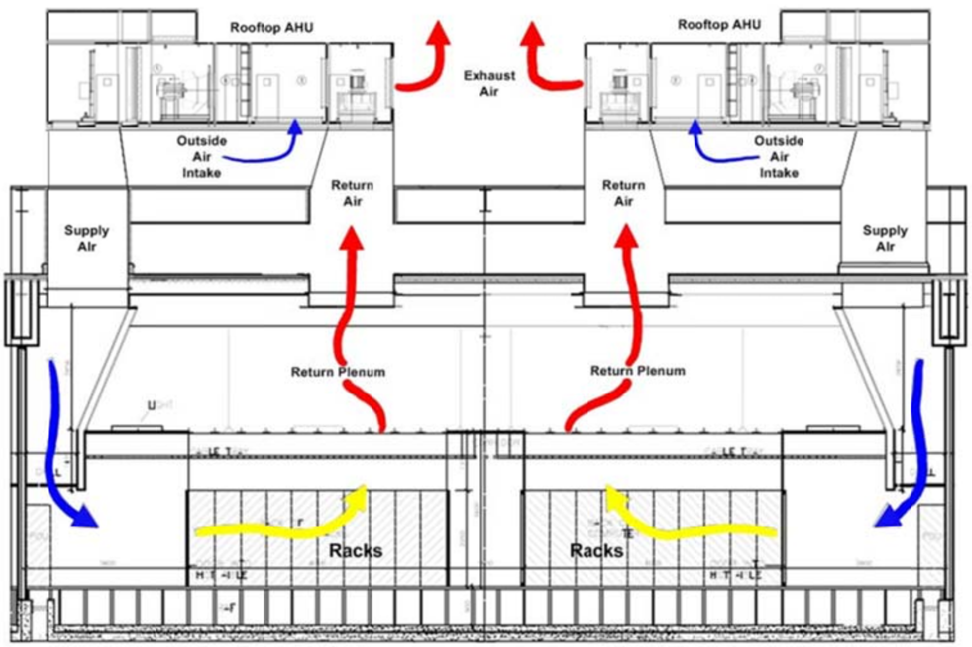


Figure 20. Example Containment System – Custom Hot Aisle Enclosures

Figure 21. Example Containment System – Standard Chimney Racks

Figure 22. Example Containment System – Standard Hot Aisle Enclosure

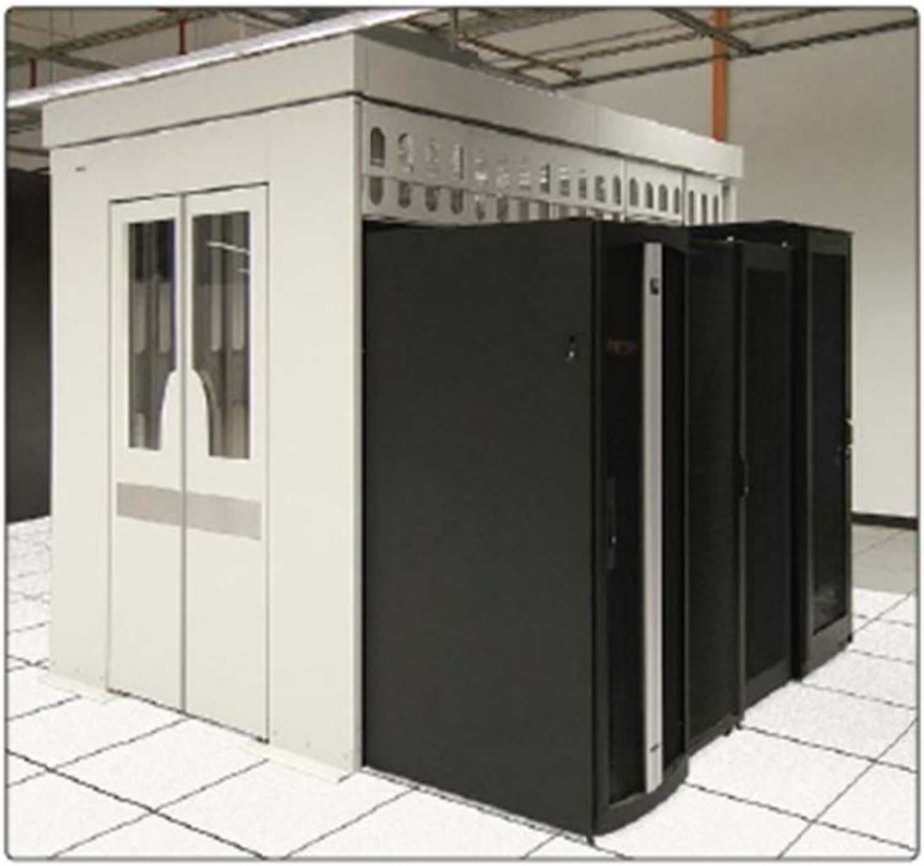
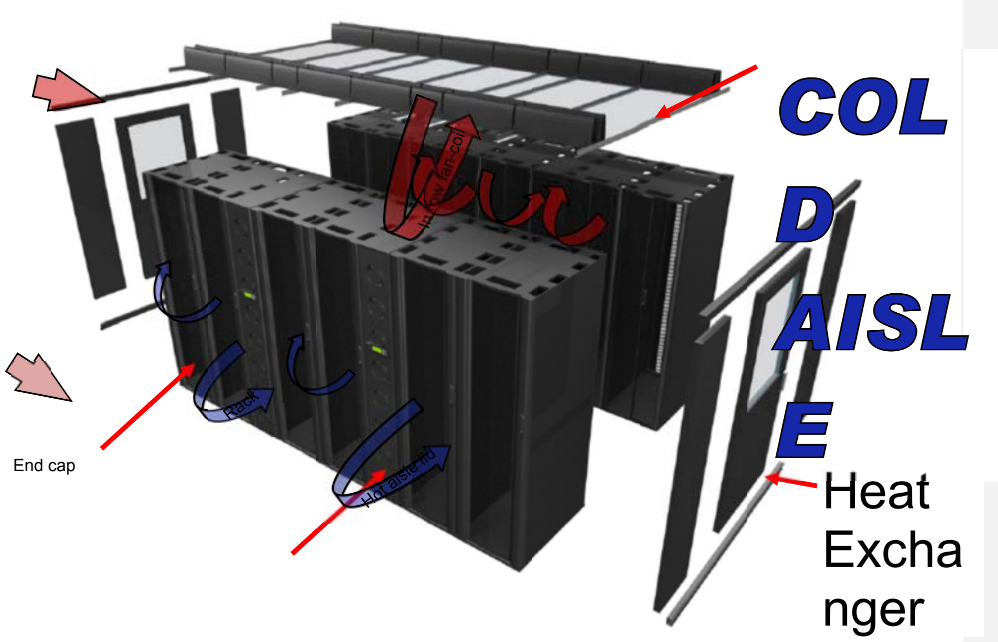


Figure 23. Containment Example – Hot Aisle End Cap and Lid

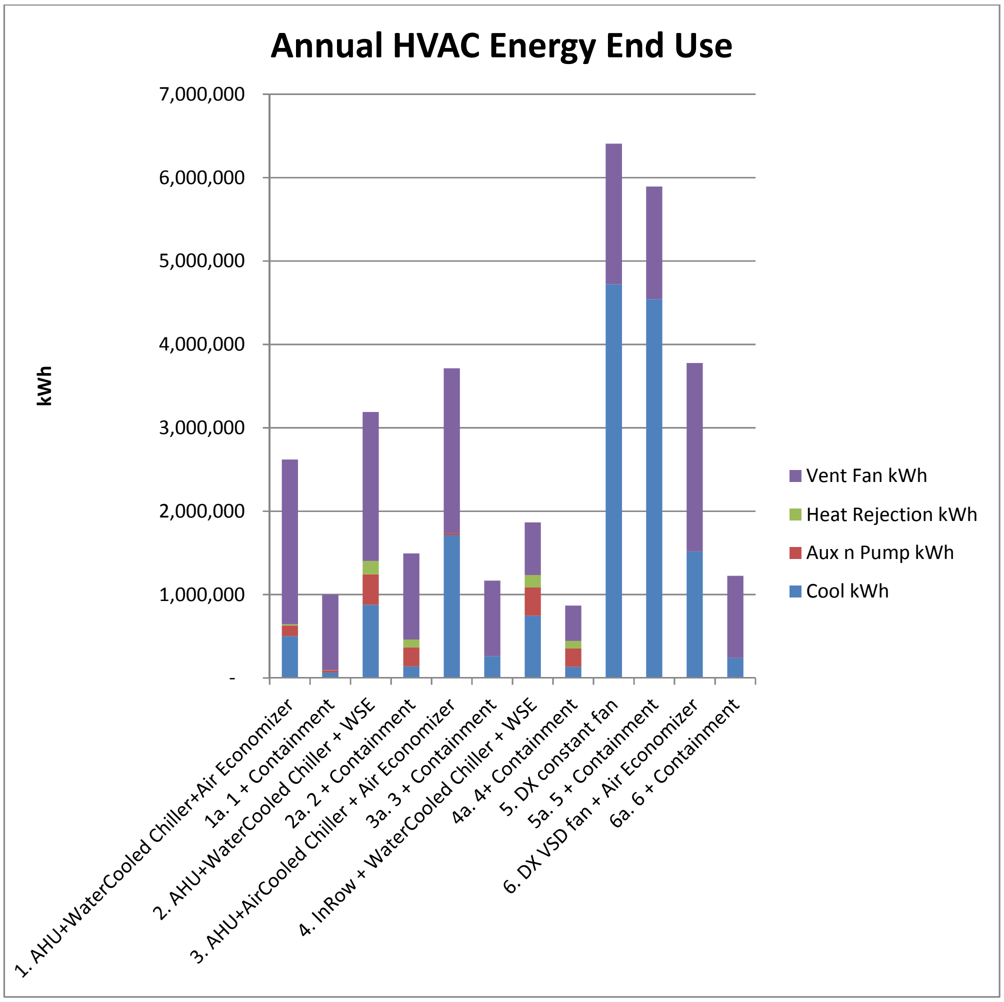


In addition to reduced ductwork costs, containment also reduces the first cost by allowing the HVAC airflow capacity to be reduced. Without containment it is necessary to design the HVAC system airflow rate to significantly exceed the server airflow rate. Thus it is common to design data center HVAC without containment for 15 to 20°F airside ΔT. With containment it is common to design for 25-30°F airside ΔT. This is a 20% to 50% reduction is fan sizing, duct sizing, etc.

The energy benefits of containment are dramatic, particularly for a system with an airside or waterside economizer. Containment allows not only significantly less supply air but allows the supply air to be delivered at significantly higher temperatures (e.g. 55°F without containment, 65°F with containment). Furthermore, the return air temperature is significantly higher (e.g. 75°F without containment, 95°F with containment). Higher supply and return temperatures allow greater economizer savings. It also reduces compressor lift by allowing higher supply air temperature for DX systems and higher chilled water temperatures for chilled water systems. Containment also allows better turndown of airflow rates at part load.

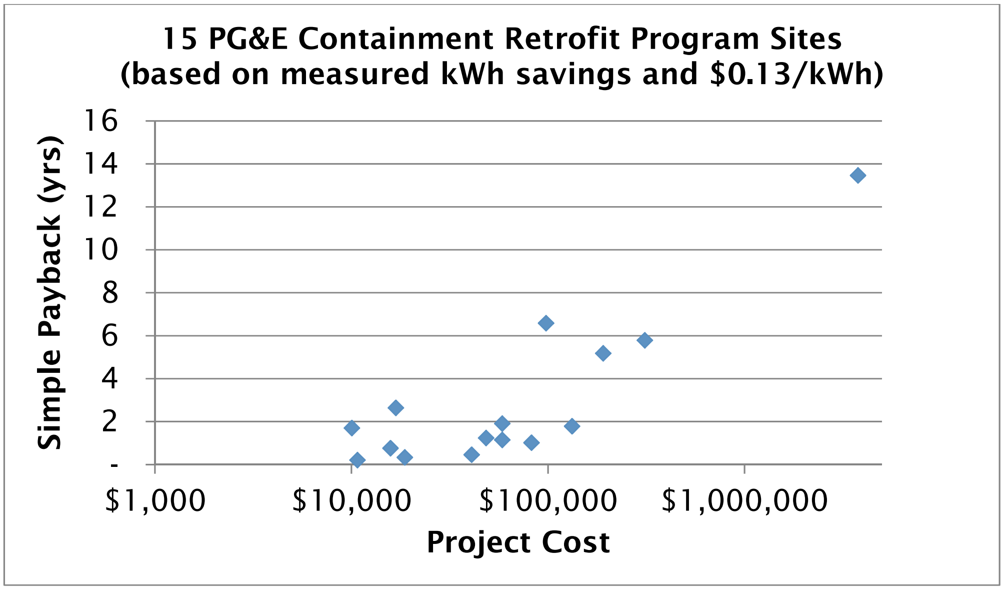
Figure 24 shows the HVAC energy of various system types with and without containment for a typical 20,000 ft^2, 100 W/ft^2 data center in San Jose. The HVAC energy use for a chilled water system with air economizer and containment (Option 1a), for example, is less than half the energy use of the same system without containment (Option 1).

Figure 24. Containment Savings for Various Data Center System Types



Retrofitting containment to existing data centers is obviously not as cost effective as containment for new data centers. For one thing the cost savings of reduced supply distribution cannot be achieved because the supply system is existing. Another issue for retrofitting containment is that the fire sprinkler system typically needs to be modified to provide proper coverage with containment or the containment system needs to be designed to fall away in the event of a fire. And of course any modifications to an existing data center are difficult because of the need to maintain continuous uptime. Existing data centers are excluded from the containment requirement. However, some recently collected first cost and savings data on some containment retrofit projects points out just how cost effective containment is. PG&E has been offering incentives to data centers to retrofit containment for a couple years. PG&E has shared both the implementation cost and energy savings from the retrofit sites. See Figure 25.

Figure 25. Simple Payback for Containment Retrofits



4 Stakeholder Input

To the extent possible, explain the key issues discussed and key concerns raised by stakeholders.

Minutes of the Stakeholder Meetings are included in Appendix 7.2 and 7.3. There were no serious concerns raised by stakeholders. One of the issues discussed was the ability of central VAV systems to turn down to low flow rates when serving computer closets at night. A common misperception is that variable speed driven motors require a minimum speed to prevent motor overheating. There is actually no evidence to support this claim. Motor and drive manufacturers are quick to point out that there is no minimum speed required for motor cooling. See, for example, the letter below from a leading variable speed drive manufacturer and the letter below from an expert on motors at the Department of Energy.

July 2, 2003

Mr. Dave Eisenberg
Air Treatment Corporation
2156 Central Avenue
Alameda, CA 94501

REF: ABB DRIVES – VARIABLE TORQUE SPEED RANGE

Dave,

ABB fully endorses a 10 to 1 speed range for most variable torque loads like centrifugal fans and pumps. In this type of application the HP and therefore the current drops by the square or cube of the speed. With the reduced current the motor stays cool and has no problem with the lower speeds.

We often find that going below 30% speed does not yield positive flow so in most cases going down to 10% is not practical.

If you have any further questions, please let me know.

Jim Hoyt
Territorial Manager
ABB Drives

Department of Energy
Washington, DC 20585

October 5, 2010

Mark Hydeman
Taylor Engineering, LLC
1080 Marina Village Parkway
Suite 501
Alameda, CA 94501

Subject: Minimum Operating Speeds for Adjustable Speed Drives

Dear Mr. Hydeman,

Thank you for contacting the EERE Information Center with your questions about turndown or operating speed ratios for motors with adjustable speed drive (ASD) speed control. I understand operators of buildings with VAV systems are concerned that operation at reduced speeds can adversely impact motor life.

I spoke with a drives specialist at Dykman Electric in Vancouver, WA and he indicated that to be safe, one should not exceed the manufacturer's advertised turndown ratio for the motor. Most manufacturers specify a turndown ratio for constant torque loads (conveyors and rotary screw compressors) with a second ratio specified for variable torque loads (centrifugal fans and pumps). Acceptable turndown ratios are often stamped on the motor nameplate and can vary greatly by motor manufacturer.

US Motors states that their motors offer up to a 10:1 speed range when coupled to a variable torque load (i.e. 60 Hz down to 6 Hz) with a 1.0 service factor. Motors must have Class F or better insulation, operate at a maximum ambient temperature of 40ºC, have a maximum altitude of 3,300 feet, and have a voltage supply not exceeding 460 volts. In contrast, GE EPAct Efficiency and Premium Efficiency motors may operate with a constant torque speed range of 2:1 to 4:1 but have an infinite speed range for variable torque loads (see attachments). Their X$D Ultra Extra Severe Duty motors offer a constant torque speed range of up to 20:1 and again offer an infinite variable torque speed range. Inverter-duty motors can do even better---both Baldor-Reliance and Marathon offer inverter and vector-duty motors that are capable of operating at a 1000:1 speed range at constant torque and can go all the way to stop under variable torque conditions (see enclosures).

The affinity or fan laws state that for centrifugal or variable torque loads, power varies at the cube of the motor speed ratio. As a 10:1 speed ratio reduces input power requirements to approximately 1/1000th of the original value, there is little to be gained from an energy savings standpoint from further speed reductions.

The Industrial Technologies Program in EERE helps companies begin improving energy efficiency, environmental performance, and productivity. One of our goals is that industry and industrial service providers gain easy access to near-term and long-term solutions for improving the performance of motor, steam, compressed air, and process heating systems.

If you have any colleagues who may be interested in our services, feel free to pass on our contact information. In the meantime, if you have other energy-related questions, please call the EERE Information Center at 1-877-337-3463 between 9 AM and 7 PM Eastern. Thank you.

Sincerely,

Gilbert A. McCoy, PE
Energy Systems Engineer
EERE Information Center
Energy Efficiency and Renewable Energy
U.S. Department of Energy
(877) 337-3463
www.eere.energy.gov

Enclosures (6)

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

5.1 SECTION 101 – DEFINITIONS AND RULES OF CONSTRUCTION

101 (b) **Definitions**.

COMPUTER ROOM is a room whose primary function is to house electronic equipment and that has a design equipment power density exceeding 20 watts/ft2 of conditioned floor area (215 watts/m2).

5.2 SECTION 144 – PRESCRIPTIVE REQUIREMENTS FOR SPACE CONDITIONING SYSTEMS

144 (e) Economizers.

1. Each individual cooling fan system that has a design supply capacity over 2,500 cfm and a total mechanical cooling capacity over 75,000 Btu/hr shall include either:

   A. An air economizer capable of modulating outside-air and return-air dampers to supply 100 percent of the design supply air quantity as outside-air; or

   B. A water economizer capable of providing 100 percent of the expected system cooling load as calculated in accordance with a method approved by the Commission, at outside air temperatures of 50°F dry-bulb/45°F wet-bulb and below.

   EXCEPTION 1 to Section 144(e)1: Where it can be shown to the satisfaction of the enforcing agency that special outside air filtration and treatment, for the reduction and treatment of unusual outdoor contaminants, makes compliance infeasible.

   EXCEPTION 2 to Section 144(e)1: Where the use of outdoor air for cooling will affect other systems, such as humidification, dehumidification, or supermarket refrigeration systems, so as to increase overall building TDV energy use.

   EXCEPTION 3 to Section 144(e)1: Systems serving high-rise residential living quarters and hotel/motel guest rooms.

   EXCEPTION 4 to Section 144(e)1: 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~~.

   EXCEPTION 5 to Section 144(e)1: Where electrically operated unitary air conditioners and heat pumps have cooling efficiencies that meet or exceed the efficiency requirements of **Error! Reference source not found.** and **Error! Reference source not found.**.

   EXCEPTION 6 to Section 144(e)1: Fan systems primarily serving computer room(s)

2. If an economizer is required by Subparagraph 1, it shall be:

   A. Designed and equipped with controls so that economizer operation does not increase the building heating energy use during normal operation; and

      EXCEPTION to Section 144(e)2A: Systems that provide 75 percent of the annual energy used for mechanical heating from site-recovered energy or a site-solar energy source.

B. Capable of providing partial cooling even when additional mechanical cooling is required to meet the remainder of the cooling load.

3. Air economizers shall have high limit shutoff controls complying with **Error! Reference source not found.**.

144 (m) Additional Requirements for Computer Rooms.

1. **Economizers .** Each individual cooling fan system primarily serving computer room(s) shall include either:

   A. An integrated air economizer capable of providing 100 percent of the expected system cooling load as calculated in accordance with a method approved by the Commission, at outside air temperatures of 55°F dry-bulb/50°F wet-bulb and below; or

   B. An integrated water economizer capable of providing 100 percent of the expected system cooling load as calculated in accordance with a method approved by the Commission, at outside air temperatures of 40°F dry-bulb/35°F wet-bulb and below.

   EXCEPTION 1 to Section 144(m)1: Individual computer rooms under 5 tons in a building that does not have any economizers.

   EXCEPTION 2 to Section 144(m)1: New cooling systems serving an existing computer room in an existing building up to a total of 50 tons of new cooling equipment per building.

   EXCEPTION 3 to Section 144(m)1: New cooling systems serving a new computer room in an existing building up to a total of 20 tons of new cooling equipment per building.

   EXCEPTION 4 to Section 144(m)1: A computer room may be served by a fan system without an economizer if it is also served by a fan system with an economizer that also serves non-computer room(s) provided that all of the following are met:

   a. the economizer system is sized to meet the design cooling load of the computer room(s) when the non-computer room(s) are at 50% of their design load.

   b. the economizer system has the ability to serve only the computer room(s), e.g. shut off flow to non-computer rooms when unoccupied.

   c. the non-economizer system does not operate when the cooling load of the non-computer room(s) served by the economizer system is less than 50% of design load.

2. **Reheat.** Each computer room zone shall have controls that prevent reheating, recooling, and simultaneous provisions of heating and cooling to the same zone, such as mixing or simultaneous supply of air that has been previously mechanically heated and air that has been previously cooled, either by cooling equipment or by economizer systems.

3. **Humidification.** Non-adiabatic humidification (e.g. steam, infrared) is prohibited. Only adiabatic humidification (e.g. direct evaporative, ultrasonic) is permitted.

4. **Power Consumption of Fans**. The total fan power at design conditions of each fan system shall not exceed 27 watts per kBtuh of net sensible cooling capacity.

5. **Fan Control**. Each unitary air conditioner with mechanical cooling capacity exceeding 60,000 Btu/hr and each chilled water fan system shall be designed to vary the airflow rate as a function of actual load and shall have controls and/or devices (such as two-speed or variable speed control) that will result in fan motor demand of no more than 50 percent of design wattage at 66 percent of design fan speed.

6. **Containment**. Computer rooms with air-cooled computers in racks and with a design load exceeding 175 kW/room shall include air barriers such that there is no significant air path for computer discharge air to recirculate back to computer inlets without passing through a cooling system.

   EXCEPTION 1 to Section 144(m)6: Expansions of existing computer rooms.

   EXCEPTION 2 to Section 144(m)6: Computer racks with a design load less than 1 kW/rack.

EXCEPTION 3 to Section 144(m)6: Equivalent energy performance based on computational fluid dynamics or other analysis.

5.3 Nonresidential ACM Manual

2.4.1.5 Process Loads

Process load is the internal energy of a building resulting from an activity or treatment not related to the space conditioning, lighting, service water heating, or ventilating of a building as it relates to human occupancy. Process load may include sensible and/or latent components. Process loads for data centers includes transformers, UPS, PDU, server fans and power supplies, etc.

Modeling Rules for Standard Design (All):	The standard design shall use the same process loads for each zone as the proposed design.

Table N2-18 – Occupancy Assumptions When Lighting Plans are Submitted for the Entire Building or When Lighting Compliance is not Performed

Occupancy Type	#people per 1000 ft$^{2(1)}$	Sensible Heat per person$^{(2)}$	Latent Heat per person$^{(2)}$	Recept acle Load W/ft$^{2(3)}$	Hot Water Btu/h per person	Lighting W/ft$^{2(4)}$	Ventilation CFM/ ft$^{2(5)}$
Auditoriums (Note 8)	143	245	105	1.0	60	1.5	1.07
Classroom Building	40	246	171	1.0	108	1.1	0.32
Commercial and Industrial Building	5	268	403	0.43	108	0.6	0.15
Convention Centers (Note 8)	136	245	112	0.96	57	1.2	1.02
Data Centers	5	268	403	Note 9	108	0.8	0.15
Financial Institutions	10	250	250	1.5	120	1.1	0.15
General Commercial and Industrial Work Buildings, High Bay	7	375	625	1.0	120	1.0	0.15
General Commercial and Industrial Work Buildings, Low Bay	7	375	625	1.0	120	1.0	0.15
Grocery Stores (Note 8)	29	252	225	0.91	113	1.5	0.22
Library	10	250	250	1.5	120	1.3	0.15
Medical Buildings and Clinics	10	250	213	1.18	110	1.1	0.15
Office Buildings	10	250	206	1.34	106	0.85	0.15
Religious Facilities (Note 8)	136	245	112	0.96	57	1.6	1.03
Restaurants (Note 8)	45	274	334	0.79	366	1.2	0.38
Schools (Note 8)	40	246	171	1.0	108	1.0	0.32
Theaters (Note 8)	130	268	403	0.54	60	1.3	0.98
All Others	10	250	200	1.0	120	0.6	0.15

(1) Most occupancy values are based on an assumed mix of sub-occupancies within the area. These values were based on one half the maximum occupant load for exiting purposes in the CBC. Full value for design conditions. Full year operational schedules reduce these values by up to 50% for compliance simulations and full year test simulations.

(2) From Table 1, p. 29.4, ASHRAE 2001Handbook of Fundamentals

(3) From Lawrence Berkeley Laboratory study. This value is fixed and includes all equipment that is plugged into receptacle outlets.

(4) From Table 146-E of the Standards for the applicable occupancy. The lighting power density of the standard building, for areas where no lighting plans or specifications are submitted for permit and the occupancy of the building is not known, is 1.2 watts per square foot.

(5) Developed from §121 and Table 121-A of the Standards

(6) Hotel uses values for Hotel Function Area from Table N2-19.
(7) For retail and wholesale stores, the complete building method may only be used when the sales area is 70% or greater of the building area.
(8) For these occupancies, when the proposed design is required to have demand control ventilation by §121(c) 3 the ventilation rate is the minimum that would occur at any time during occupied hours. Additional ventilation would be provided through demand controlled ventilation to maintain CO_2 levels according to §121
(9) Receptacle load shall be specified by the user.

Table N2-19 – Area Occupancy Assumptions When Lighting Plans are Submitted for Portions or for the Entire Building or When Lighting Compliance is not Performed

Sub-Occupancy Type [1]	People per 1000 ft^{2} [2]	Sensible heat per person[3]	Latent heat per person[3]	Receptacle Load W/ft^{2} [4]	Hot water Btu/hper person	Lighting W/ft^{2} [5]	Ventilation CFM/ ft^{2} [6]
Auditorium (Note 10)	143	245	105	1.0	60	1.5	1.07
Auto Repair	10	275	475	1.0	120	0.9i	1.50
Bar, Cocktail Lounge and Casino (Note 10)	67	275	275	1.0	120	1.1	0.50
Beauty Salon	10	250	200	2.0	120	1.7	0.40
Classrooms, Lecture, Training, Vocational Room	50	245	155	1.0	120	1.2	0.38
Civic Meeting Place (Note 10)	25	250	200	1.5	120	1.3	0.19
Commercial and Industrial Storage (conditioned or unconditioned)	3	275	475	0.2	120	0.6	0.15
Commercial and Industrial Storage (refrigerated)	1	275	475	0.2	0	0.7	0.15
Computer Room	3	275	475	Note 11	120	0.8	0.15
Convention, Conference, Multi-purpose and Meeting Centers (Note 10)	67	245	155	1.0	60	1.4	0.50
Corridors, Restrooms, Stairs, and Support Areas	10	250	250	0.2	0	0.6	0.15
Dining (Note 10)	67	275	275	0.5	385	1.1	0.50
Electrical, Mechanical Room	3	250	250	0.2	0	0.7	0.15
Exercise, Center, Gymnasium	20	255	875	0.5	120	1.0	0.15
Exhibit, Museum (Note 10)	67	250	250	1.5	60	2.0	0.50
Financial Transaction	10	250	250	1.5	120	1.2	0.15
Dry Cleaning (Coin Operated)	10	250	250	3.0	120	0.9	0.30
Dry Cleaning (Full Service Commercial)	10	250	250	3.0	120	0.9	0.45
General Commercial and Industrial Work, High Bay	10	275	475	1.0	120	1. 0	0.15
General Commercial and Industrial Work, Low Bay	10	275	475	1.0	120	0.9	0.15
General Commercial and Industrial Work, Precision	10	250	200	1.0	120	1. 2	0.15
Grocery Sales (Note 10)	33	250	200	1.0	120	1.6	0.25
High-Rise Residential Living Spaces [9]	5	245	155	0.5	(7)	0.5	0.15
Hotel Function Area (Note 10)	67	250	200	0.5	60	1.5	0.50
Hotel/Motel Guest Room [9]	5	245	155	0.5	2800	0.5	0.15
Housing, Public and Common Areas: Multi-family, Dormitory	10	250	250	0.5	120	1.0	0.15
Housing, Public and Common Areas:, Senior Housing	10	250	250	0.5	120	1.5	0.15
Kitchen, Food Preparation	5	275	475	1.5	385	1.6	0.15
Laboratory, Scientific	10	250	200	1.0	120	1.4	0.38
Laundry	10	250	250	3.0	385	0.9	0.15
Library, Reading Areas	20	250	200	1.5	120	1.2	0.15
Library, Stacks	10	250	200	1.5	120	1.5	0.15
Lobby, Hotel	10	250	250	0.5	120	1.1	0.15
Lobby, Main Entry	10	250	250	0.5	60	1.5	0.15
Locker/Dressing Room	20	255	475	0.5	385	0.8	0.15
Lounge, Recreation (Note 10)	67	275	275	1.0	60	1.1	0.50
Malls and Atria (Note 10)	33	250	250	0.5	120	1.2	0.25
Medical and Clinical Care	10	250	200	1.5	160	1.2	0.15
Office (Greater than 250 square feet in floor area)	10	250	200	1.5	120	0.9	0.15

Office (250 square feet in floor area or less)	10	250	200	1.5	120	1.1	0.15
Police Station and Fire Station	10	250	200	1.5	120	0.9	0.15
Religious Worship (Note 10)	143	245	105	0.5	60	1.5	1.07
Retail Merchandise Sales, Wholesale Showroom (Note 10)	33	250	200	1.0	120	1.6	0.25
Tenant Lease Space	10	250	200	1.5	120	1.0	0.15
Theater, Motion Picture) (Note 10)	143	245	105	0.5	60	0.9	1.07
Theater, Performance) (Note 10)	143	245	105	0.5	60	1.4	1.07
Transportation Function (Note 10)	33	250	250	0.5	120	1.2	0.25
Waiting Area	10	250	250	0.5	120	1.1	0.15
All Others	10	250	200	1.0	120	0.6	0.15

(1) Subcategories of these sub-occupancies are described in Section 2.4.1.1 (Occupancy Types) of this manual.

(2) Values based on one half the maximum occupant load for exiting purposes in the CBC. Full value for design conditions. Full year operational schedules reduce these values by up to 50% for compliance simulations and full year test simulations.

(3) From Table 1, p. 29.4, ASHRAE 2001 Handbook of Fundamentals.

(4) From Lawrence Berkeley Laboratory study. This value is fixed and includes all equipment that is plugged into receptacle outlets.

(5) From Table 146-F of the Standards for the applicable occupancy. Compliance software shall use this value for the standard building design when lighting compliance is performed for the zone or area in question.

(6) Developed from §121 and Table 121-A of the Standards.

(7) Refer to residential water heating method.

(8) The use of this occupancy category is an exceptional condition that shall appear on the exceptional conditions checklist and thus requires special justification and documentation and independent verification by the local enforcement agency.

(9) For hotel/motel guest rooms and high-rise residential living spaces all these values are fixed and are the same for both the proposed design and the standard design. Compliance software shall ignore user inputs that modify these assumptions for these two occupancies. Spaces in high-rise residential buildings other than living spaces shall use the values for Housing, Public and Common Areas (either multi-family or senior housing).

(10) For these occupancies, when the proposed design is required to have demand control ventilation by §121(c) 3 the ventilation rate is the minimum that would occur at any time during occupied hours. Additional ventilation would be provided through demand controlled ventilation to maintain CO_2 levels according to §121.

(11) Receptacle load shall be specified by the user.

Table N2-20 – Schedule Types of Occupancies & Sub-Occupancies

Occupancy or Sub-Occupancy Type	Schedule
Atrium	Table N2-8: Nonresidential
Auditorium	Table N2-8: Nonresidential
Auto Repair	Table N2-8: Nonresidential
Bar, Cocktail Lounge and Casino	Table N2-8: Nonresidential
Beauty Salon	Table N2-8: Nonresidential
Classrooms, Lecture, Training, Vocational Room	Table N2-8: Nonresidential
Civic Meeting Place	Table N2-8: Nonresidential
Commercial and Industrial Storage	Table N2-8: Nonresidential
Computer Room, Data Center	Table N2-13: Computer Room
Convention, Conference, Multipurpose, and Meeting Centers	Table N2-8: Nonresidential
Corridors, Restrooms, Stairs, and Support Areas	Table N2-8: Nonresidential
Dining	Table N2-8: Nonresidential
Electrical, Mechanical, Telephone Room	Table N2-8: Nonresidential
Exercise Center, Gymnasium	Table N2-8: Nonresidential
Exhibit, Museum	Table N2-8: Nonresidential
Financial Transaction	Table N2-8: Nonresidential
Dry Cleaning (Coin Operated)	Table N2-8: Nonresidential
Dry Cleaning (Full Service Commercial)	Table N2-8: Nonresidential
General Commercial and Industrial Work, High Bay	Table N2-8: Nonresidential
General Commercial and Industrial Work, Low Bay	Table N2-8: Nonresidential
General Commercial and Industrial Work, Precision	Table N2-8: Nonresidential
Grocery Sales	Table N2-8: Nonresidential
High-rise Residential with Setback Thermostat	Table N2-10: Residential / with Setback
High-rise Residential without Setback Thermostat	Table N2-11: Residential / without Setback
Hotel Function Area	Table N2-9: Hotel Function
Hotel/Motel Guest Room with Setback Thermostat	Table N2-10: Residential / with Setback
Hotel/Motel Guest Room without Setback Thermostat	Table N2-11: Residential / without Setback
Hotel/Motel Hallways	Table N2-9 Hotel Function
Housing, Public and Commons Areas, Multi-family with Setback Thermostat	Table N2-10: Residential / with Setback
Housing, Public and Commons Areas, Multi-family without Setback Thermostat	Table N2-11: Residential / without Setback
Housing, Public and Common Areas, Dormitory, Senior Housing with Setback Thermostat	Table N2-10: Residential / with Setback
Housing, Public and Commons Areas, Dormitory, Senior Housing without Setback Thermostat	Table N2-11: Residential / without Setback
Kitchen, Food Preparation	Table N2-8: Nonresidential
Laboratory, Scientific	Table N2-8: Nonresidential
Laundry	Table N2-8: Nonresidential
Library, Reading Areas	Table N2-8: Nonresidential
Library, Stacks	Table N2-8: Nonresidential
Lobby, Hotel	Table N2-9: Hotel Function
Lobby, Main Entry	Table N2-8: Nonresidential
Locker/Dressing Room	Table N2-8: Nonresidential
Lounge, Recreation	Table N2-8: Nonresidential
Mall	Table N2-12: Retail

Occupancy or Sub-Occupancy Type	Schedule
Medical and Clinical Care	Table N2-8: Nonresidential
Office	Table N2-8: Nonresidential
Police Station and Fire Station	Table N2-8: Nonresidential
Religious Worship	Table N2-8: Nonresidential
Retail Merchandise Sales, Wholesale Showroom	Table N2-12: Retail
Tenant Lease Space	Table N2-8: Nonresidential
Theater, Motion Picture	Table N2-8: Nonresidential
Theater, Performance	Table N2-8: Nonresidential
Transportation Function	Table N2-8: Nonresidential
Waiting Area	Table N2-8: Nonresidential
All Other	Table N2-8: Nonresidential

The following occurs multiple times:

Compliance software shall use the same default assumptions, listed in Table N2-5 through Table ~~N2-12~~ N2-13

NOTE: There is already a Table N2-13 through N2-27 so either those will have to be renumbered or this new table will have to be renumbered.

Table N2-13 – Computer Room Occupancy Schedules

-	-	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Heating (ºF)		60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
Cooling (ºF)		80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80
Lights (%) Uncontrolled	WD	5	5	5	5	10	20	40	70	80	85	85	85	85	85	85	85	85	80	35	10	10	10	10	10
	Sat	5	5	5	5	5	10	15	25	25	25	25	25	25	25	20	20	20	15	10	10	10	10	10	10
	Sun	5	5	5	5	5	10	10	15	15	15	15	15	15	15	15	15	15	10	10	10	5	5	5	5
Equipment (%)	Jan, May, Sept	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25
	Feb, Jun, Oct	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50
	Mar, Jul, Nov	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75
	Apr, Aug, Dec	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100

-	-	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Fans		on	on	on	on	on	on	on	on	on	on	on	on	on	on	on	on	on	on	on	on	on	on	on	on
Infiltration (%)		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
People (%)	WD	0	0	0	0	5	10	25	65	65	65	65	60	60	65	65	65	65	40	25	10	5	5	5	0
	Sat	0	0	0	0	0	0	5	15	15	15	15	15	15	15	15	15	15	5	5	5	0	0	0	0
	Sun	0	0	0	0	0	0	0	5	5	5	5	5	5	5	5	5	5	5	5	5	0	0	0	0
Hot Water (%)	WD	0	0	0	0	10	10	50	50	50	50	70	90	90	50	50	70	50	50	50	10	10	10	10	0

2.5.2.4 Standard Design Systems

Description: The reference method will assign one of five Standard Design System types for all proposed HVAC systems in order to establish an energy budget for the standard building. This system is generated and modeled for all buildings, even if no mechanical heating or cooling is included in the building permit.

Compliance software shall require the user to input the following for each system:

1. **Building Type** - low-rise nonresidential, high-rise nonresidential, residential and hotel/motel guest room
2. **System Type** - single zone, multiple zone
3. **Heating Source** - fossil fuel, electricity
4. **Cooling Source** - hydronic, other (for high-rise residential and hotel/motel guest room, only)

All Compliance software shall accept input for and be able to model the following system types for both the standard and proposed design:

- **System 1**: Packaged Single Zone (PSZ), Gas furnace and electric air conditioner.
- **System 2**: Packaged Single Zone (PHP), Electric heat pump and air conditioner.
- **System 3**: Packaged Variable Air Volume (PVAV), Central gas boiler with hydronic reheat and electric air conditioner.
- **System 4**: Built-up Variable Air Volume (VAV), Central gas boiler with hydronic reheat and central electric chiller with hydronic air conditioning.

- **System 5**: Built-Up Single Zone (BSZ), Central gas boiler and electric chiller serving individual units with hydronic heating and cooling coils.
- **System 6**: Computer Room Air Handlers (CRAHs), water-cooled central electric chiller
- **System 7**: Computer Room Air Conditioners (CRACs), air-cooled air conditioners

Modeling Rules for Standard Design (New):

The standard design system selection is shown in Table N2-21. The reference method chooses the standard HVAC system only from the ~~five~~ seven minimum systems listed above. The reference method will select its standard system according to Table N2-21, for the standard design system, regardless of the system type chosen for the proposed design. For example, a hydronic heating system served by a gas-fired boiler to supply hot water to the loop for a low-rise nonresidential building is considered a single zone (fan) system with fossil fuel for a heating source, and would be compared to System #1 - a Packaged Single Zone Gas/Electric System. Likewise a gas-fired absorption cooling system with a gas-fired furnace serving a single zone would be compared to System #1 also. **Error! Reference source not found.** through **Error! Reference source not found.** describes the five standard design system types. If more than 75% of the proposed building cooling capacity serves computer rooms then the entire building is modeled as System 6 or System 7. If less than 75% of the proposed building cooling capacity serves computer rooms but there are any computer rooms with a cooling capacity exceeding 120,000 Btuh then the computer rooms shall be modeled with a separate system than the non-computer rooms.

Table N2-21 – Standard Design HVAC System Selection

Building Type	System Type	Proposed Design Heating Source	System
Low-Rise Nonresidential (three or fewer stories above grade)	Single Zone	Fossil	System 1 – Packaged Single Zone, Gas/Electric
		Electric	System 2 – Packaged Single Zone, Heat Pump
	Multiple Zone	Any	System 3 – Packaged VAV, Gas Boiler with Reheat
High Rise Nonresidential (four or more stories)	Single Zone	Any	System 5 – Built-up Single Zone System with Central Plant
	Multiple Zone	Any	System 4 – Central VAV, Gas Boiler with Reheat
All Residential including Hotel/Motel Guest Room	Hydronic	Any	System 5 – Four Pipe Fan Coil System with Central Plant
	Other	Fossil	System 1 (No economizer) – Packaged Single Zone, Gas/Electric
		Electric	System 2 (No economizer) – Packaged Single Zone, Heat Pump
Total computer room design load is over 3,000,000 Btuh or the non-computers are System 4 or 5	Single Zone	Any	System 6 – CRAH units
Computer rooms that do not meet the System 6 conditions	Single Zone	Any	System 7 – CRAC units

Table N2-??22 – System #6 Description

System Description:	CRAH Units
Supply Fan Power:	0.49 W/cfm at design flow (see equipment sizing).
Supply Fan Control:	variable speed drive. Fan power ratio at part load = speed ratio ^3 (e.g. 12.5% of design power at 50% speed).
Return Fan Control:	No return fans
Minimum Supply Temp:	60
Equipment sizing	CFM and cooling capacity sized at 110% of the calculated load. One fan system per room.
Cooling System:	Chilled water
Chilled Water Pumping System	Same as System 4
Cooling Efficiency:	Same as System 4
Maximum Supply Temp:	80
Heating System:	none
Economizer:	Integrated differential dry bulb economizer
Supply Temp and Supply Fan Control:	Supply air temperature setpoint shall be linearly reset from minimum at 50% cooling load and above to maximum at 0% cooling load. Fan volume shall be linearly reset from 100% air flow at 100% cooling load to minimum air flow at 50% cooling load and below. Minimum fan volume setpoint shall be 50%. (this is effectively an “airflow first” sequence”)

Table N2-??23 – System #7 Description

System Description:	CRAC Units
Supply Fan Power:	0.49 W/cfm at design flow (see equipment sizing) where economizer is required, 0.39 W/cfm where economizer is not required.
Supply Fan Control:	Constant speed if total cooling capacity for the room < = 5 tons, otherwise: variable speed drive. Fan power ratio at part load = speed ratio ^3 (e.g. 12.5% of design power at 50% speed).
Return Fan Control:	No return fans
Minimum Supply Temp:	60
Cooling System:	Air-cooled DX
Equipment sizing	CFM and cooling capacity sized at 120% of the calculated room load. One fan system per room.
Cooling Efficiency:	Minimum packaged air conditioner efficiency based on calculated total cooling capacity for each room ● If cooling capacity > 20 tons then use 10 ton min efficiency ● If cooling capacity <20 tons then use capacity/2 min efficiency
Maximum Supply Temp:	80
Heating System:	None
Economizer:	No economizer if total cooling capacity for the room < 5 tons and building does not have any economizers, otherwise: Integrated differential dry bulb economizer
Supply Temp and Supply Fan Control:	VAV: Supply air temperature setpoint shall be linearly reset from minimum at 50% cooling load and above to maximum at 0% cooling load. Fan volume shall be linearly reset from 100% air flow at 100% cooling load to minimum air flow at 50% cooling load and below. Minimum fan volume setpoint shall be 50%. (this is effectively an “airflow first” sequence") CV: supply air temperature setpoint modulates to meet the load.

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7 Appendices

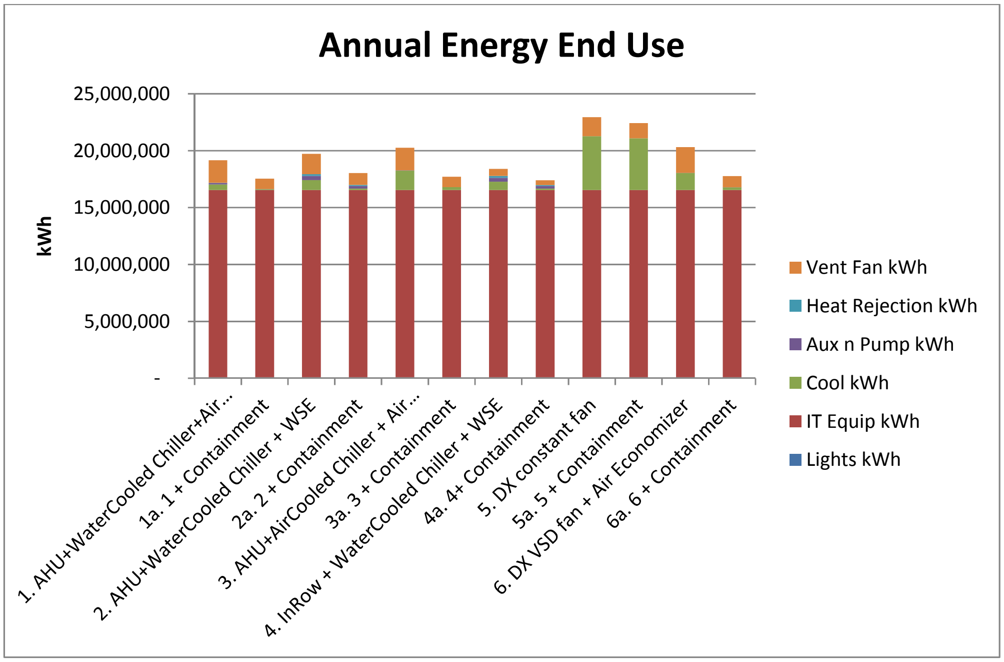
7.1 Typical Energy Benefits Calculation

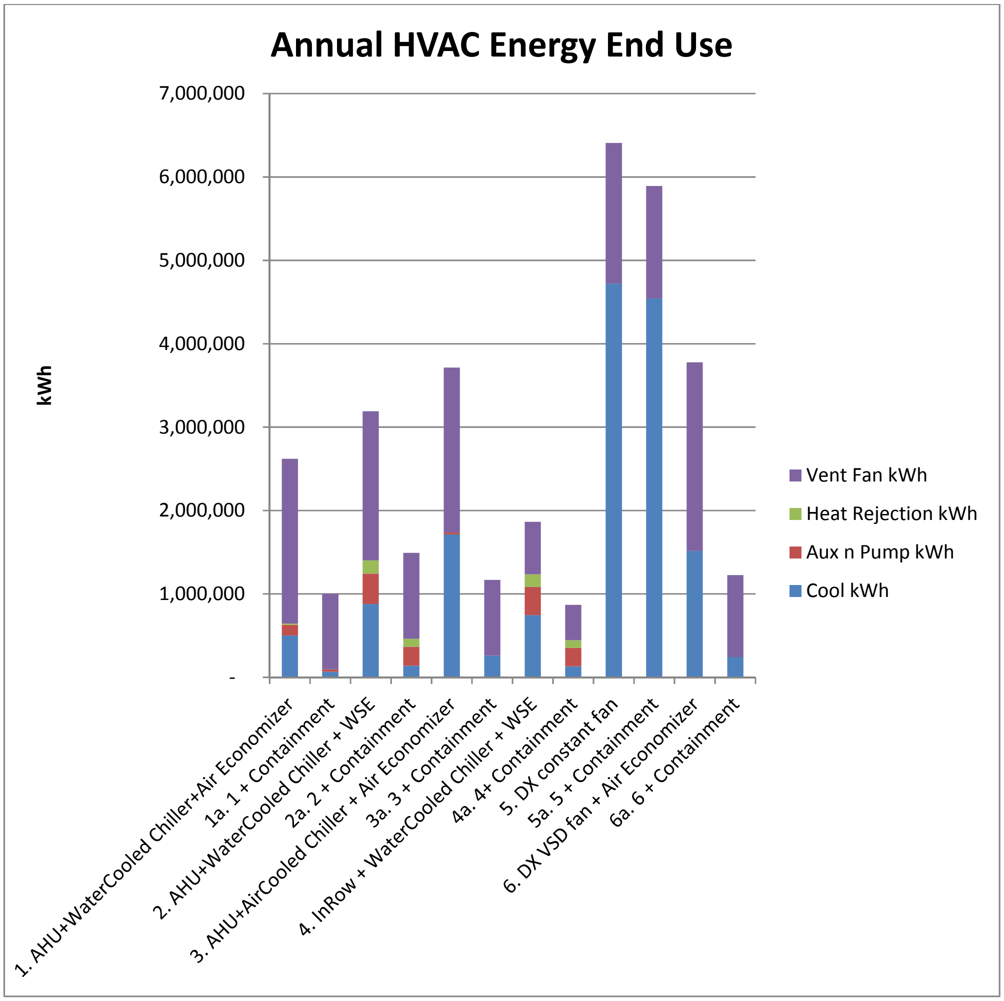
Basecase: Air-cooled DX CRAC units, no economizer (case 5 below)

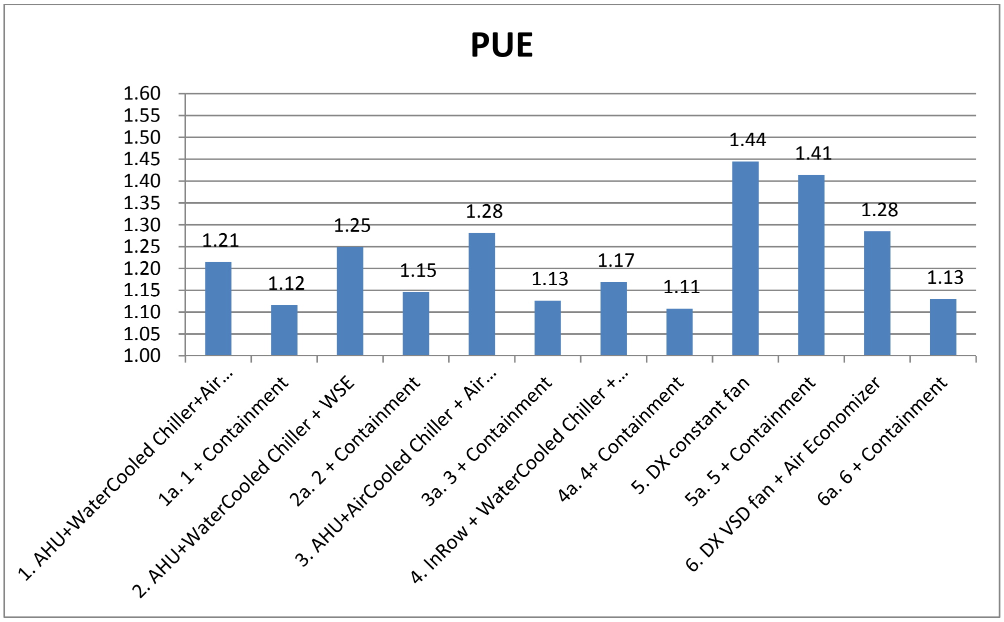
Proposed case: Chilled water air handlers with airside economizers and containment (case 1a)

Location: San Jose

IT density: 100 W/ft^2




Annual HVAC Energy End Use
7,000,000
6,000,000
5,000,000
4,000,000
3,000,000
2,000,000
1,000,000
-
kWh
Vent Fan kWh
Heat Rejection kWh
Aux n Pump kWh
Cool kWh
1. AHU+WaterCooled Chiller+Air Economizer
1a. 1 + Containment
2. AHU+WaterCooled Chiller + WSE
2a. 2 + Containment
3. AHU+AirCooled Chiller + Air Economizer
3a. 3 + Containment
4. InRow + WaterCooled Chiller + WSE
4a. 4+ Containment
5. DX constant fan
5a. 5 + Containment
6. DX VSD fan + Air Economizer
6a. 6 + Containment


PUE
1.60
1.55
1.50
1.45
1.40
1.35
1.30
1.25
1.20
1.15
1.10
1.05
1.00
1.21
1.12
1.25
1.15
1.28
1.13
1.17
1.11
1.44
1.41
1.28
1.13
1. AHU+WaterCooled Chiller+Air…
1a. 1 + Containment
2. AHU+WaterCooled Chiller + WSE
2a. 2 + Containment
3. AHU+AirCooled Chiller + Air…
3a. 3 + Containment
4. InRow + WaterCooled Chiller +…
4a. 4+ Containment
5. DX constant fan
5a. 5 + Containment
6. DX VSD fan + Air Economizer
6a. 6 + Containment

7.2 Stakeholder Meeting 1 Minutes

Minutes

CASE for Data Centers
Stakeholder Meeting 1

May 11, 2010

Held at NetApp, San Jose

Attendees:

Rick Brotherton, Core Support Systems
Jerry Burkhardt, Syska Hennesey Group
Michael Cohen, Advanced Data Centers
Chas Escher, Advanced Data Centers
Jon Haas, Intel
Phil Hughes, Clustered Systems Co
Ian Kucma, Syska Henessey Group
Denis Harker, Redwood City Electric
Dan Hyman, Custom Mechanical Systems
Greg Stover, NER Data Products, Inc.
Meisa Kassis, Cupertino Electric
Ted Marwitz, Data Aire
John Pappas, Mazzetti & Associates
Mark Parry, Custom Mechanical Systems
Steve Press, Kaiser Permananente
Ralph Renne, NetApp
Neil Risch, Core Support Systems
Bob Seese, Advanced Data Centers
Jon Shank, Pelio & Associates
John Sheputis, Fortune Data Centers
Victor Steffen, Syska Hennesey Group
Whitney Stone, Syska Hennesey Group
Jeff Stein, Taylor Engineering
Jeff Trower, Data Aire
Shlomo Novotny, Vette
Stan Levinstone, Vette
Tim Twomey, AT&T

Project and Utility Staff:

Mark Bramfitt, Bramfitt Consulting
Mark Hydeman, Taylor Engineers

Devin Rauss, Southern California Edison

Attendees by Conference Call:

(Due to failure of full webinar capability, these attendees were asked to attend a subsequent webinar to review the slide materials)

Bob Pereira, HP
Joe Loyer, California Energy Commission
Mukesh Khattar, Oracle
Ken Baker, HP
Eric Abeyta, State of CA DGS
John Tucillo, Green Grid
Mike Hogan, IBM
Jack Pouchet, Liebert Corp.
Ken Traber, Rumsey Engineers
Dale Sartor, LBNL
Bill Tschudi, LBNL

Overview

Devin Rauss of Southern California Edison delivered a description of the CEC's regulatory authority to implement building and appliance energy efficiency codes, and how investor-owned utilities are charged with supporting code updates and extensions as part of their energy efficiency program efforts regulated by the CPUC.

The regulatory criteria for establishing Title 24 new construction codes was explained: standards must meet a lifetime cost-effectiveness standard based on an avoided utility capacity cost methodology provided by the CEC.

Technical Discussion

Mark Hydeman and Jeff Stein of Taylor Engineering described the measures and technologies under consideration for the proposed standard, as well as the types of data center construction that would be covered.

Jon Hass of Intel asked whether the standards would be different depending on the Uptime Institute Tier rating (related to reliability and redundancy) of the design.

Steve Press of Kaiser pointed out that EPA gave up on the TIER distinctions for their programs

John Pappas of Mezetti & Associates asked about how conversion of office space, particularly in a multi-tenant facility, would be handled. The concern is that some very small server rooms or closets may not be a good fit for the new standards, because there are physical limitations to extending air conditioning capacity to these rooms. (For example, an outside air system may not be feasible for an interior room with no exterior wall or roof exposure.)

Jeff Stein of Taylor Engineering responded that we have a measure to address this (see slides).

There was some general discussion regarding economizers, with some participants opining that certain types of economizer systems may not provide enough energy savings to warrant inclusion in the code.

John Pappas of Mazzetti asked asked if blanking plates would be included in airflow containment requirements, if they were included in the proposed code.

Jeff Stein said that this would be included in the User's Manual.

John Pappas of Mazzetti asked if the thresholds for economizers should be per tenant or per building.

[Mark Hydeman, these are generally per permit application]

John Pappas of Mazzetti and Ralph Renne of NetApp had some concerns about running a large central fan for small IDF rooms. Jeff Stein of Taylor Engineering said that Taylor Engineering does this all the time and has had no problems in practice.

John Pappas of Mazzetti asked if the proposal would allow an "econo coil" to comply.

Jeff Stein and Mark Hydeman of Taylor Engineering expressed that these are not nearly as effective as air-side economizers or integrated water-side economizers on a chilled water plant.

Greg Stover of NER Data Products, Inc. suggested that we require control by pressure not temperature on systems with containment. In his experience temperature is too slow in high density racks.

Jeff Stein and Mark Hydeman of Taylor Engineering stated this was in issue for the User's Manual and that the requirement is for containment not fan control.

Steve Press of Kaiser Permanente mentioned that they are looking into using heat pumps for heating the generator blocks in their facility in Napa.

There was lots of discussion on LCCA criteria and the baselines but general agreement about the scope of the proposals.

Next Meeting:

The second stakeholder meeting will be set in the third quarter, where a review of the technical work justifying code measures will be presented and discussed.

Minutes prepared by Mark Bramfitt.
Direct comments, additions, or corrections to mark@markbramfitt.com.

7.3 Stakeholder Meeting 2 Minutes

Minutes

CASE for Data Centers
Stakeholder Meeting 2

September 16, 2010

Held at Pacific Gas and Electric Company's
Pacific Energy Center, San Francisco

Physical Attendees

Todd Masters, Digital Realty Trust
Mark Parry, CMS
Magnus Herrlin, ANCIS
Victor Steffen, Syska Hennessey Group
Greg Stover, NER Data
Ron Tessing NER Data
Tin Tse, Equinix
Ryan Matley, Pacific Gas and Electric Company
Kip Hensley, Syska Hennessey Group
Jamy Bacchus, National Resources Defense Council

Webinar Attendees

Ian Kucma Syska Hennessey Group
Dan Hyman CMS
Jennifer Pomi CMI
Joe Loyer CEC
Mike Moreno SMUD
Robert Clevenger SMUD
Jon McHugh McHugh Energy
Randall Higa SCE
Jose Herrera Syska Hennessey Group
Ted Marwitz Data Air

Project and Utility Staff:

Devin Rauss, Southern California Edison
Mark Hydeman, Taylor Engineering
Jeff Stein, Taylor Engineering
Randall Higa, Southern California Edison (via webinar)

Mark Bramfitt, Consultant

Overview/General

The Title 24 2011 proposed requirements are based on those already adopted for ASHRAE 90.1-2010 (Addendum AQ and BU).

In both Title 24 and 90.1 the scope of the standard needed changing to make it clear which processes are included. The proposed requirements also considered requirements currently in the Oregon and Washington State energy codes.

A common misperception (fueled in part by the Google Policy Blog, http://tiny.cc/lejwb) is that 90.1 mandated air-side economizers for data centers. This is not true for either the requirements adopted for 90.1-2010 nor the proposed Title 24 requirements. Both standards require either air- or water-side economizers prescriptively. You can comply using an air economizer, a water-side economizer or no economizer and equivalent energy (TDV) performance using the Performance method. In addition we studied the cost effectiveness of meeting the economizer requirement with a closed heat exchanger (like Z-duct) in the event that you are concerned about gaseous contaminants.

We talked about the CASE initiatives and how the process for Title 24 2011 differs from previous rounds of the standards. In 2008 and previous rounds of the Title 24 standard most measures were introduced during the official CEC hearings. In this round the Investor Owned Utility Companies (IOUs) are holding targeted stakeholder workshops like this one to gather input and vet proposed requirements. Stakeholders are encouraged to comment on proposed standards through this CASE study development process. The comment period during the official California Energy Commission rule making hearings will be limited.

What we commonly refer to as "Title 24," is really Title 24, Part 6 (the energy code). Title 24, Part 6 provides minimum requirements for new buildings and new construction in existing buildings. Some of the measures, as noted on the slides, may not be evaluated for inclusion in the Title 24, Part 6 code, and instead may be included in the "REACH" code (aka the California Green Building Code, Title 24, Part 11). The REACH code can be adopted by local jurisdictions at their discretion, and is also intended to indicate where future code development work should be undertaken. The reach code, like USGBC's LEED ratings and ASHRAE/USGBC's Standard 189 include non-energy resource issues like recycling, transportation and water usage. However for the first time Title 24, Part 6 2011 will include water usage in its scope.

The analysis presented in this workshop used California Climate data newly developed for the 2011 Standard but the TDV (Time Dependant Valuation) files from Title 24 2008. These TDV numbers should be conservative as the energy and demand costs have gone up. The TDV files in 2011 and previous standards vary by climate zone. They include: energy costs, demand costs and to some extent energy infrastructure costs (e.g. transmission and distribution equipment in constrained areas). The 2011 TDV values were not available at the time of this workshop. Information on the TDV evaluation can be found on the CEC's website: http://www.energy.ca.gov/title24/.

The timeline for development of the proposed Title 24, Part 6 code for 2011 is due to be complete in 2010. The CEC will hold their hearings in 2011, it will go into effect in January 2013. Between adopting the code and it's taking effect, the CEC will also need to adopt a user's manual, compliance forms and the ACM (alternative calculation method) manual. They will also do trainings for the AHJs and time is needed for the compliance program vendors to program their tools using the new ACM rules (EnergyPro and eQuest are the programs used for the non-residential standards).

Measure Review

Captured below is the discussion for each of the sections of the slide presentation. Refer also to the posted slides to get a full picture.

Scope (Slide 15-20)

The proposal is to create new definitions for Exempt Processes and Covered Processes. Covered Processes are listed on Slide 18. Examples for use of these definitions are covered in the measures that follow.

We also propose to add a definition of a Computer Room (from ASHRAE 90.1). This is on slide 19.

Economizers (Slides 21-45)

The proposed prescriptive requirement for economizers in Title 24 2011 is broader than that in 90.1 2010 but less broad than that in Oregon or Washington State.

Slide 27 shows the proposed base requirement for either and air- or water-side economizer based on project size. The scope of a "project" in Title 24 is the scope of the permit. In general you do not have to retrofit existing systems to meet the requirements for subsequent work. However you might include a new water-side economizer in an existing chilled water plant to meet the requirements for a new data center served by that plant.

Slides 28 and 29 present a special case of a small computer room (or IDF closet) that is located in a larger building. The intention of this requirement is to accommodate a small computer room where it might need supplemental cooling (e.g. from a chilled water fan coil) when the rest of the building is at peak load but can be satisfied by the main building system during evenings and weekends.

Slides 32 to 45 summarize our analysis done to date.

Slide 34 shows how we modulated the computer load to show the part-load performance of the computer room systems. In real projects data centers are built up over time (in many cases over several years). There is also the effect of running standby equipment in parallel with primary equipment (an efficiency strategy) and the impact of load shedding measures like virtualization and disk management. We are not mandating virtualization or disk management. This load profile is part of an addendum for data centers that will be part of ASHRAE 90.1 2014.

Slides 32 to 39 show that air-side economizers are cost effective down to 5 ton units serving a stand-alone data centers. The parametric runs included:

- Integrated, partially integrated and non-integrated economizers.
- Use of an air-to-air heat exchanger to prevent gaseous contaminants.

Jamie Baccus NRDC: Are there exemptions for climate zones? Not from our preliminary analysis, these measures appear to be cost effective in all 16 California climate zones.

Jamie Baccus NRDC (regarding the analysis presented in slides 40 to 42): Is the central system capacity determined at the building level or tenant level? The intention is the building level or tenant's current system level (e.g. if they own the AHU for that floor). The systems serving the space in question.

Mark Perry: For replacement of existing CRAC/CRAH units what is covered? The unit efficiencies (fan power and dx efficiency), and the economizer (or equivalent performance). Slide 27 covers the proposed exceptions to the economizer requirements.

Tin Tse: Is this code retroactive? No,it only applies to new equipment and systems.

Magnus Herrlin: Why not use Washington Energy Code? Because we are not sure that it will be cost effective.

Jamie Baccus NRDC: Can we aggregate projects to prevent owners from submitting multiple permits? We're open to input or recommendations if you have specific wording.

Unknown: Why was there a difference in the controls costs? Because the controls contractor was lock specified in the Pleasanton job. The LA job was competitively bid.

Magnus Herrlin: You should emphasize the fact that the analysis is worst case and conservative.

Humidification (Slides 46-53)

No steam or infrared humidifiers would be allowed under the proposed code (only adiabatic humidifiers like ultrasonic or direct evaporative cooling). Studies and general practice have shown that humidification is generally not required. This is the subject of a current ASHRAE research project and a recent ASHRAE Journal Article. NEBS a standard used for telecom central office facilities has no lower humidity limit. They use the same servers, storage devices and switch gear that are in other data centers.

Slide 52 is an example of how we will apply the new exempt process definition.

There was no discussion in this section.

Equipment Efficiency (Slides 54-58)

The existing Title 20 standards already cover the efficiency of CRAHs and CRACs. Although Standard 127 has undergone a major renovation and new SCOP ratings were adopted by ASHARE we could not find any cost or rated performance data from the major CRAC/H unit manufacturers. No changes are proposed for the existing Title 20 requirements.

Jon McHugh: If we use EERs from old metric and not SCOPs are we backsliding? There are two issues to consider

1) We can't get cost data from the manufacturers so we can't do LCCA;

2) You can't compare the two tables as the rating methods and conditions are different.

Joe Loyer, CEC: We can't put it in Part 6 without LCCA.

Tin Tse: Are there different SCOPs in the 90.1 table for upflow and downflow units? Yes

Fan Power Limitation (Slides 59-62)

The proposed requirement is to limit fan power to 27 Watts per kBtuh of cooling capacity. This limit will not preclude very many CRAC or CRAH models, but will prevent designers from specifying systems that try to move too much air through units.

Slide 62 is another example of the use of the exempt process load definition.

Jamy Bacchus: What was assumed for filters? MERV 7 or 8 as these are all recirculating units without economizers.

Fan Speed Control (Slides 63 to 69)

Existing fan speed control requirements that were adopted in Title 24 2008 and ASHRAE 90.1-2010 will go into effect on 1/1/2013. Due to the scope changes (Addendum AQ) in 90.1-2010 and the proposed categorization of data centers as a covered process in Title 24-2011, CRAC/H units 10 tons and larger will have to be VAV. This requirement can be met with 2-speed motors but most manufacturers of CRAC/H units are already using ECM motors or variable speed drives.

The proposed Title 24 2011 standards will require all direct expansion units over five tons and all chilled water units to have variable air volume capability. CRAC manufacturers offer equipment that meets the proposed standards today and appear to be supportive of the proposed standard.

Containment (Slides 70 to 87)

Data centers with 175 kW of load or more (about 200 tons of cooling) will be required to have suitable containment or air barrier systems to prevent discharge air from IT equipment from re-circulating.

Fan systems will be required to have backflow dampers.

There was discussion about modifying the proposed code language, and whether using the Rack Temperature Index could be used as a measure of compliance.

PG&E's RCX program showed that containment was very cost effective in retrofit. In new construction it can actually reduce cost (for instance by eliminating the need for a raised floor and by reducing the connected cooling load infrastructure).

Magnus: would it be better to use a measurable performance criteria like RTI? We are open to suggestion however this is a design requirement, enforcement might be a problem. Perhaps we'll add an acceptance test using return temperature index (RTI).

Lighting and Electrical (Slides 88 to 91)

The 90.1 committee is currently developing LPDs for data centers. Our consultant showed that you could meet the IESNA illumination guidelines between racks at ~0.82 w/sf.

Occupancy sensors turning off all but the emergency lighting (~60w/row) appear to be cost effective. These were based on the more expensive dual detection (IR and ultrasonic) sensors.

PDUs with NEMA premium efficiency transformers are currently only offered by two manufacturers. As a result the costs are high. We plan to include this in the Title 24, Part 11 REACH code.

Due to complexity with the topologies we don't feel that we can complete work on UPS systems and building transformers in this code cycle.

We are very interested in getting case studies with cost and energy savings that people have on these measures.

Performance Method Baseline (Slides 92 to 99)

As part of code development, a baseline must be established for data centers and added to the ACM manual. Our proposal builds on an Addendum developed for Appendix G of ASHRAE 90.1.

The proposal is to add two new system types to the manual:

- System 6: chilled water-based CRAH designs of over 250 tons
- System 7: direct expansion CRAH systems for small computer rooms

PUE Measurement (Slides 100 to 101)

A consortium including EnergyStar, ASHRAE and Green Grid recently published a standard for measuring PUE, "Recommendations for Measuring and Reporting Overall Data Center Efficiency," July 15th 2010. This standard has 4 potential levels of measurement. We discussed whether it would be prudent to require in either the base code (Part 6) or reach code (Part 11) a minimum level of measurement.

Most of the participants thought that this might be a good idea. We also appeared to have consensus that Level 0 (UPS output for peak) or Level 3 (server input) would be inappropriate at this time.

Magnus Herlin this is not a good metric for comparing different data centers. We wouldn't propose it for cross facility comparisons, we are proposing it for comparing a data center to itself over time.

Mark Bramfitt PUE is not useful for utility incentives, it is too easy to minupulate. What we need as an industry is something to measure utilization of equipment.

Mark Bramfitt if you add the measurement than some municipality will set a threshold on it.

Tin how about requiring measuring to Category 1 and calculating to Category 2?

Unknown: How do we deal with shared HVAC and shared lighting?

Jamy Bacchus NRDC: If you do this do you need an exception for small data centers?

Discussion

It may be useful to cite all capacity levels in both IT equipment loads and cooling loads.

The language requiring containment needs to be amended, and the measure may be difficult to enforce given how many data center projects are developed. (The building shell and cooling and power delivery systems may be installed, but IT racks and containment systems may not, at the point that a developer wishes to secure sign-off on the project.)

Next Meeting:

The third and final stakeholder meeting will be set late in the fourth quarter of 2010. The final proposed code will be presented and discussed at that meeting. The date and location of that meeting has not been set yet. You can get updates from the project website: http://www.taylor-engineering.com/T24%202011%20Data%20Centers/. Also any stakeholder who is listed as interested in data centers will receive an email announcement for this next meeting. To get listed as a stakeholder, click on this URL, https://app.e2ma.net/app/view:Join/signupId:1413047/acctId:1405070. Feel free to invite other interested parties to attend these meetings.

Minutes prepared by Mark Bramfitt.
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