CODES AND STANDARDS ENHANCEMENT INITIATIVE (CASE)

Fan Control and Integrated Economizers

2013 California Building Energy Efficiency Standards

California Utilities Statewide Codes and Standards Team   September 2011

This report was prepared by the California Statewide Utility Codes and Standards Program and funded by the California utility customers under the auspices of the California Public Utilities Commission.

Proposer: Taylor Engineering

Table of Contents

Table of Figures

1. Overview

1.1 Measure Title

Fan Control and Integrated Economizers

Originally this measure was titled “Single Zone VAV”. However, the name was changed for several reasons:

- VAV implies continuously variable volume but two speed fans meet the requirement
- The Title 24 definition of VAV implies multiple zone systems
- The measure was expanded to cover multiple zone systems and single zone systems
- The measure was expanded to more explicitly address economizer integration since integration is critical to achieving the expected energy savings

1.2 Description

This measure extends the current Single Zone VAV requirement (144(l)) from 10 tons down to 6 tons for DX equipment (starting 1/1/2015) and down to ¼ HP for chilled water equipment. It also clarifies the definition of an integrated economizer: Systems that require an economizer must be able to modulate cooling capacity (e.g. compressor output) down to 20% or less of total capacity.

1.3 Type of Change

This is a prescriptive measure. It expands and clarifies existing prescriptive requirements.

1.4 Energy Benefits

The following are simulation results for a prototype building in a typical California climate zone. Details of the analysis for this and other climate zones are presented in Section3.5.

	Electricity Savings (kWh/yr)	Natural Gas Savings (Therms/yr)
Per Prototype Building*	14,081	-8.0
Savings per square foot	2.61	-0.0015

* Prototype building: 5400 ft^2, single story, 5 zone office building, packaged single zone DX systems.

The Air-Conditioning, Heating, and Refrigeration Institute (AHRI) conducted a survey of packaged rooftop units sold in California in 2010. According to this data, there are approximately 40,000 tons of packaged rooftop units between 65,000 Btu/h and 110,000 Btu/h in California (see Section 3.5.1.2 and Appendix 7.9). Based on the average annual electricity savings of 965 kWh/ton for the 6 climate zones that were modeled, the annual statewide electricity savings would be 38 million kWh in the first year. This is a very conservative estimate of statewide savings since this is only for packaged rooftop units. It does not include any savings from CHW systems. It also does not include energy savings from improved compressor part load performance or savings from integrated economizers on systems over 110,000 Btu/h.

As noted in Section 3, the energy analysis is highly conservative. It uses conservative assumptions and does not include entire categories of energy and cost savings, such a compressor efficiency savings, maintenance cost savings and electrical system downsizing.

1.5 Environmental Impact

There are no significant potential adverse environmental impacts of this measure. This measure requires two speed or variable speed fans and more sophisticated controls that matches compressor output to zone load. Thus the negative environmental impacts are very small:

- when two speed motors are used a little extra copper in two speed motors for a fan motors serving a 7.5 ton RTU.
- when electronically commutated motors are used, control circuiting is slightly different for a very small change in number of components used in the electronic commutator
- additional control logic may have little to none impact on the materials used in controls

However this small negative environmental impact is dwarfed by the positive environmental impact.

1.6 Non-Energy Benefits

Non-energy benefits include:

- Improved indoor air quality – Systems that only have one or two steps of cooling capacity are not truly integrated economizers. To prevent freezing the evaporator coil they often shut off the economizer a significant fraction of the time when it could provide free cooling. A truly integrated economizer is able to keep the economizer enabled in economizer mode and provide more fresh air. Furthermore, systems without proper integration end up excessively cycling the economizer dampers. This often causes the dampers to fail prematurely. Typically they fail in the closed position or are put into the closed position and thus always provide minimum or no outside air.
- Improved comfort – Improved capacity turndown results in more stable space temperature and in a more uniform supply air temperature. Fluctuating supply air temperature can be a comfort issue due to the "dumping" sensation.
- Improved acoustics – Reduced fan speed reduces noise and improved capacity turndown reduces the noise of compressor cycling.
- Increased equipment life – Reduced fan speed reduces wear on fan motors and bearings. Reduced economizer and compressor cycling also increases equipment life.
- Better control of relative humidity – Both the fan control and the integrated economizer measure result in improved humidity control
- Electrical system stability - Reduction in the large in-rush currents associated with frequent starting of large compressor drive motors.

1.7 Technology Measures

1.7.1 Measure Availability:

There are currently commercially available technologies from multiple manufacturers that can meet the various applications of this measure, including:

- Two speed or variable speed fans on 6 ton or smaller DX units
    a. Aaon
    b. Carrier
    c. Daikin
    d. Mitsubishi
    e. JCI/York
    f. Mammoth
    g. Trane
- Multiple stage or variable capacity compressors on 6 ton or smaller DX units
    a. Aaon – digital scroll
    b. Carrier – digital scroll

c. Daikin – variable speed compressor
d. Mitsubishi – variable speed compressor
e. JCI/York – variable speed compressor expected by 2012

Variable speed models of reciprocating, scroll and rotary compressors are all commercially available. They have been in widespread use for many years in Asia and are now increasingly used in the U.S.

This measure has also been reviewed by the members of the AHRI Unitary Large Equipment (ULE) Committee. The members of that committee were polled to provide incremental cost data. They also approved by majority vote sharing the average cost data with the authors of this measure. AHRI will not release cost data of this nature unless at least three (3) manufacturers provide data. Thus AHRI has indicated that the measure is or will be commercially available from at least three (3) manufacturers.

A number of manufacturers, including Greenheck, JCI, and McQuay now offer EC fan motors or variable speed drives as standard or optional on their chilled water fan coils. Furthermore, variable speed drives can be easily field mounted on constant speed fan coils. Variable speed drives are available from at least a dozen manufacturers.

1.7.2 Useful Life, Persistence, and Maintenance:

Energy savings from this measure will persist for the life of the system. For DX systems, compliance with this measure will most likely mean specifying DX equipment that is designed for 2 speed or variable speed fan control and designed for cooling capacity turndown. Such systems are designed, tested, rated, certified and mass produced. There are no significant issues with persistence.

Similarly, chilled water units that include factory mounted EC motors or variable speed drives typically include the hardware and controls to properly vary fan speed out of the box. For CHW units where the VFD is added in the field some programming and commissioning is required but it can be quite simple (e.g. fan speed controlled by same signal controlling the CHW valve) and once the system is commissioned the savings should not erode over time.

For DX systems, incremental maintenance cost is included as a conservative placeholder. For the integrated economizer measures, incremental maintenance may well be negative due to reduced wear and tear on compressors and dampers due to reduced cycling.

Incremental maintenance cost data for CHW systems was provided by a Bay Area service contractor. It is estimated to be half an hour per year at a labor rate of $100/hr. Incremental maintenance will also come down in the future once the measure is adopted.

1.8 Performance Verification of the Proposed Measure

There is already an acceptance test for Supply Fan Variable Flow Controls. This test is intended for multiple zone VAV systems but with some minor proposed clarifications it can be used for both single and multiple zone variable flow systems.

Regarding the integrated economizer measure, the existing economizer acceptance test includes this: "verify that the economizer remains 100 percent open when the cooling demand can no longer be met by the economizer alone". A more detailed functional test to verify the amount of turndown before false loading would not be practical because it would require creating not only an actual cooling load but also manipulating the outside air temperature or at least the coil entering temperature which is not practical. However, the economizer construction inspection requirement should be expanded to include verifying the stages of compression and/or compressor capacity control.

1.9 Cost Effectiveness

Life cycle cost (LCC) per unit and per prototype building were calculated using the California 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$)		PV of[4] Energy Cost Savings – Per Proto Building - 15 yr measure life (PV$)	Change in 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
Fan Control – Single Zone DX unit	$496 (for 6 ton unit)	$901	$496	$901	$1,190	$2,162	$14,281	-$11,218	-$11,218
Fan Control – CHW fan coil	$282 (1/4 HP fan)	$2,820	$282	$2,820	$595	$5,950	$19,128	-$10,358	-$10,358
Fan Control – CHW AHU	$770 (per fan motor)	$7,700	$539	$5,390	$595	$5,950	$18,984	-$5,334	-$7,644
Integrated Economizer – Multiple Zone DX	$941 (10 tons)	$1,654	$941	$1,654	$595	$1,045	$3,891	-$1,192	-$1,192
Fan Control + Integrated Economizer – Single Zone DX	$2,133 (6 tons)	$4,239	$2,133	$4,239	$1,190	$2,365	$23,948	-$17,344	-$17,344

Figure 1. Lifecycle Cost Effectiveness for CZ06

1.10 Analysis Tools

Some modifications to the performance compliance software programs are likely to be required in order to easily quantify energy savings and peak electricity demand reductions resulting from the proposed measure. eQuest can be used to model 2 speed fan motors but it requires post-processing of high and low speed runs (see Section 3). eQuest can also model variable speed single zone systems using the PVVT system type (see Section 3). eQuest does not have the ability explicitly to model stepped capacity control and partially integrated economizer as compared to fully integrated economizers. It does have the ability to reasonably approximate some types of partially integrated economizers using either high limit controls or the ECONO-LOCKOUT keyword (see Section 3).

1.11 Relationship to Other Measures

Title 24 already includes requirements for single zone VAV systems. Those requirements are effectively extended to cover smaller size equipment by this requirement. Title 24 also already essentially has a requirement for variable speed drives on multiple zone VAV systems over 10 HP. Again, this measure effectively extends that requirement to cover smaller size equipment.

There is another CASE proposal entitled "Light Commercial Unitary HVAC" that includes a proposal to lower the economizer threshold from 7.5 tons to 4 tons. This proposal has been coordinated with that one and is not in conflict.

2. Methodology

This measure affects several types of equipment, including DX and chilled water, single zone and multiple zone and systems with and without minimum outside air controls. 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 system. Each individual measure and the associated analysis are described in more detail in the next section. In addition to analyzing the cost effectiveness of just the fan control measure and just the integrated economizer measure, we have also evaluated the cost effectiveness of both measures combined for DX systems. These analyses have shown that the fan control measure and the integrated economizer measure are cost effective individually and in combination. Furthermore, these analyses do not account for the full energy savings of these measures and are therefore conservative.

2.1 Stakeholder Meeting Process

All of the main approaches, assumptions and methods of analysis used in this proposal have been presented for review at one of three public Nonresidential HVAC Stakeholder Meetings. At each meeting, the utilities' CASE team invited feedback on the proposed language and analysis thus far, and sent out a summary of what was discussed at the meeting, along with a summary of outstanding questions and issues.

A record of the Stakeholder Meeting presentations, summaries and other supporting documents can be found at www.calcodes.com. Stakeholder meetings were held on the following dates and locations:

- First Nonresidential HVAC Stakeholder Meeting: April 27, 2010, California Lighting Technology Center, Davis, CA.
- Second Nonresidential HVAC Stakeholder Meeting: December 7, 2010, San Ramon Valley Conference Center, San Ramon, CA
- Third Nonresidential HVAC Stakeholder Meeting: March 2011, via webinar.

In addition to the Stakeholder Meetings, informal outreach and working sessions were conducted to allow detailed review of specific technical issues. See Section 0 Figure 35. Combined Measures: Fan Control and Integrated Economizer – Single Zone DX: Lifecycle Cost Results for 6 Ton Unit

Stakeholder Input.

2.2 Statewide Savings Estimation

The statewide energy savings associated with the proposed measures will be calculated by multiplying the energy savings per square foot with the statewide estimate of new construction in

2014. Details on the method and data source of the nonresidential construction forecast are in **Error! Reference source not found.**.

3. Analysis and Results

3.1 Fan Control – Single Zone DX

The first measure and system analyzed is fan control for single zone DX systems. In this measure, units that are 6 tons or larger require fans with two speeds and compressors with two stages. It was evaluated by comparing a basecase unit with a single speed fan and single stage compressor to a proposed case unit with a two speed fan and a two stage compressor. The energy analysis is very conservative for the following reasons:

- The proposed case uses a relatively high low fan speed (66%) and a relatively high low speed fan power (30%). Most if not all systems that will be used to meet this requirement when it goes into effect will have variable speed fans that will have a lower minimum speed and lower fan power at minimum speed. The higher fan speed and power were used in the analysis as a worst case scenario at the request of one of the stakeholders.
- The analysis does not account for the significant reduction in energy losses associated with on-off cycling of single stage compressors compared to two stage compressors. Several small losses are incurred when cycling. These can include motor starting power, refrigerant charge bleeding from the high to low side, reevaporation of moisture from the evaporator coil surface, and the initial delivery of warm, moist air when restarting, before the evaporator cools back down to the steady-state operating temperature. These losses are well understood (Dieckmann, 2011) but not easily captured with DOE-2.2.

3.1.1.1 Energy Analysis Methodology

This measure was evaluated by comparing a basecase unit with a single speed fan and single stage compressor to a proposed case unit with a two speed fan and a two stage compressor.

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

Two parametric runs, as described in **Error! Reference source not found.** were created then spliced together in a spreadsheet to create the proposed case. The basecase is equal to parametric run 1. Parametric runs 1 and 2 were spliced together using the sequence described in Section 3.1.1.1.10 to create the proposed case.

Parametric Run	Fan speed	Economizer
1	High (100%)	Partially-integrated
2	Low (66% airflow, 30% power)	Partially-integrated

Figure 2. Fan Control – Single Zone DX Parametric Runs

3.1.1.1.1 Building Envelope

The building used in this analysis is based on the ASHRAE 90.1 Prototype Building Modeling Specifications, as developed by the Pacific Northwest National Laboratory. These assumptions are consistent with Title 24-2008.

1. 5,400 square foot, single story building with an unconditioned attic. Aspect ratio is 1.5. See **Error! Reference source not found.**.
2. Floor to ceiling height is 10 feet. No plenum.
3. 4 perimeter zones, 1 core zone. See **Error! Reference source not found.**.
4. 16% WWR overall (11% on South façade, 25% on North façade, 15% on the East and West façades).
5. Windows are double-paned, low-e, with a center-of-glass U-value of 0.486, SHGC of 0.54, and VT of 0.77.
6. Exterior wall construction is R-13. Roof is R-30.
7. No skylights. No daylighting controls.

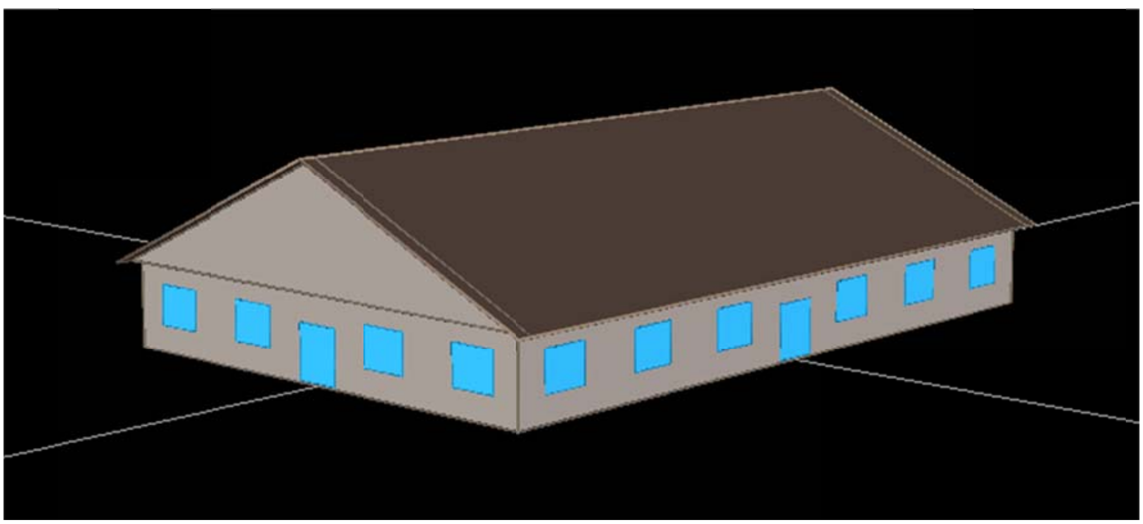


Figure 3. View of northeast corner of the 5400 square foot office building model

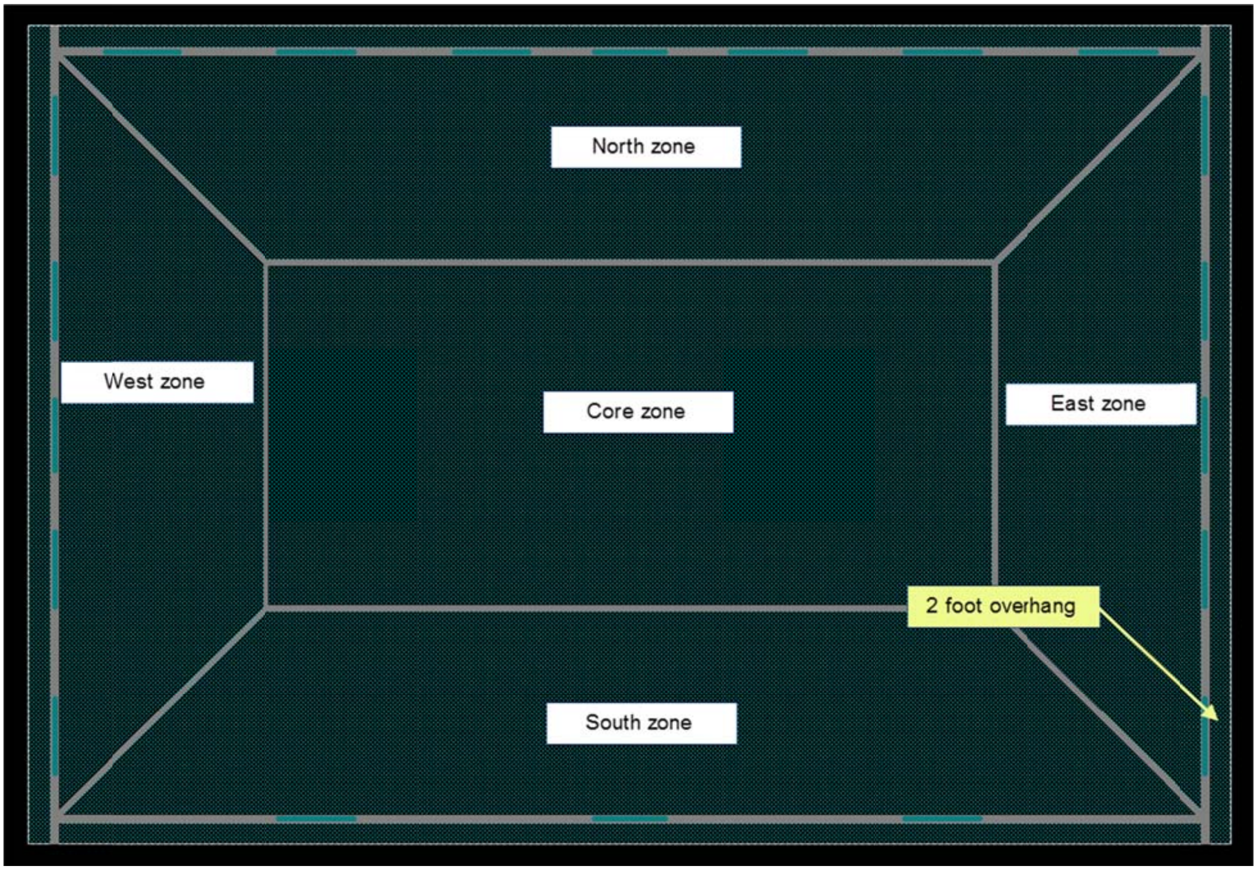


Figure 4. Overhead view of the 4 perimeter and 1 interior zones of the 5400 square foot office building model

3.1.1.1.2 Climate

The simulation was run in three climates that are representative of the climates where the majority of the State's projected new construction is located:

1. CZ03: Oakland
2. CZ06: Torrance (coastal Los Angeles)
3. CZ12: Sacramento

The weather files that were used in this simulation came from the California Energy Commission (CEC) and were developed for Title 24 – 2013. The simulation year used for all models was 2009, per Title 24 CASE requirements.

3.1.1.1.3 Internal Loads

All internal load values came from the PNNL ASHRAE 90.1 Prototype Building Modeling Specifications and are consistent with typical small office buildings in California.

1. Lighting power density: 1.0 W/ft^2
2. Equipment power density: 0.63 W/ft^2

3. Occupancy density: 200 ft^2/person

3.1.1.1.4 Schedules

The schedule assumes the building is occupied from 8am to 6pm on Monday through Friday, excluding holidays. The schedules for occupancy, lighting, and miscellaneous equipment follow the occupancy hours. The default eQuest schedules for a small office building were used because they are representative of typical small office buildings in California..

The fan schedule assumes the fans operate from 7am to 6pm on weekdays, excluding holidays, and 9am to 3pm on Saturdays. They cycle on during unoccupied hours only when they are needed to meet the setback temperatures. The minimum ventilation is zero even when fans cycle on during unoccupied hours.

3.1.1.1.5 Temperatures

The following temperature settings were used:

1. Setpoint temperatures: 75ºF cooling, 70°F heating
2. Setback temperatures: 80ºF cooling, 60°F heating
3. Supply Air Temperatures: 55ºF minimum for cooling, 104°F maximum for heating

3.1.1.1.6 System Properties

1. SYSTEM-TYPE: Packaged Single Zone with DX cooling and gas furnace for heating.
2. RETURN-AIR-PATH: Duct
3. Supply fan:
    a. SUPPLY-STATIC: 2.5”
    b. SUPPLY-EFF: 53%
    c. SUPPLY-MECH-EFF: 65%
    d. FAN-CONTROL: CONSTANT-VOLUME
    e. NIGHT-CYCLE-CTRL: CYCLE-ON-FIRST. The fans cycle on for the hour if the temperature in the zone goes out of range for heating and cooling setback temperatures during unoccupied hours.
4. Cooling:
    a. DX cooling
    b. MIN-SUPPLY-T: 55°F
    c. COOLING-EIR: 0.3496. Converted from the minimum efficiency of 9.7 SEER for a Unitary AC that is less than 65,000 Btu/h from Title 24 and ASHRAE Standard 90.1.
    d. CONDENSER-TYPE: AIR-COOLED.
    e. MIN-UNLOADING-RATIO: 1.0. Compressors only cycle, they do not modulate.
    f. MIN-HGB-RATIO: 1.0. No hot gas bypass.
    g. COOL-CTRL-RANGE: 0°F per Taylor Engineering’s Energy Modeling Standards

5. Heating:
   a. HEAT-SOURCE: Gas furnace.
   b. FURNACE HIR: 1.2407 Btu/Btu. Converted from 78% AFUE.
6. Outside Air:
   a. MIN-AIR-SCH: Set equal to a created schedule of type Frac/Design that has the value 0 during unoccupied hours and -999 during occupied hours. During occupied hours the system will default to the normal ventilation values. During unoccupied hours when the system cycles on to reach the setback temperatures, there is no outside air ventilation.
   b. 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.
   c. MAX-OA-FRACTION: 1.0
   d. DRYBULB-LIMIT: n/a (blank)
   e. ECONO-LOCKOUT: YES. The economizer is only locked out if it cannot meet the entire load. This is equivalent to a packaged unit that controls the economizer and the compressor separately off of supply air temperature. For example, if the design supply air temperature is 55°F but the load can be satisfied with 65°F supply air and the outside air temperature is < 65°F then the economizer operates. This effectively models a partially integrated economizer and is the appropriate modeling assumption for a unit without variable capacity or variable speed compressors. This is different from a non-integrated economizer that has a fixed low limit economizer lockout temperature, such as 60°F.

3.1.1.1.7 Zone Properties

1. OA-FLOW: 0.15 CFM per square foot was used to calculate minimum outside air:
   a. South and North: 182 CFM
   b. East and West: 108 CFM
   c. Core: 231 CFM
2. DESIGN-COOL-T: 75°F
3. DESIGN-HEAT-T: 70°F
4. TYPE-ZONE
   a. Perimeter and Core Zones: CONDITIONED
   b. Roof Zone: UNCONDITIONED
5. THROTTLING RANGE: 0.5°F per Taylor Engineering's Energy Modeling Standards

3.1.1.1.8 System Sizing for Full Speed Model

The auto-sizing feature in DOE-2 is not reliable. Therefore, the model was run iteratively: first it was run to determine the peak loads. Then the cooling system and fans were manually input into the model as 115% of the actual peak cooling load and 150% of the peak heating load to account for the availability of discrete equipment sizes. The model was originally run at 120% of the actual peak cooling load and 150% of the peak heating load. Dick Lord, a Senior Engineer at Carrier Corp., suggested using 115% over-sizing instead since it is more conservative than the 120% over-sizing ratio.

3.1.1.1.9 Low Speed Parametric Run

Next, a low speed case was added as a parametric run. This run was utilized to create the proposed case, which has two-speed fans and two-stage compressors. It is identical to the high speed base case model except:

1. The cooling capacity was sized as 50% of the high speed capacity, which simulates 2 stage cooling capacity.
2. The airflow was set to two-thirds of the high speed flow.
3. The fan power was modeled as 30% of the high speed flow, which is more conservative than a reduction of power down to 21.6% power at 66% speed if calculated using the cube law. 66% speed and 30% power are conservative. Most systems, particularly those with variable speed or EC motors, will be able to do better on both accounts.

With these modifications, the low speed run effectively models two speed fans and two stage compression.

3.1.1.1.10 Proposed Case using Spreadsheet Post-processing

The proposed case was modeled using spreadsheet post-processing of the high speed case and the low speed case output on an hourly basis.

Case	Compressor	Fan
Base	Single stage compressor	Single speed fan
Proposed	Two-stage compressor	Two-speed fan

Figure 5. Fan Control – Single Zone DX Base and Proposed Cases

In the proposed case, both the fan and the compressor drop to low speed when the low speed case can meet the entire cooling load. Additionally, if the fan and compressor use less energy in the high speed case than in the low speed case for that hour due to economizer benefits, it switches back to high speed (note: switching back to high speed does not significantly affect savings since this rarely occurs in the simulations). The following logic was used in the spreadsheet post-processing to create the proposed case:

- Fan and compressor energy equals the low speed case when:
    - The amount of unmet heating or cooling for the low speed = 0, and
    - Zone temperature is within the allowed throttling range

- Fan and compressor energy equals high speed case when:
    - The above requirements for low speed are not met, or
    - The fan + compressor energy for the high speed case is less than the fan + compressor energy for the low speed case, or
    - When the system is in heating mode

3.1.1.2 Energy Results

The cooling load profile for CZ03: Oakland is shown in **Error! Reference source not found.**. This is similar for other climates. This figure shows that the cooling load rarely exceed 50% of the design load, providing a large opportunity to provide the necessary cooling with a low speed fan.

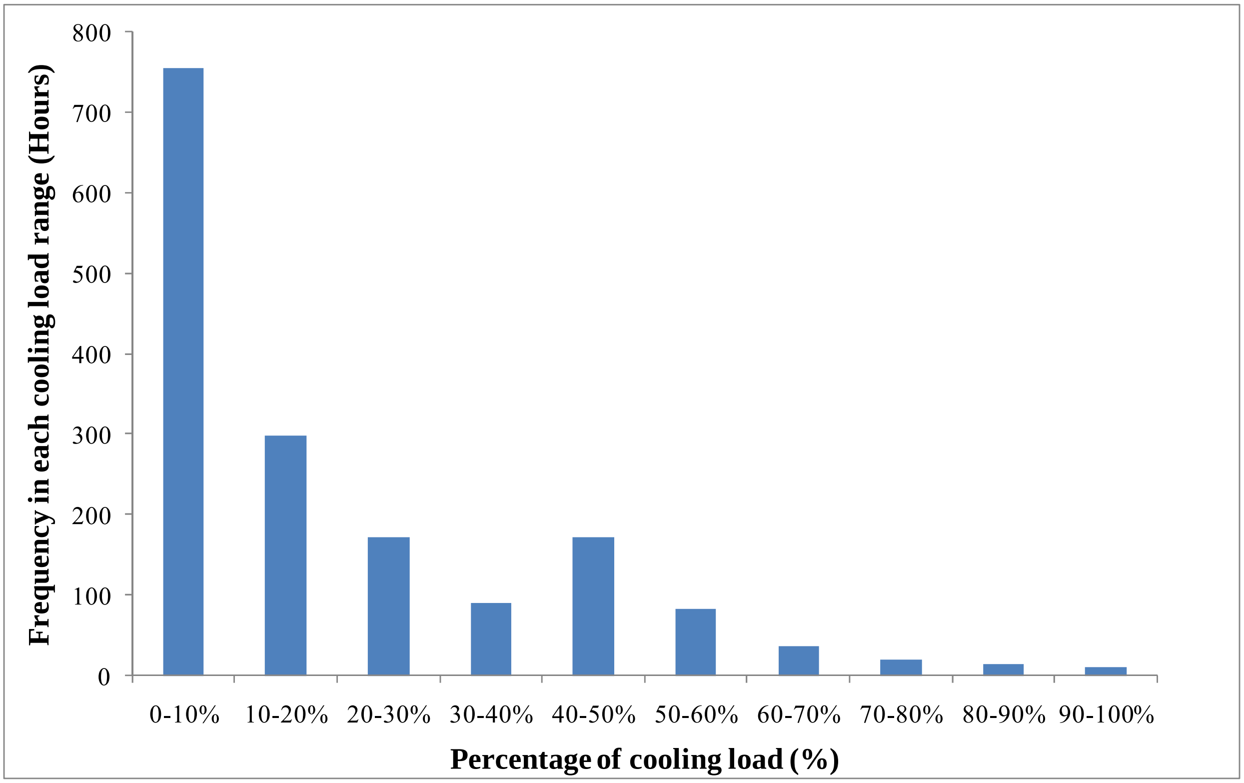


Figure 6. Cooling Load Profile for CZ03: Oakland

As shown in **Error! Reference source not found.**, the majority of the fan hours in the proposed case are low speed. Results are similar for other climate zones.

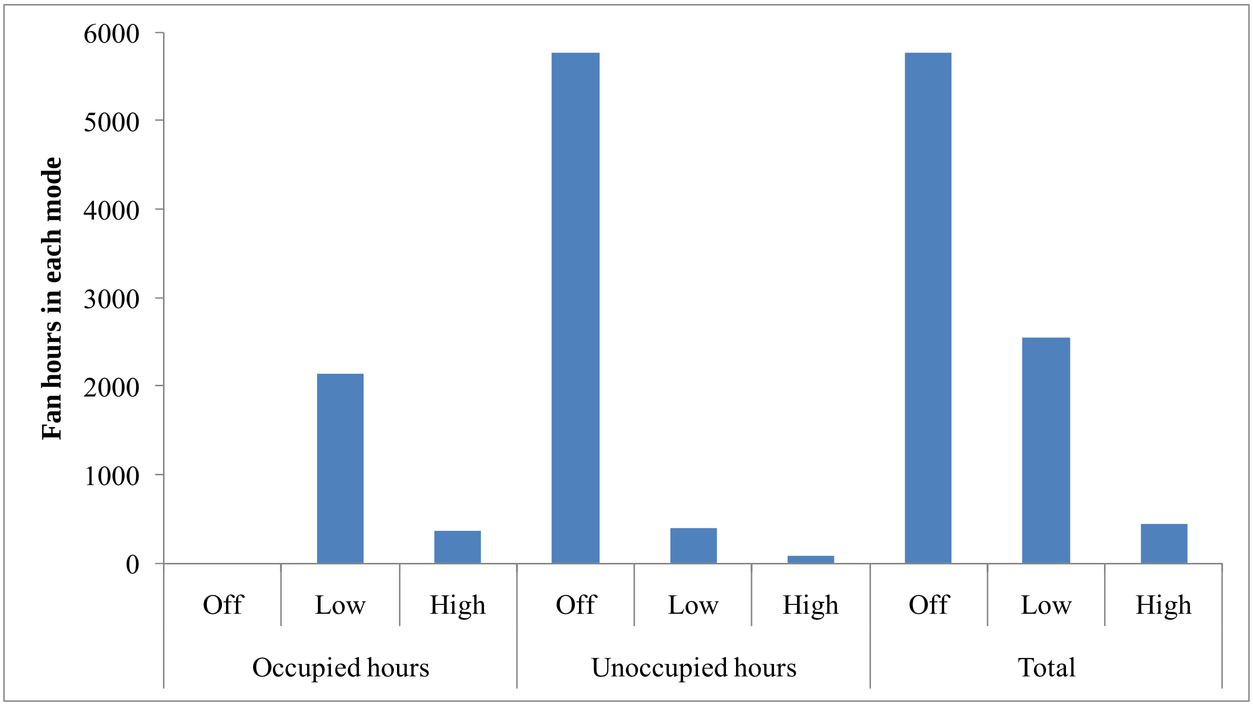


Figure 7. Fan Hours for Proposed Case for CZ06: Torrance

Error! Reference source not found. shows a warm day in CZ06: Torrance. The proposed case switches to high speed for one hour in the morning to increase free cooling; the total HVAC energy for the high speed case is lower than the low speed case for that hour due to compressor energy savings from utilizing free cooling from the economizer. The fan also switches to high speed for 3 hours mid-day when the cooling load cannot be met with the low speed case. Even on a warm day, when mid-day outside air temperatures exceed 80°F, the cooling load can be satisfied with the low speed fan for most of the day. The CFM per square foot for the high and low speed runs in the three climate zones are compared in **Error! Reference source not found.**.

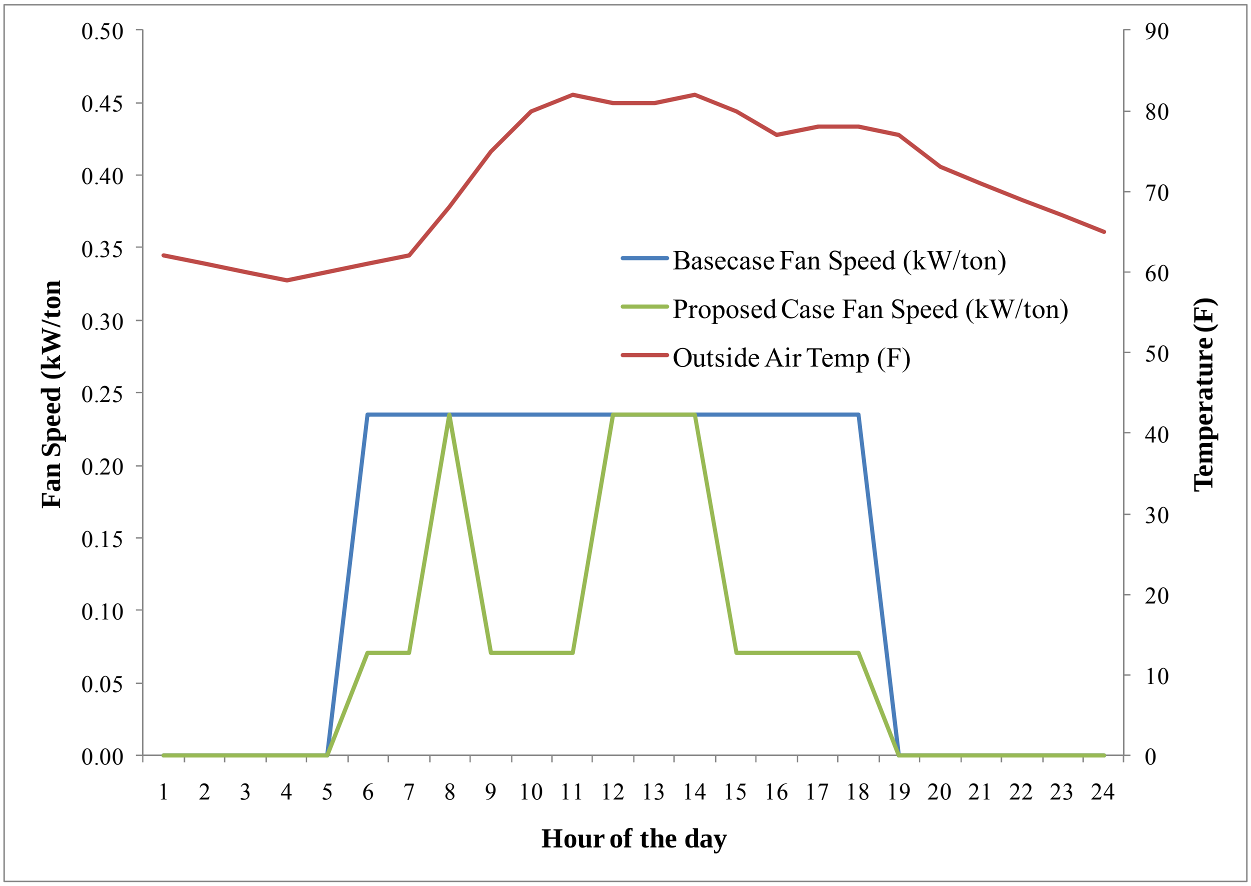


Figure 8. Warm Day in CZ06: Torrance

Location	ft2/ton	High speed run CFM/ft2	Low speed run CFM/ft2
CZ03: Oakland	503	0.83	0.55
CZ06: Los Angeles	495	0.84	0.56
CZ09: Sacramento	491	0.85	0.57

Figure 9. Fan Control – Single Zone DX Flow per square foot for High and Low Speed Runs

The total HVAC end-use energy consumption per ton is shown for three climate zones in **Error! Reference source not found.**. HVAC end-use energy is the sum of the fan, pump, and cooling energy.

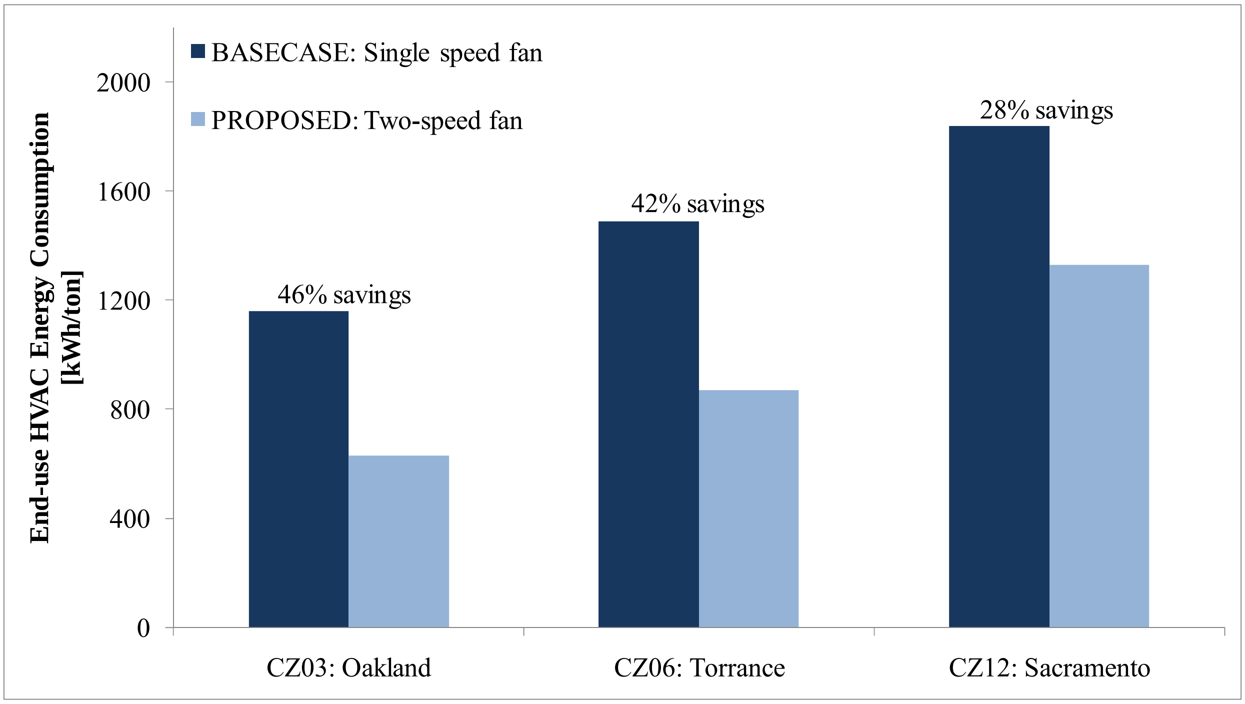


Figure 10. Fan Control – Single Zone DX End-use HVAC Energy Consumption: Base and Proposed Cases for Three Climate Zones

3.1.1.3 Measure Cost

Total measure cost include both incremental equipment cost and maintenance cost.

3.1.1.3.1 Incremental Installed Cost

Incremental cost data was provided by the AHRI Unitary Large Equipment (ULE) Committee in January 2011 based on a poll of the full AHRI ULE section and averaging the reported costs. The incremental cost for a compressor with two stages and a two-speed fan over a single stage compressor and a single speed fan is $496 for a 6 ton unit. This incremental cost includes additional installation, start-up and contractor mark up costs but does not include additional maintenance costs.

The AHRI ULE Committee incremental cost data represents current, national costs if this measure became required by ASHRAE 90.1. Taylor Engineering will be proposing the same measures described herein to 90.1 and there is a reasonable chance that it will be adopted by 90.1, particularly if it is adopted by Title 24. It is not clear if the incremental cost would increase if this was only required in California as opposed to nationally. It is clear, however, that the incremental cost in 2015 (the proposed effective date for this measure) will be lower than the current incremental cost as advancements continue to be made in the areas of variable frequency drives, EC motors, etc. It is reasonable to assume therefore that the incremental cost in 2015 will be lower than the current incremental cost, even if this is only adopted in California, but we have not accounted for lowered costs in our lifecycle cost effectiveness.

3.1.1.3.2 Maintenance Cost

An incremental maintenance cost data placeholder was provided by a Bay Area service contractor. It is conservatively estimated to be 1 hour per year at a labor rate of $100/hr. The incremental annual maintenance would have to increase dramatically to 4 to 6 hours per year, depending on the climate zone, to negate the energy cost savings.

3.1.1.4 Lifecycle Cost Results

As shown in **Error! Reference source not found.**, the measure is highly cost effective using TDV energy rates. Simple payback periods will be around 2 years.

	CZ03	CZ06	CZ12
Incremental Installed Cost	$496	$496	$496
Incremental Annual Maint.	$100	$100	$100
NPV of Annual Maint.	$1,190	$1,190	$1,190
Total Incremental Cost	$1,686	$1,686	$1,686
NPV of Energy Savings	$7,320	$7,860	$5,520
Lifecycle cost savings	$5,634	$6,174	$3,834

Figure 11. Fan Control – Single Zone DX: Lifecycle Cost Results for a 6 Ton Unit

3.2 Fan Control – Single Zone CHW Fan Coil

The second measure and system analyzed is variable speed control for single zone CHW fan coils. There are basically three types of chilled water fan systems that this measure would apply to:

- Small fan coils without outside air
- Small air handlers with minimum outside air but below the size where an economizer is required
- Medium to large air handlers where an economizer is required

The incremental cost for a medium to large air handler with an economizer will be about the same as a small air handler without an economizer but the energy savings will be much greater. Therefore, if the measure is cost effective for the first two types of CHW systems then it is clearly cost effective for larger systems as well. Small fan coils without outside air is analyzed in this section. Small air handlers with minimum outside air are analyzed in section 3.3.

Typical applications for small fan coils without outside air would be an IDF (computer) closet or electrical room, i.e. small systems with 24/7 operation and no outside air.

This system was evaluated by comparing a base case unit with a single speed fan to a proposed case unit with a variable speed fan with a minimum fan speed of 50% of the design fan speed. This analysis assumes that this is achieved by using an electronically commutated motor (ECM) instead of a standard motor, such as a permanent split capacitor (PSC).

The energy analysis is conservative for the following reasons:

- It does not take credit for reduced fan heat cooling energy.
- It does not take credit for the increased motor efficiency of an ECM versus a standard PSC motor

3.2.1.1 Energy Analysis

Since the loads are entirely internal in this application and credit is only being taken for fan energy savings it was not necessary to run a DOE-2 simulation in multiple climates. Thus the energy analysis was completed in a spreadsheet. An IDF closet with 24/7 operation and no outside air was modeled. The cooling load is assumed to be evenly distributed between 25%, 50%, 75%, and 100% of the design load, such as the load distribution shown in **Error! Reference source not found.**.

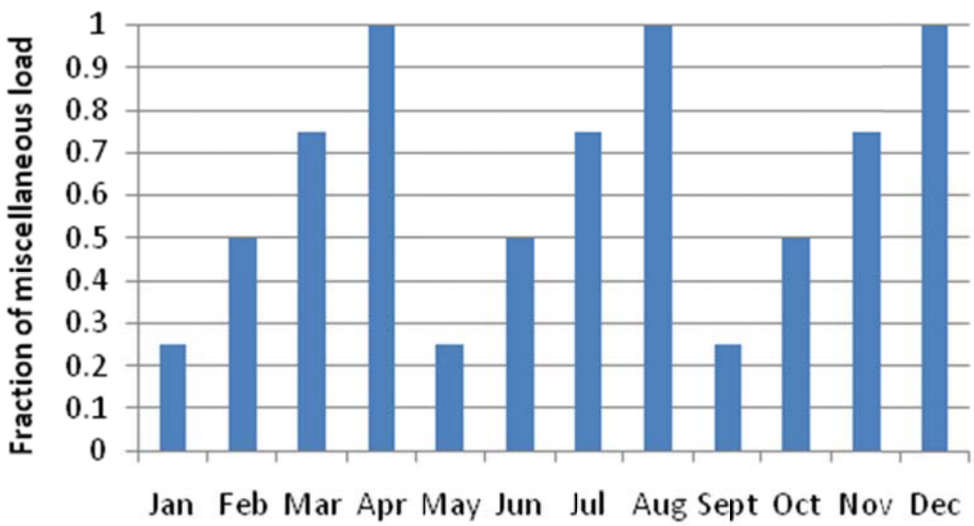


Figure 12. Fan Control – Single Zone CHW Fan Coil: Example Cooling Load Distribution

The actual space size and actual space loads are not relevant because the fan system is sized for the actual load (plus oversizing). All that matters is the load profile on the fan system. This tells us the savings per horsepower. The net present value of the energy savings is then compared to the incremental cost for various size fan systems to find the smallest size system at which the measure is cost effective.

The following assumptions were used in this analysis:

- Break horsepower (BHP) is 85% of motor horsepower (MHP) on average
- HVAC system is oversized by 120% to account for conservatism and since small fan systems are only available in discrete sizes.
- Fan speed is proportional to the load
- Cube law fan power
- Minimum fan speed is 50%

3.2.1.2 Energy Results

The energy savings for one IDF closet served by a single fan are shown in **Error! Reference source not found.**.

Basecase annual fan energy	1,388 kWh/yr
Proposed case annual fan energy	1,088 kWh/yr
Energy savings	372 kWh/yr
Peak power savings	0.07 kW

Figure 13. Fan Control – Single Zone CHW Fan Coil: Annual Energy Savings for a ¼ HP Fan Motor

3.2.1.3 Measure Cost

Total measure cost include both incremental equipment cost and maintenance cost.

3.2.1.3.1 Incremental Installed Cost

Incremental cost data was provided by a Bay Area service contractor. The incremental cost for an electronically commutated motor (ECM) over a standard motor, such as a permanent split capacitor (PSC), is $140 for a ¼ HP motor. The contractor mark-up was conservatively estimated to be 30%. The incremental start-up and commissioning cost was conservatively estimated to be $100. No additional controls are included since fan speed can be controlled off the CHW valve signal.

Incremental motor cost	$140
Contractor markup (30%)	$42
Start-up/Commissioning	$100
Total incremental cost	$282

Figure 14. Fan Control – Single Zone CHW Fan Coil: Incremental Cost for ¼ HP Motor

3.2.1.3.2 Maintenance Cost

Incremental maintenance cost data was provided by a Bay Area service contractor. It is conservatively estimated to be an additional half hour per year at a labor rate of $100/hr.

3.2.1.4 Lifecycle Cost Results

As shown in **Error! Reference source not found.**, the measure is highly cost effective in all climate zones using TDV energy rates. The energy savings in kwh/yr are the same in each climate zone but the energy savings in dollars are different because each climate zone has a different average TDV rate.

	CZ01	CZ02	CZ03	CZ04	CZ05	CZ06
Incremental Installed Cost	$282	$282	$282	$282	$282	$282
Incremental Annual Maint.	$50	$50	$50	$50	$50	$50
NPV of Annual Maint.	$595	$595	$595	$595	$595	$595
Total Incremental Cost	$877	$877	$877	$877	$877	$877
NPV of Energy Savings	$1,948	$1,929	$1,932	$1,930	$1,937	$1,913
Lifecycle cost savings	$1,071	$1,052	$1,055	$1,053	$1,060	$1,036

	CZ07	CZ08	CZ09	CZ10	CZ11	CZ12
Incremental Installed Cost	$282	$282	$282	$282	$282	$282
Incremental Annual Maint.	$50	$50	$50	$50	$50	$50
NPV of Annual Maint.	$595	$595	$595	$595	$595	$595
Total Incremental Cost	$877	$877	$877	$877	$877	$877
NPV of Energy Savings	$1,948	$1,921	$1,911	$1,911	$1,940	$1,940
Lifecycle cost savings	$1,071	$1,044	$1,034	$1,034	$1,063	$1,063

	CZ13	CZ14	CZ15	CZ16
Incremental Installed Cost	$282	$282	$282	$282
Incremental Annual Maint.	$50	$50	$50	$50
NPV of Annual Maint.	$595	$595	$595	$595
Total Incremental Cost	$877	$877	$877	$877
NPV of Energy Savings	$1,945	$1,917	$1,922	$1,926
Lifecycle cost savings	$1,068	$1,040	$1,045	$1,049

Figure 15. Fan Control – Single Zone CHW Fan Coil: Lifecycle Cost Results for a ¼ HP Fan

3.3 Fan Control – Single Zone CHW AHU

The third measure and system analyzed is variable speed control for single zone CHW air handlers below the economizer size threshold. A typical application for such a system would be an office zone. This system was evaluated by comparing a base case unit with a single speed fan to a proposed case unit with a variable speed fan with a minimum fan speed of 50% of the design fan speed. This analysis assumes that this is achieved by using an electronically commutated motor (ECM) or a variable frequency drive (VFD) instead of a standard motor, such as a permanent split capacitor (PSC). Additionally, a modulating actuator is required to maintain minimum outside air as the supply flow varies. The basecase is a fixed position minimum outside air damper.

Typical applications for this measure would be a small auditorium AHU, classroom unit ventilator, or hotel room fan coil.

The energy analysis is conservative for the following reasons:

- It does not take credit for reduced fan heat cooling energy.
- It does not take credit for the increased motor efficiency of an ECM versus a standard PSC motor.

3.3.1.1 Energy Analysis Methodology

The base case in this analysis is a single zone CHW AHU with a constant volume fan and minimum outside air. The proposed case is a single zone CHW AHU with a variable volume fan, minimum fan speed of 50%, and minimum outside air.

Case	Fan Control	Minimum Fan Speed	Economizer
Base case	Constant	100%	None – Minimum outside air
Proposed case	Variable	50%	None – Minimum outside air

Figure 16. Fan Control – Single Zone CHW AHU: Base and Proposed Case Inputs

The majority of the energy savings from this measure will be from fan savings. A minor amount of savings will come from cooling savings. To simplify the analysis and increase consistency between analyses, the fan savings from the eQuest model used in Section 3.5 “Combined Measures: Fan Control and Integrated Economizer – Single Zone DX” was used in this analysis with the following input:

3.3.1.1.1 Building Envelope

See Section 3.1.1.1.1.

3.3.1.1.2 Climate

The simulation was run in three representative climate zones:

1. CZ03: Oakland
2. CZ06: Torrance (coastal Los Angeles)
3. CZ12: Sacramento

The weather files that were used in this simulation came from the California Energy Commission (CEC) and was developed for Title 24 – 2013. The simulation year used for all models was 2009, per Title 24 CASE requirements.

3.3.1.1.3 Internal Loads

1. Lighting power density: 1.0 W/ft^2
2. Equipment power density: 1.5 W/ft^2

3. Occupancy density: 200 ft^2/person

3.3.1.1.4 Schedules

See Section 3.1.1.1.4.

3.3.1.1.5 Temperatures

See Section 3.1.1.1.5.

3.3.1.1.6 System Properties

1. SYSTEM-TYPE: PVVT. The model had a Packaged Single Zone with DX cooling and gas furnace for heating per zone. Then this was changed to PVVT in a parametric run that became the basecase.
2. MIN-FLOW-RATIO: 1.0 to model a constant volume unit.
3. RETURN-AIR-PATH: Duct
4. Supply fan:
   a. SUPPLY-STATIC: 2.5”
   b. SUPPLY-EFF: 53%
   c. SUPPLY-MECH-EFF: 65%
   d. FAN-CONTROL: CONSTANT-VOLUME
   e. MIN-FLOW-RATIO: 1.0.
   f. NIGHT-CYCLE-CTRL: CYCLE-ON-FIRST. The fans cycle on for the hour if the temperature in the zone goes out of range for heating and cooling setback temperatures during unoccupied hours.
5. Cooling:
   a. DX cooling
   b. MIN-SUPPLY-T: 55°F
   c. COOLING-EIR: 0.3496. Converted from the minimum efficiency of 9.7 SEER for a Unitary AC that is less than 65,000 Btu/h from Title 24 and ASHRAE Standard 90.1.
   d. CONDENSER-TYPE: AIR-COOLED.
   e. MIN-UNLOADING-RATIO: 1.0. Compressors only cycle, they do not modulate.
   f. MIN-HGB-RATIO: 1.0. No hot gas bypass.
   g. COOL-CTRL-RANGE: 0°F per Taylor Engineering’s Energy Modeling Standards
6. Heating:
   a. HEAT-SOURCE: Gas furnace.
   b. FURNACE HIR: 1.2407 Btu/Btu. Converted from 78% AFUE.
7. Outside Air:
   a. MIN-AIR-SCH: Set equal to a created a schedule of type Frac/Design that has the value 0 during unoccupied hours and -999 during occupied hours. During occupied hours the system will default to the normal ventilation values. During

unoccupied hours when the system cycles on to reach the setback temperatures, there is no outside air ventilation.

b. OA-CONTROL: OA-TEMP.
c. MAX-OA-FRACTION: 1.0
d. DRYBULB-LIMIT: 60°F
e. ECONO-LOCKOUT: NO. The compressor(s) can operate simultaneously with the economizer to meet the cooling load. The economizer only shuts off when the outside air is warmer than 60°F.

3.3.1.1.7 Zone Properties

1. OA-FLOW: 0.15 CFM per square foot was used to calculate minimum outside air:
    a. South and North: 182 CFM
    b. East and West: 108 CFM
    c. Core: 231 CFM
2. DESIGN-COOL-T: 75°F
3. DESIGN-HEAT-T: 70°F
4. TYPE-ZONE
    a. Perimeter and Core Zones: CONDITIONED
    b. Roof Zone: UNCONDITIONED
5. THROTTLING RANGE: 0.5°F per Taylor Engineering's Energy Modeling Standards

3.3.1.1.8 System Sizing for Full Speed Model

The auto-sizing feature in DOE-2 is not reliable. Therefore, the model was run iteratively: first it was run to determine the peak loads. Then the cooling system, fan flow, and heating system were manually input into the model as 115% of the actual peak cooling load to account for the availability of discrete equipment sizes.

3.3.1.1.9 Proposed Case Fan Control

1. Proposed case Supply fan:
    a. FAN-CONTROL: FAN-EIR-FPLR. Fans are variable and ride the zero fixed static fan curve.
    b. FAN-EIR-FPLR: Zero fixed static fan curve
    c. MIN-FLOW-RATIO: 0.5. The fans are variable down to 50% design fan speed.

3.3.1.1.10 Outside Air

Both the base and proposed cases do not have an economizer and only minimum outside air. Thus, OA-CONTROL is set to FIXED in both cases.

3.3.1.1.11 Zone Area

This measure applies on a zonal basis. The prototype building is 5,400 square feet. It is assumed that each zone is 540 square feet, which is a typical zone size, thus there are 10 zones per prototype building that are assumed to be evenly distributed throughout the building. The results of the analysis were divided by ten to change them to a per zone basis, thus they represent the average savings per zone, despite its location in the building.

3.3.1.2 Energy Results

The cooling load is dominated by hours below 50% of the design cooling load, as shown in **Error! Reference source not found.**. The vast majority of the fan hours in the proposed case fall in the 50-60% speed range, since the fan speed was limited to a 50% minimum speed, as shown in **Error! Reference source not found.**.

Error! Reference source not found. shows the fan energy savings for this measure. This analysis is conservative since the reduction in cooling energy from reduced fan heat is not included. Since the vast majority of the fan hours are spent at the 50-60% fan speed range, the energy consumed by the fan is reduced by around 80% when the fans are switched from constant to variable volume.

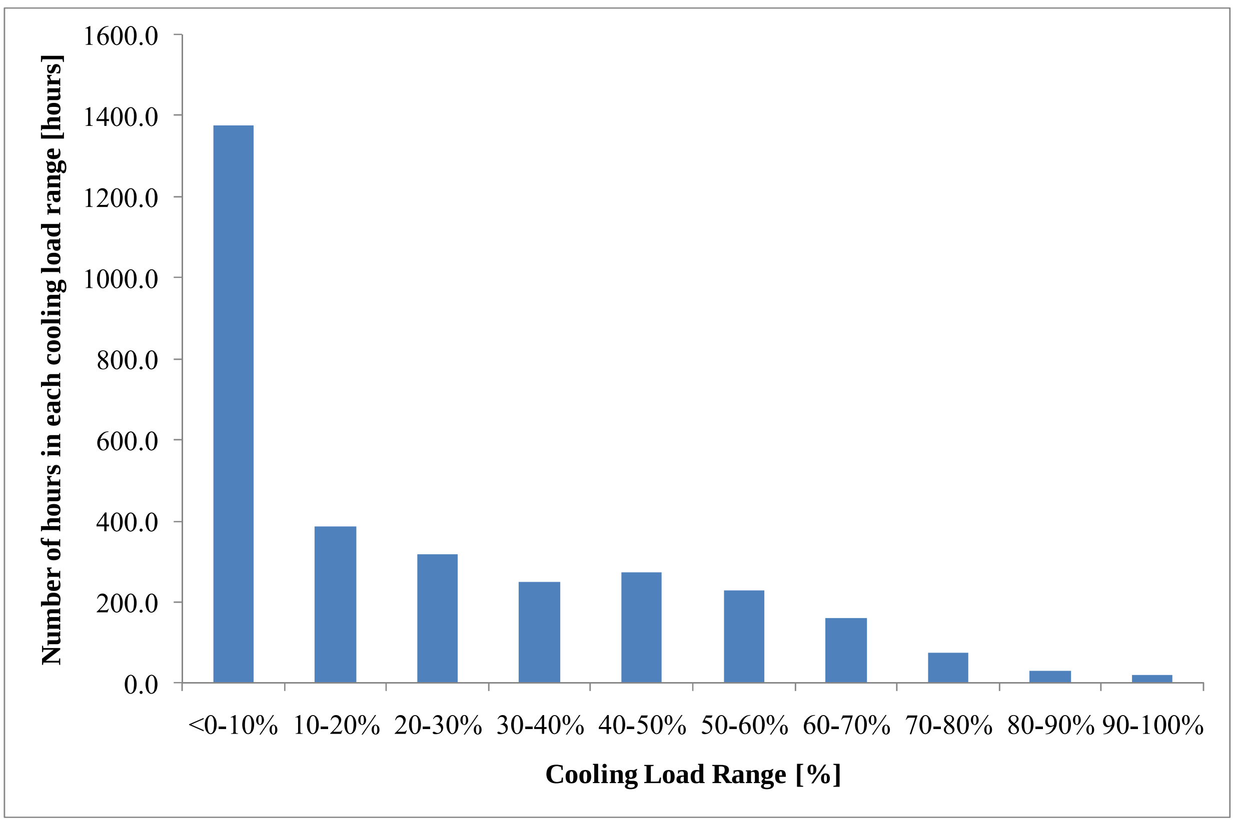


Figure 17. Fan Control – Single Zone CHW AHU: Number of hours in each cooling load range for a representative zone

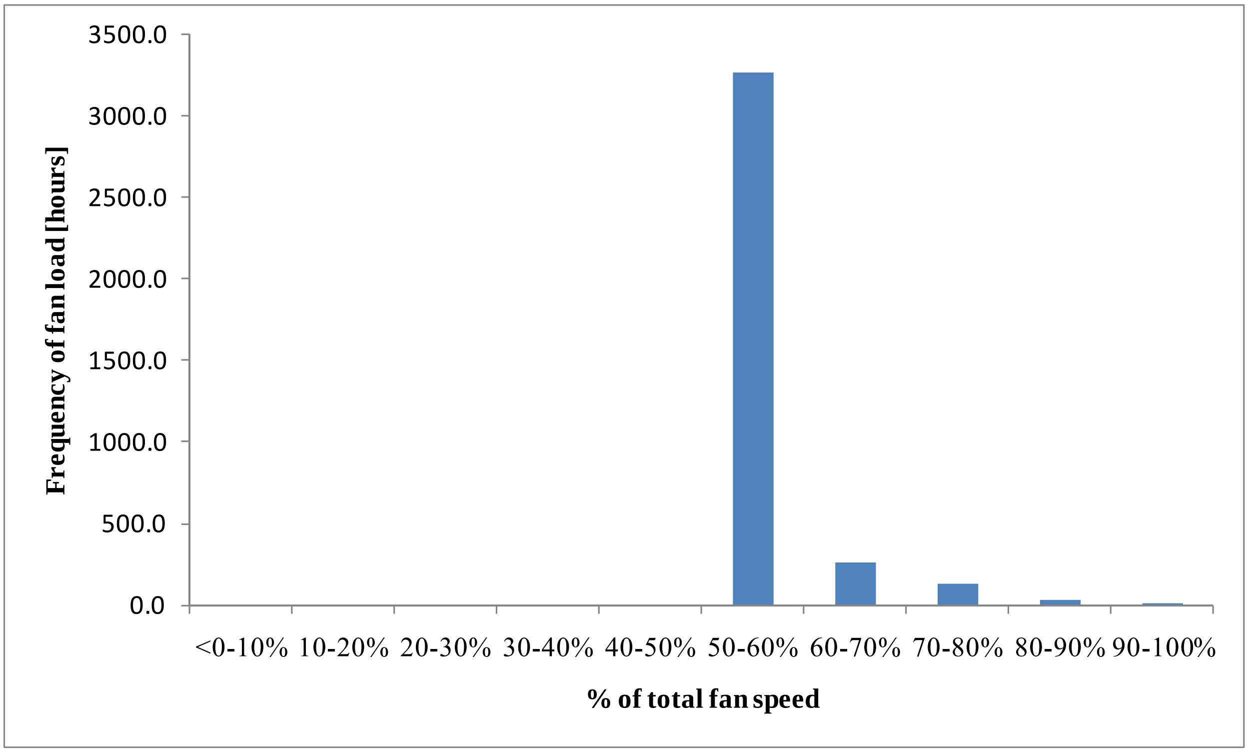


Figure 18. Fan Control – Single Zone CHW AHU: Number of hours in each fan speed range for the south zone

	CZ03	CZ06	CZ12
Fan savings per prototype building [kWh/yr]	8,451	8,939	10,402
Fan savings per 540 sf zone [kWh/yr]	845	894	1,040
Fan savings [%]	82%	79%	80%

Figure 19. Fan Control – Single Zone CHW AHU: Energy Analysis Results for Fan Energy

3.3.1.3 Measure Cost

Total measure cost include both incremental equipment cost and maintenance cost.

3.3.1.3.1 Incremental Installed Cost

Incremental cost data was provided by a Bay Area service contractor and also a valve actuator supplier. The incremental cost for a variable fan motor and a modulating actuator for a single zone CHW unit is currently $770. This includes contractor mark-up, additional start-up, commissioning, control, and balancing. Incremental cost data has been presented to stakeholders at stakeholder meetings and at ASHRAE meetings and there was general agreement that the costs are conservative.

Small variable speed single zone chilled water units are not common. It is estimated that less than 10% of small single zone chilled water units today in California are variable speed. Accordingly the incremental costs are relatively high for such a specialty product. Incremental costs will come down dramatically for variable speed systems once the requirement goes into effect. Note that the existing requirement in Title 24-2008 for single zone VAV does not go into effect until 1/1/2012. The future incremental cost was conservatively estimated to be 30% lower than the current incremental cost, at $539 per unit. This predicted future incremental cost is not used in the lifecycle cost analysis in this section but is used for the projected future incremental cost in **Error! Reference source not found.**.

ECM/VFD cost	$200
modulating actuator cost	$200
Contractor markup	30%
Add for start-up/Cx	$100
Add for controls	$100
Add for balancing	$50
Total incremental cost	**$770**

Figure 20. Fan Control – Single Zone CHW AHU: Current Incremental Cost Data

3.3.1.3.2 Maintenance Cost

Incremental maintenance cost data was provided by a Bay Area service contractor. It is estimated to be half an hour per year at a labor rate of $100/hr. Incremental maintenance will also come down in the future once the measure is adopted.

3.3.1.4 Lifecycle Cost Results

As shown in **Error! Reference source not found.**, the measure is cost effective assuming current costs and TDV energy rates.

	CZ03	**CZ06**	**CZ12**
Incremental Installed Cost	$770	$770	$770
Incremental Annual Maint.	$50	$50	$50
NPV of Annual Maint.	$595	$595	$595
Total Incremental Cost	$1,365	$1,365	$1,365
NPV of Energy Savings	$1,828	$1,898	$2,194
Lifecycle cost savings	$463	$533	$829

Figure 21. Fan Control – Single Zone CHW AHU: Lifecycle Cost Results per Zone

3.4 Integrated Economizer – Multiple Zone DX

This analysis examines the requirements of the integrated economizer on packaged VAV-reheat units. When fully integrated economizers cannot satisfy the entire cooling load, compressors will cycle on to satisfy the rest of the load. However, since compressors typically come on in discrete steps and have minimum run times, this often leads to economizer cycling. The economizer closes temporarily to prevent freezing the evaporator coil before the compressor can reach its minimum run time and cycle off. Some packaged unit controls attempt to keep the economizer dampers open longer by allowing the supply air temperature to go well below design temperature before closing the economizer. It is not clear if this is actually more efficient than closing the economizer because excessively cold supply air means higher than necessary latent loads and higher compressor head pressure, i.e. lower compressor efficiency.

The phenomenon of economizer cycling is well known. We at Taylor Engineering have seen it on most projects with DX equipment, even on very large DX units. Average economizer damper positions of 50%-75% open are common on packaged units in economizer mode (see Appendices 7.3, 7.4, and 7.5)

In order to take full advantage of economizer savings, the economizer needs to be more fully integrated. Multiple zone VAV units typically have at least 2 compressors. If at least one of those compressors is a variable speed compressor or a variable capacity compressor (e.g. digital scroll) then the minimum compressor capacity will typically be less than 10-20% of the maximum capacity. The Copeland Digital Scroll, for example, has a 10% minimum capacity and the cycle time can be 10 seconds, i.e. the compressor can be loaded for one of every 10 seconds. With turndown in this range economizer cycling can effectively be eliminated.

This analysis is conservative because it only takes credit for economizer savings and does not take credit for compressor efficiency savings. Compressor efficiency savings come from two main areas:

Reduced Cycling - Several small losses are incurred when cycling. These can include motor starting power, refrigerant charge bleeding from the high to low side, reevaporation of moisture from the evaporator coil surface, and the initial delivery of warm, moist air when restarting, before the evaporator cools back down to the steady-state operating temperature. These losses are well understood but not easily captured with DOE-2.2.

More Effective Use of Heat Exchangers – Continuous operation of cooling equipment at reduced capacity, instead of on-off operation at full capacity, results in less temperature lift and hence increased COP. When less than full capacity is needed operating in on-off mode results in the condensing and evaporating temperatures being close to the level they would be if running continuously at full capacity. During the off cycle, the heat transfer capacity of the heat exchanger is unused. By comparison, when a system runs continuously at say 50% capacity, the difference between the condensing temperature and the outdoor temperature will be only half of the difference at full capacity, significantly raising the EER of the compressor. For variable speed and variable capacity compressors the efficiency tends to be maintained close to the full load efficiency over the entire capacity range (see Appendix 7.1)

3.4.1.1 Energy Analysis

The base case in this analysis is an office building served by a multiple zone packaged unit with a typically performing economizer, which is modeled by only accounting for a 75% of the economizer savings from a fully integrated economizer in DOE-2.2. 75% is a reasonably conservative basecase assumption based on field observations of economizers as discussed above. The proposed case is the same building with a fully integrated, thus properly functioning, economizer.

The model is a 10,000 square foot generic office building. It is based on a model that was developed for evaluating the Title 24 – 2013 Reduce Reheat Measure and follows the Title 24 ACM manual basecase design building. This model was used to best represent a multiple zone building. Thus, some of the inputs in the measure vary slightly from those in the single zone DX Sections 3.1 and 3.5.

Parametric run 1, which is an office building that does not have an economizer, is described in the sections below. Parametric run 2 has a fully integrated economizer. The difference between the non-economizer run and the fully integrated economizer parametric run represents the total potential economizer savings. The base case was calculated by de-rating the total potential economizer savings by 25% on an hourly basis. The proposed case is equal to parametric run 2, which has a fully integrated economizer.

eQuest version 3.63b, build 6510 was used to perform the simulation runs. DOE-2.2 is the calculation engine.The eQuest energy model for parametric run 1 are described below. Parametric 2 is described in Section 3.4.1.1.9.

3.4.1.1.1 Building Envelope

1. Single story, 10,000 ft^2 square building. 4 perimeter zones (each 1,275 ft^2) and 1 interior zone (4,900 ft^2). Floor to floor height is 12 feet, plenum height is 3 feet.
2. Continuous strip of glazing, double pane, low-e glass (DOE-2 code 2004 for north-facing glass, 2203 for all other exposures). 38% WWR.
3. 15-foot deep perimeter zones.
4. Exterior wall construction is R-13. Roof is R-19.
5. No skylights, no daylighting controls.

3.4.1.1.2 Climate

The simulation was run in three representative climates:

1. CZ03: Oakland
2. CZ06: Torrance (coastal Los Angeles)
3. CZ12: Sacramento

The weather file that was used in this simulation for the Los Angeles run came from the California Energy Commission (CEC) and was developed for Title 24 – 2013. The simulation year used for all models was 2009, per Title 24 CASE requirements.

3.4.1.1.3 Internal Loads

1. Lighting power density: 0.9 W/ft^2
2. Equipment power density: 1.5 W/ft^2
3. Occupancy density: 100 ft^2/person

3.4.1.1.4 Schedules

Realistic office occupancy load profiles were generated. Five different schedules were generated and randomly assigned to each zone on different days. This was done to accurately model the net effect on a multiple zone air handler. For simplicity the same schedules were used for lights, people, and equipment. These five schedules are displayed in Appendix 7.7 "Load Profiles for Integrated Economizer".

The fans operate for 13 hours on weekdays from 6am to 7pm. They cycle on during unoccupied hours only when they are needed to meet the setback temperatures. The minimum ventilation is zero when fans cycle on during unoccupied hours.

3.4.1.1.5 Thermostat Setpoints

75°F cooling, 70°F heating

3.4.1.1.6 Setback Temperatures

82ºF cooling, 64°F heating

3.4.1.1.7 System Properties

1. SYSTEM-TYPE: Powered Induction Unit with DX cooling and hot water heating.
2. RETURN-AIR-PATH: PLENUM-ZONES.
3. Supply fan:
    a. SUPPLY-STATIC: 3.5"
    b. SUPPLY-EFF: 53%
    c. FAN-CONTROL: SPEED. The motor speed varies.
    d. MIN-FAN-RATIO: 0.10; MAX-FLOW-RATIO: 1.10
    e. NIGHT-CYCLE-CTRL: CYCLE-ON-FIRST. The fans cycle on for the hour if the temperature in the zone goes out of range for heating and cooling setback temperatures during unoccupied hours.
4. Outside air:
    a. OA-CONTROL: FIXED FRACTION
    b. MIN-AIR-SCH: Set equal to a manually created schedule of type Frac/Design that has the value 0 during unoccupied hours and -999 during occupied hours. During occupied hours the system will default to the normal ventilation values.

During unoccupied hours when the system cycles on to reach the setback temperatures, there is no outside air ventilation.

5. Cooling:
    a. COOL-SOURCE: ELEC-DX
    b. EIR: 0.2580. Converted from an EER of 11.0, which is the minimum required by Title 24 for units between 115,000 Btu/h and 240,000 kBtu/h.
    c. MIN-SUPPLY-T: 55°F
    d. COOL-CONTROL: WARMEST. The supply air temperature will reset when in cooling mode.
    e. RESET-PRIORITY: SIMULTANEOUS. This is the most realistic SAT reset method. Having no SAT reset would understate the savings. Using temperature-first as the reset method would overstate the savings.
    f. COOL-MAX-RESET-T: 65°F
    g. COOL-MIN-RESET-T: 55°F
    h. MIN-UNLOAD-RATIO: 1.0. The compressor can run at any PLR instead of cycling.
    i. MIN-HGB-RATIO: 0.
6. Heating:
    a. HEAT-SOURCE: HOT-WATER
    b. ZONE HEAT-SOURCE: HOT-WATER
    c. REHEAT-DELTA-T: 35°F.
    d. HEAT-CONTROL: n/a. Thus, there is no supply air temperature reset during heating mode.

3.4.1.1.8 Zone Properties

1. Outdoor air calculated based on the larger of 15 cfm/person and 0.15 cfm/sqft. In this case, since the occupant density is 100 sqft/person, the outdoor air required based on 15 cfm/person and 0.15 cfm/sqft is the same.
2. TERMINAL-TYPE: Std VAV Terminal
3. THERMOSTAT-TYPE: Reverse Action. For VAV systems, this thermostat type behaves like a dual maximum thermostat, it allows the airflow rate to rise above the minimum design heating airflow rate.
4. THROTTLING-RANGE: 5°F. A higher throttling range must be used when specifying supply air temperature reset that is based on the warmest zone for a variable air volume system. The warning in eQuest:

    "*If using the COLDEST or WARMEST options in conjunction with a variable air volume system, there are two actions within the throttling range. To reflect reality and to prevent instability in the simulation, THROTTLING-RANGE should be increased to 4-6F (2-3K).*"

5. MIN-FLOW-RATIO: The box minimum is set to 20% for the perimeter zones and 35% for the core zone. The ventilation requirement is the greater of 0.15 cfm/sqft and 15 cfm/person. At the given occupant density of 100 sqft/person, the ventilation required based on area and based on people is the same. The minimum flow ratio for each zone is the greater of the box minimum and the ventilation requirement.

3.4.1.1.9 Parametric Run 2: Add an Integrated Economizer

A parametric run was created that has an integrated economizer. The end-use energy hourly reports were used to determine the economizer savings. The difference between the run without an economizer and the parametric run with an integrated economizer results in the economizer savings.

1. Outside Air System Properties for Parametric Run 2
   a. 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.
   b. MAX-OA-FRACTION: 1.0
   c. DRYBULB-LIMIT: n/a (blank)
   d. ECONO-LOCKOUT: NO. This models a fully integrated economizer. If the economizer cannot meet the entire cooling load, it can still remain on to meet part of the load. The compressor(s) can operate simultaneously with the economizer to meet the remaining cooling load. The economizer is only locked out when the outside air is warmer than the return air.

3.4.1.1.10 Spreadsheet Analysis

The hourly reports for the non-economizer case and the fully integrated economizer case were exported to a spreadsheet for post-processing. The output for the fully integrated economizer run was subtracted from the non-economizer run for each hour, broken down by HVAC end-use type: cooling, fans, and pumps. This is the economizer savings. To model economizer cycling, the savings were de-rated by 25% by multiplying each end-use value by 0.75 on an hourly basis. This value was subtracted from the non-economizer case to result in the base case. Thus, the base case has 75% of the fully integrated economizer savings. The proposed case is equal to the parametric run, which is the fully integrated economizer run, thus has 100% of the fully integrated economizer savings.

3.4.1.2 Energy Results

The design flow per square foot for each climate zone are shown in **Error! Reference source not found.** and are all similar and reasonable values. **Error! Reference source not found.** shows the cooling end-use energy in the three climate zones. Oakland saves the most cooling energy, 13%, out of the three climate zones modeled. Cooling energy dominates the HVAC energy end-use, as shown in **Error! Reference source not found.**. Thus, 13% cooling energy savings in Oakland equates to 11% HVAC energy savings. The reduction in cooling energy is proportional to the reduction of hours that require mechanical cooling. Hours of cooling for each climate zone are shown in **Error! Reference source not found.**.

Location	CFM/ft2
CZ03: Oakland	0.84
CZ06: Torrance	0.91
CZ12: Sacramento	0.93

Figure 22. Integrated Economizer – Multiple Zone DX: Flow per square foot

Location	Basecase [hours]	Proposed [hours]	Percent reduction of cooling hours
CZ03: Oakland	2984	1850	38%
CZ06: Torrance	3128	2627	16%
CZ12: Sacramento	2951	2041	31%

Figure 23. Integrated Economizer – Multiple Zone DX: Hours in cooling

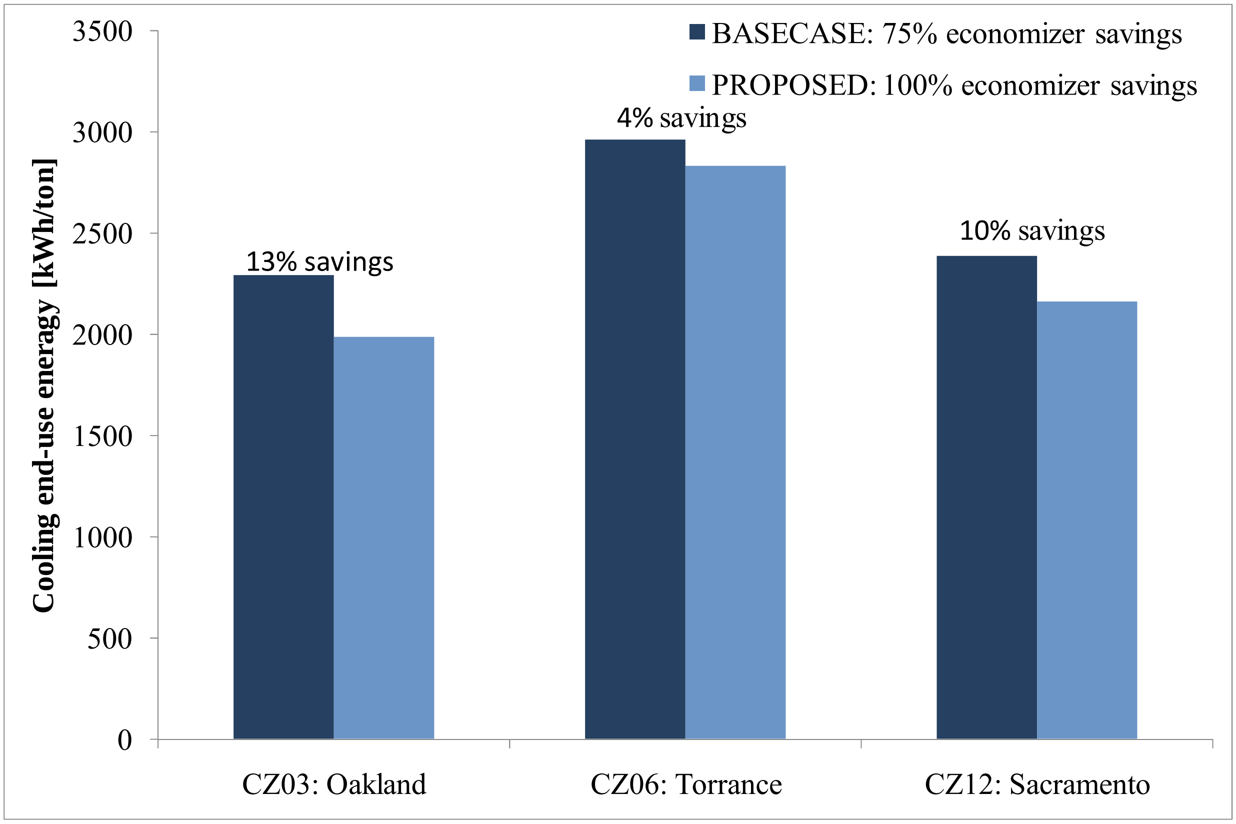


Figure 24. Integrated Economizer – Multiple Zone DX: Cooling end-use energy for the base and proposed cases

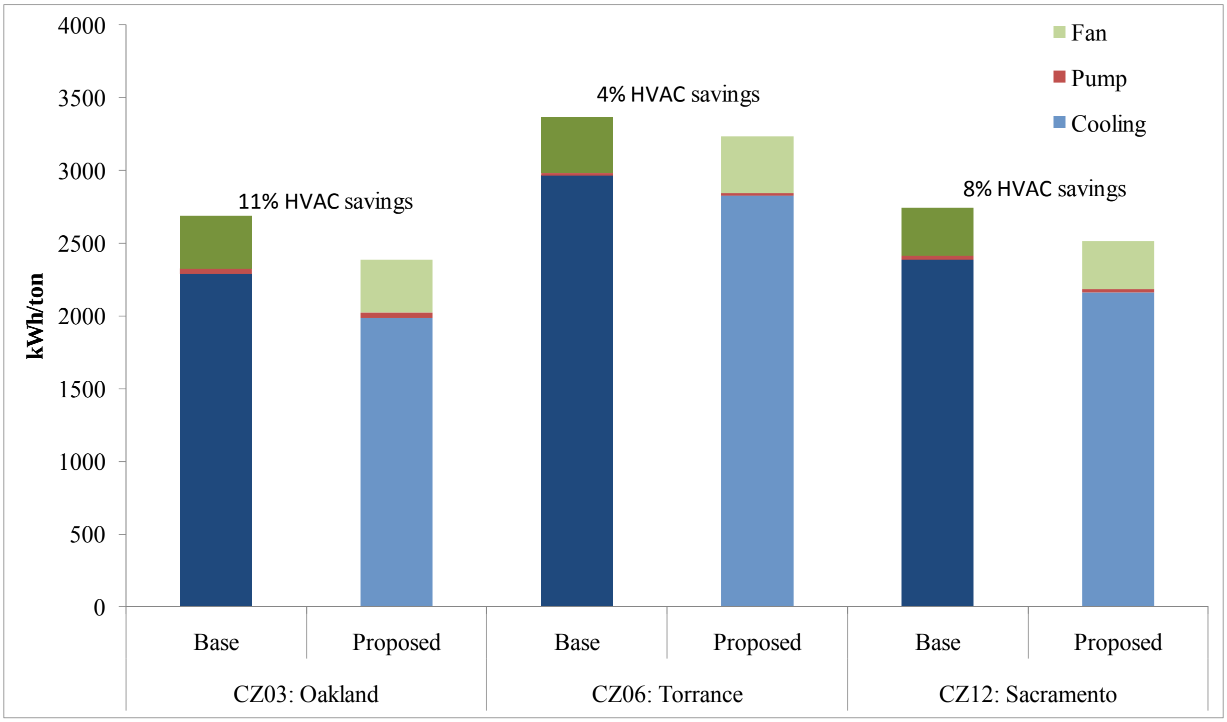


Figure 25. Integrated Economizer – Multiple Zone DX: HVAC end-use energy for the base and proposed cases

3.4.1.3 Incremental Installed Cost

Incremental cost data was provided by the AHRI Unitary Large Equipment (ULE) Committee based on a poll of the full AHRI ULE section and averaging the reported costs. See Appendix 7.8 for the email exchange with Dick Lord, the AHRI representative, regarding this cost information.

The incremental cost for this measure is the cost of variable capacity compressor over a two-stage compressor. Cost data was provided for the incremental cost for a compressor with two stages and a two-speed fan over a single stage compressor and a single speed fan as well as the incremental cost for a variable compressor and a two-speed fan over a single stage compressor and a single speed fan. Therefore, the incremental cost for this measure is the difference between these two, which are shown below in **Error! Reference source not found.**. This incremental cost includes additional installation, start-up and markup costs but does not include additional maintenance costs.

Incremental Cost	6 ton	7 ton	8 ton	10 ton
Variable capacity compressor, two-speed fan	$1,190.33	$1,306.00	$1,484.33	$1,663.33
Two stage compression, two-speed fan	$496.00	$556.00	$655.67	$722.00
Variable capacity compressor only	**$694.33**	**$750.00**	**$828.66**	**$941.33**

Figure 26. Integrated Economizer – Multiple Zone DX: Incremental cost data

The AHRI ULE Committee incremental cost data represents current, national costs if this measure became required by ASHRAE 90.1. Taylor Engineering will be proposing the same measures described herein to 90.1 and there is a reasonable chance that it will be adopted by 90.1, particularly if it is adopted by Title 24. It is not clear if the incremental cost would increase if this was only required in California as opposed to nationally. It is clear, however, that the incremental cost in 2015 will be lower than the current incremental cost as advancements continue to be made in the areas of variable capacity and variable speed compressors. It is reasonable to assume therefore that the incremental cost in 2015 will be lower than the current incremental cost, even if this is only adopted in California.

There is also a potential incremental cost reduction for this measure due to electrical system downsizing. Variable speed compressors, for example, can eliminate the large in-rush currents associated with large compressor motor drives. To be conservative this cost savings is not included in the analysis.

3.4.1.4 Maintenance Cost

According to service contractors incremental maintenance for this measure should be negative, due to reduced economizer and compressor cycling. To be conservative, incremental maintenance is assumed to be zero for this analysis.

3.4.1.5 Lifecycle Cost Results

As shown in **Error! Reference source not found.**, the measure is highly cost effective.

6 ton unit	**CZ03**	**CZ06**	**CZ12**
Incremental Installed Cost	$694	$694	$694
Incremental Annual Maint.	$0	$0	$0
Total Incremental Cost	$694	$694	$694
NPV of Energy Savings	$3,053	$1,329	$2,312
Lifecycle cost savings	$2,358	$635	$1,618

7 ton unit	**CZ03**	**CZ06**	**CZ12**
Incremental Installed Cost	$750	$750	$750
Incremental Annual Maint.	$0	$0	$0
Total Incremental Cost	$750	$750	$750
NPV of Energy Savings	$3,561	$1,550	$2,698
Lifecycle cost savings	$2,811	$800	$1,948

8 ton unit	**CZ03**	**CZ06**	**CZ12**
Incremental Installed Cost	$829	$829	$829
Incremental Annual Maint.	$0	$0	$0
Total Incremental Cost	$829	$829	$829

NPV of Energy Savings	$4,070	$1,772	$3,083
Lifecycle cost savings	$3,242	$943	$2,255

10 ton unit	**CZ03**	**CZ06**	**CZ12**
Incremental Installed Cost	$941	$941	$941
Incremental Annual Maint.	$0	$0	$0
Total Incremental Cost	$941	$941	$941
NPV of Energy Savings	$5,088	$2,215	$3,854
Lifecycle cost savings	$4,146	$1,273	$2,913

Figure 27. Integrated Economizer – Multiple Zone DX: Lifecycle Cost Results

3.5 Combined Measures: Fan Control and Integrated Economizer – Single Zone DX

This section describes the analysis that was run to show that the combination of the fan control and integrated economizer measures for a single zone DX unit is cost effective.

3.5.1.1 Energy Analysis

The same eQuest model was used in this analysis as was used in Section 3.1 Fan Control – Single Zone DX. Modifications to the model and analysis are described in this section.

Run	**Economizer**	**Fan Speed**
Base	Outside air temperature with 60°F drybulb limit	Constant
Proposed	Fully integrated dual temperature	Variable

Figure 28. Combined Measures: Fan Control and Integrated Economizer – Single Zone DX: Base and Proposed Cases

3.5.1.1.1 Building Envelope

See Section 3.1.1.1.1.

3.5.1.1.2 Climate

The simulation was run in six climates:

4. CZ03: Oakland
5. CZ04: San Jose
6. CZ06: Torrance (coastal Los Angeles)
7. CZ09: Pasadena (inland Los Angeles)
8. CZ10: Riverside
9. CZ12: Sacramento

The weather file that was used in this simulation for the Los Angeles run came from the California Energy Commission (CEC) and was developed for Title 24 – 2013. The simulation year used for all models was 2009, per Title 24 CASE requirements.

3.5.1.1.3 Internal Loads

4. Lighting power density: 1.0 W/ft^2
5. Equipment power density: 1.5 W/ft^2
6. Occupancy density: 200 ft^2/person

3.5.1.1.4 Schedules

See Section 3.1.1.1.4.

3.5.1.1.5 Temperatures

See Section 3.1.1.1.5.

3.5.1.1.6 System Properties

8. SYSTEM-TYPE: PVVT. The model had a Packaged Single Zone with DX cooling and gas furnace for heating per zone. Then this was changed to PVVT in a parametric run that became the basecase.
9. MIN-FLOW-RATIO: 1.0 to model a constant volume unit.
10. RETURN-AIR-PATH: Duct
11. Supply fan:
    a. SUPPLY-STATIC: 2.5”
    b. SUPPLY-EFF: 53%
    c. SUPPLY-MECH-EFF: 65%
    d. FAN-CONTROL: CONSTANT-VOLUME
    e. NIGHT-CYCLE-CTRL: CYCLE-ON-FIRST. The fans cycle on for the hour if the temperature in the zone goes out of range for heating and cooling setback temperatures during unoccupied hours.
12. Cooling:
    a. DX cooling
    b. MIN-SUPPLY-T: 55°F
    c. COOLING-EIR: 0.3496. Converted from the minimum efficiency of 9.7 SEER for a Unitary AC that is less than 65,000 Btu/h from Title 24 and ASHRAE Standard 90.1.
    d. CONDENSER-TYPE: AIR-COOLED.
    e. MIN-UNLOADING-RATIO: 1.0. Compressors only cycle, they do not modulate.
    f. MIN-HGB-RATIO: 1.0. No hot gas bypass.
    g. COOL-CTRL-RANGE: 0°F per Taylor Engineering’s Energy Modeling Standards
13. Heating:

   a. HEAT-SOURCE: Gas furnace.
   b. FURNACE HIR: 1.2407 Btu/Btu. Converted from 78% AFUE.
14. Outside Air:
   a. MIN-AIR-SCH: Set equal to a created a schedule of type Frac/Design that has the value 0 during unoccupied hours and -999 during occupied hours. During occupied hours the system will default to the normal ventilation values. During unoccupied hours when the system cycles on to reach the setback temperatures, there is no outside air ventilation.
   b. OA-CONTROL: OA-TEMP.
   c. MAX-OA-FRACTION: 1.0
   d. DRYBULB-LIMIT: 60°F
   e. ECONO-LOCKOUT: NO. The compressor(s) can operate simultaneously with the economizer to meet the cooling load. The economizer only shuts off when the outside air is warmer than 60°F.

3.5.1.1.7 Zone Properties

6. OA-FLOW: 0.15 CFM per square foot was used to calculate minimum outside air:
   a. South and North: 182 CFM
   b. East and West: 108 CFM
   c. Core: 231 CFM
7. DESIGN-COOL-T: 75°F
8. DESIGN-HEAT-T: 70°F
9. TYPE-ZONE
   a. Perimeter and Core Zones: CONDITIONED
   b. Roof Zone: UNCONDITIONED
10. THROTTLING RANGE: 0.5°F per Taylor Engineering's Energy Modeling Standards

3.5.1.1.8 System Sizing for Full Speed Model

The auto-sizing feature in DOE-2 is not reliable. Therefore, the model was run iteratively: first it was run to determine the peak loads. Then the cooling system, fan flow, and heating system were manually input into the model as 115% of the actual peak cooling load to account for the availability of discrete equipment sizes.

3.5.1.1.9 Proposed Case System Properties

2. SYSTEM-TYPE: PVVT. The model had a Packaged Single Zone with DX cooling and gas furnace for heating per zone. Then this was changed to PVVT in a parametric run that became the base case.
3. MIN-FLOW-RATIO: 0.5 to allow the fans to throttle down to 50% speed, but not below.
4. Supply fan:
   a. FAN-CONTROL: FAN-EIR-FPLR. Fans are variable and ride the zero fixed static fan curve.

    b. FAN-EIR-FPLR: Zero fixed static fan curve
5. Outside Air:
    a. MIN-AIR-SCH: Set equal to a created a schedule of type Frac/Design that has the value 0 during unoccupied hours and -999 during occupied hours. During occupied hours the system will default to the normal ventilation values. During unoccupied hours when the system cycles on to reach the setback temperatures, there is no outside air ventilation.
    b. OA-CONTROL: DUAL-TEMP.
    c. MAX-OA-FRACTION: 1.0
    d. DRYBULB-LIMIT: n/a (blank)
    e. ECONO-LOCKOUT: NO. This models a fully integrated economizer. If the economizer cannot meet the entire cooling load, it can still remain on to meet part of the load. The compressor(s) can operate simultaneously with the economizer to meet the remaining cooling load. The economizer is only locked out when the outside air is warmer than the return air.

3.5.1.1.10 DOE-2 Version

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

3.5.1.2 Energy Results

The cooling load dominated by hours below 50% of the design cooling load, as shown in **Error! Reference source not found.**. The vast majority of the fan hours in the proposed case fall in the 50-60% speed range, since the fan speed was limited to a 50% minimum speed, as shown in **Error! Reference source not found.**. Thus, there is a large opportunity for low fan speed and economizer hours.

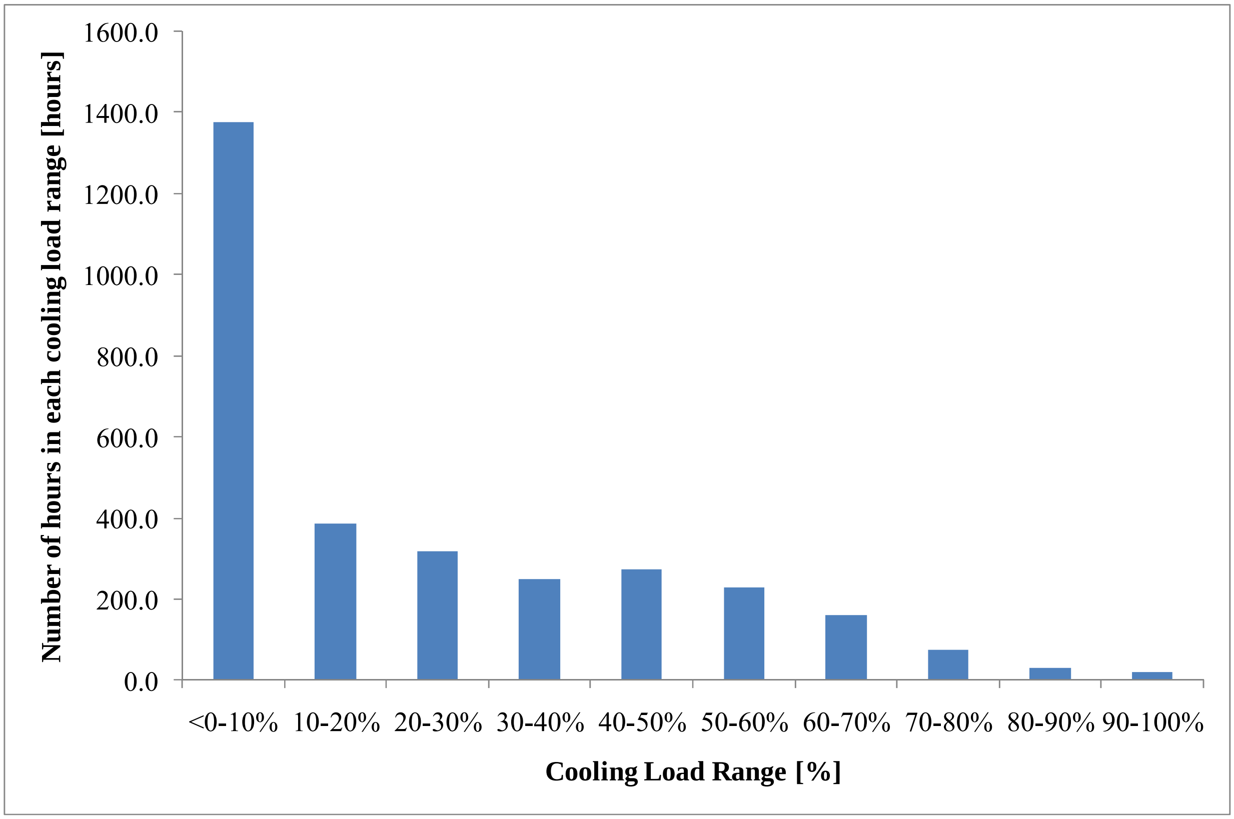


Figure 29. Combined Measures: Fan Control and Integrated Economizer – Single Zone DX: Number of hours in each cooling load range for the south zone

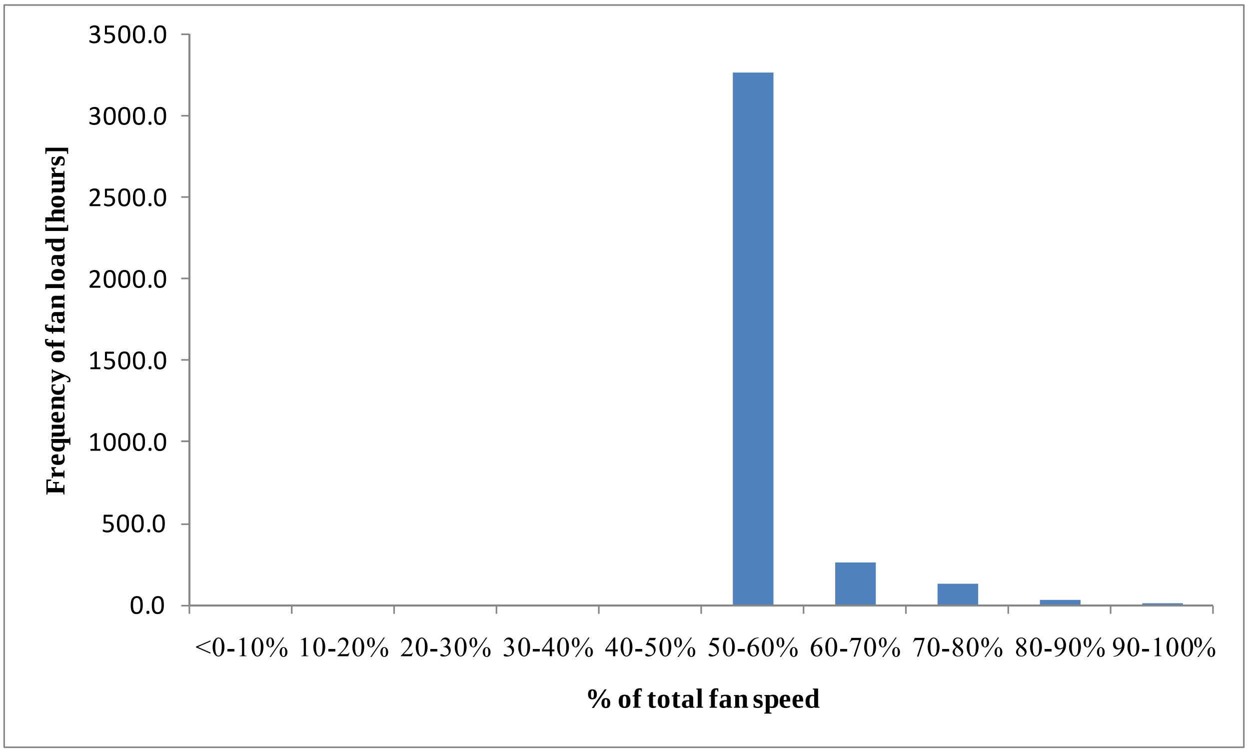


Figure 30. Combined Measures: Fan Control and Integrated Economizer – Single Zone DX: Number of hours in each fan speed range for the south zone

The figure below shows the percent of design fan speed versus the percent of design cooling load. A linear trendline is shown since there is a linear relationship between load and fan speed above 50%, which is the minimum allowed fan speed. **Error! Reference source not found.** demonstrates that there is a large opportunity for fan speed to be reduced to 50%.

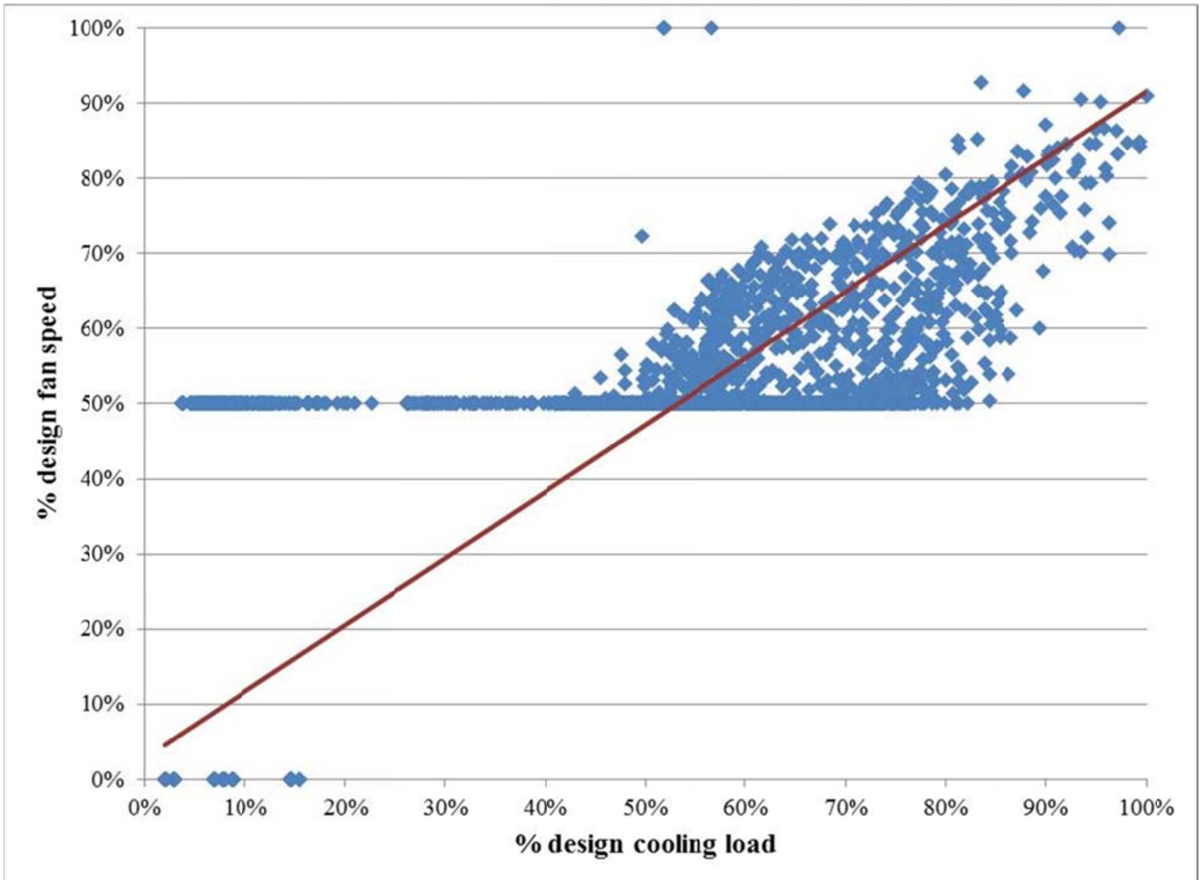


Figure 31. Combined Measures: Fan Control and Integrated Economizer – Single Zone DX: % design fan speed versus % design cooling load for a representative zone

The cooling and HVAC energy savings are large when the fan control and integrated economizer measures for a single zone DX unit are combined, as shown in **Error! Reference source not found.** and **Error! Reference source not found.** below. Most of the HVAC energy savings come from the reduced fan energy, as shown in **Error! Reference source not found.**.

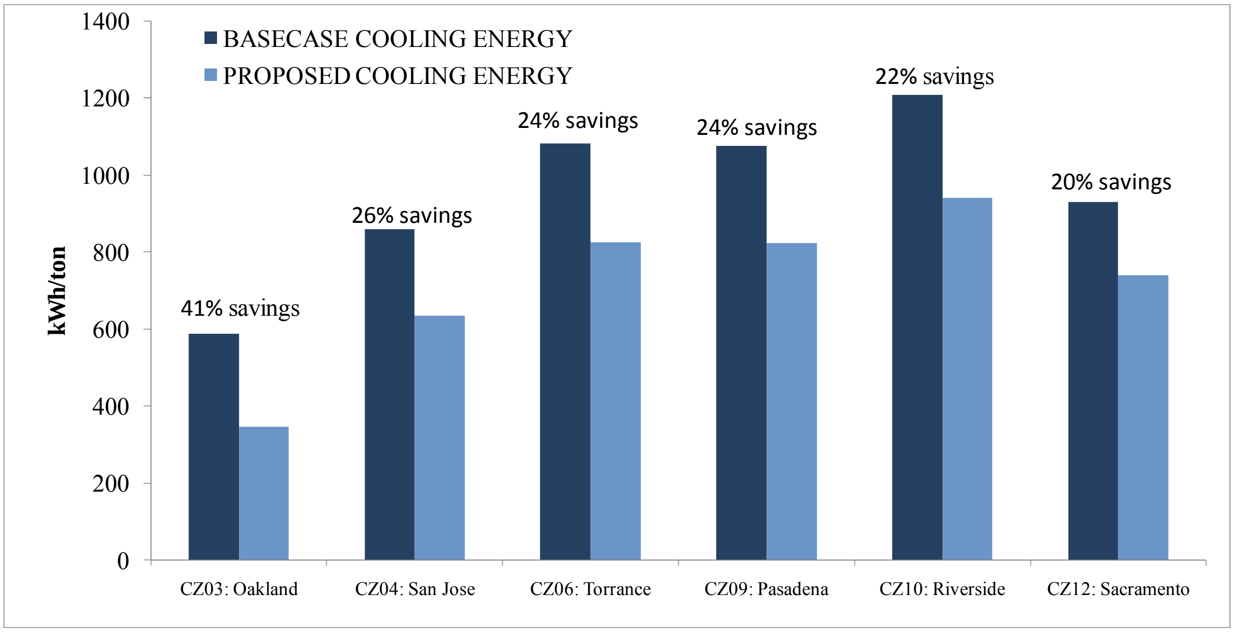


Figure 32. Combined Measures: Fan Control and Integrated Economizer – Single Zone DX: End-use cooling energy consumption

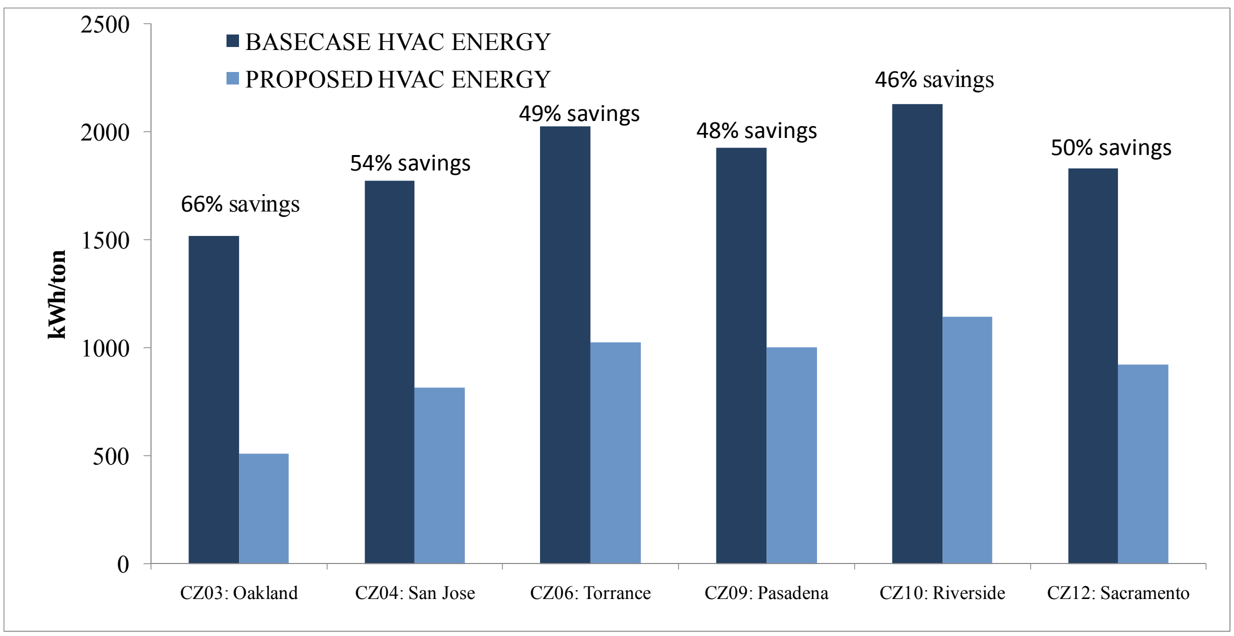


Figure 33. Combined Measures: Fan Control and Integrated Economizer – Single Zone DX: HVAC energy consumption

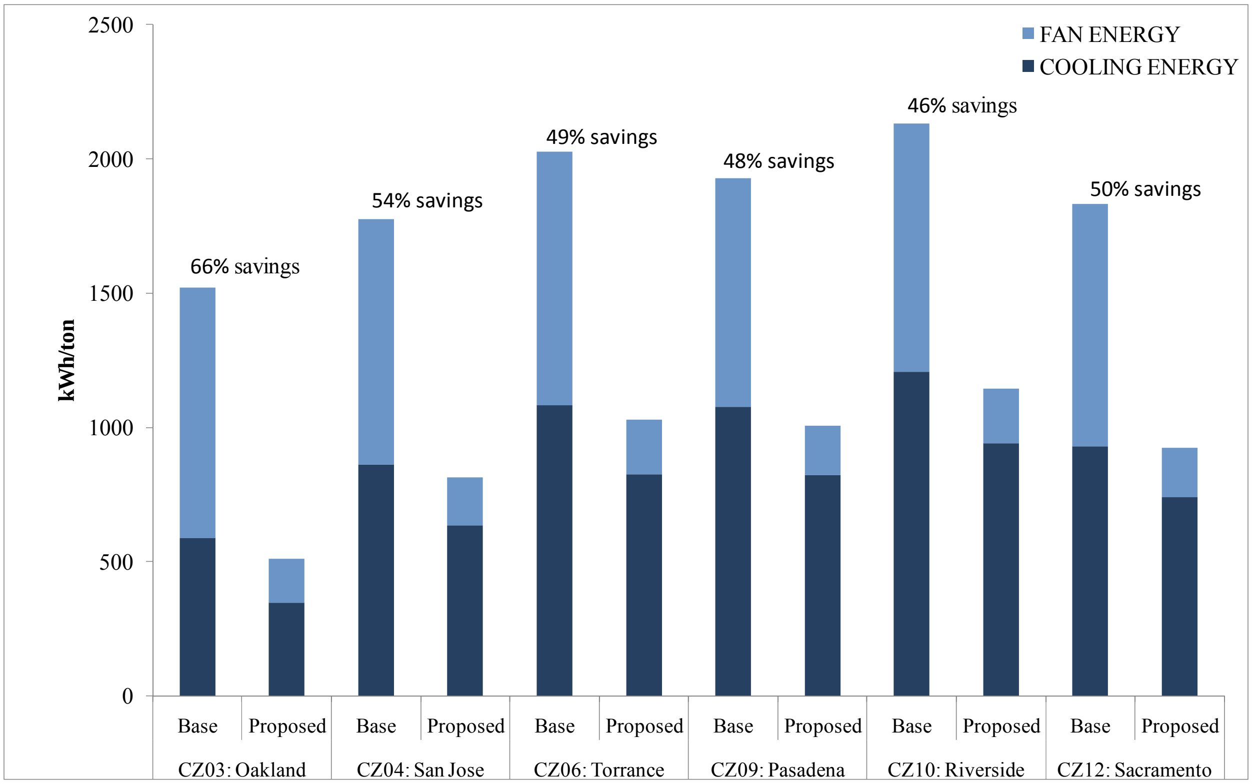


Figure 34. Combined Measures: Fan Control and Integrated Economizer – Single Zone DX: HVAC energy consumption by end-use

The Air-Conditioning, Heating, and Refrigeration Institute (AHRI) conducted a survey of packaged rooftop units sold in California in 2010. According to this data, there are approximately 40,000 tons of packaged rooftop units between 65,000 Btu/h and 110,000 Btu/h in California (see Appendix 7.9). Based on the average annual kWh savings per ton of 965 kWh/ton for the 6 climate zones that were modeled, the annual statewide electricity savings would be 38 million kWh in the first year. This is a very conservative estimate of statewide savings since this is only for packaged rooftop units. It does not include any savings from CHW systems. It also does not include energy savings from improved compressor part load performance or savings from integrated economizers on systems over 110,000 Btu/h.

3.5.1.3 Incremental Installed Cost

Incremental cost data was provided by the AHRI Unitary Large Equipment (ULE) Committee based on a poll of the full AHRI ULE section and averaging the reported costs. The incremental cost for a variable capacity compressor and a variable fan over a single stage compressor and a single speed fan is $2,133 for a 6 ton unit. This incremental cost includes additional installation, start-up and commissioning costs but does not include additional maintenance costs.

The AHRI ULE Committee incremental cost data represents current, national costs if this measure became required by ASHRAE 90.1. Taylor Engineering will be proposing the same measures described herein to 90.1 and there is a reasonable chance that it will be adopted by 90.1, particularly if it is adopted by Title 24. It is not clear if the incremental cost would increase if this was only required in California as opposed to nationally. It is clear, however, that the incremental cost in 2015 will be lower than the current incremental cost as advancements continue to be made in the areas of variable speed drives, EC motors, and variable capacity and variable speed compressors. It is reasonable to assume therefore that the incremental cost in 2015 will be lower than the current incremental cost, even if this is only adopted in California.

3.5.1.4 Maintenance Cost

Incremental maintenance cost is conservatively estimated to be 1 hour per year at a labor rate of $100/hr. As noted above, incremental maintenance may well be negative due to reduced wear and tear on compressors and dampers due to reduced cycling. For additional maintenance costs to negate the savings in the climate zone with the lowest lifecycle cost savings, 8 additional hours of maintenance a year would be required, which is unrealistically high.

3.5.1.5 Lifecycle Cost Results

As shown in **Error! Reference source not found.**, applying the fan control and integrated economizer measures together for a single zone DX unit is extremely cost effective.

	CZ03	**CZ04**	**CZ06**	**CZ09**	**CZ10**	**CZ12**
Incremental Installed Cost	$2,133	$2,133	$2,133	$2,133	$2,133	$2,133
Incremental Annual Maint.	$100	$100	$100	$100	$100	$100
NPV of Annual Maint.	$1,190	$1,190	$1,190	$1,190	$1,190	$1,190
Total Incremental Cost	$3,323	$3,323	$3,323	$3,323	$3,323	$3,323
NPV of Energy Savings	$13,520	$12,370	$12,050	$11,500	$12,020	$11,610
Lifecycle cost savings	$10,197	$9,047	$8,727	$8,177	$8,697	$8,287

Figure 35. Combined Measures: Fan Control and Integrated Economizer – Single Zone DX: Lifecycle Cost Results for 6 Ton Unit

4. Stakeholder Input

4.1 Notes from Stakeholder Meeting #2 on December 9, 2010

Integrated economizer

- Add "Digital Scroll Compressor" or "Variable Capacity Scroll Compressor" to Slide 29
- Concern over use of "digital scroll compressors" since only one manufacturer has patent to digital scroll compressor at this time
- Language is 20% min capacity - can use other methods to achieve this requirement such as variable speed or staged capacity compressors
- Industry is headed to digital scrolls

4.2 ASHRAE 90.1 Mechanical Subcommittee - Packaged Units Working Group

A Packaged Units Working Group was established at the summer 2010 ASHRAE meeting within the ASHRAE 90.1 Mechanical Subcommittee. Their feedback was appreciated and incorporated into these analyses. Similar measures will be proposed for the next revision of ASHRAE 90.1.

4.2.1 Members of Packaged Units Working Group

Members of the committee came from the following groups:

- Taylor Engineering
- Pacific Northwest National Lab (PNNL)
- Trane
- Carrier
- Southern California Edison (SCE)
- Ring & Duchateau
- Air-Conditioning, Heating, and Refrigeration Institute (AHRI)
- Arizona State University

4.2.2 Packaged Units Working Group Meetings and Memo Timeline

The following is a list of dates of meetings and memos:

- Formed in June 2010
- July 14, 2010 Conference call
- September 3, 2010 – Memo #1: Fan Control Analysis distributed to working group members
- September 7, 2010 – Conference call
- October 22, 2010 – Memo #2: Fan Control Analysis distributed to working group members
- October 22, 2010 – Memo #3: Integrated Economizer distributed to working group members

- November 3, 2010 – Conference call
- November 22, 2010 – Memo #2: Fan Control Analysis – Revised and Memo #3: Integrated Economizer Analysis – Revised sent to the working group, incorporating comments from the November 3rd conference call
- January 30, 2010 – Presented at ASHRAE Conference in Las Vegas, NV to the 90.1 Mechanical Subcommittee

4.2.3 Packaged Units Working Group Meeting Minutes

See Appendix 0 "

Meeting Minutes for ASHRAE 90.1 Mechanical Subcommittee Packaged Working Group Conference Calls".

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**.

Multiple Zone System: an air distribution system that supplies air to more than one thermal zone each of which has one or more devices (such as dampers, cooling coils, and heating coils) that regulate airflow, cooling, or heating capacity to the zone**.**

Single Zone System: an air distribution system that supplies air to one thermal zone

5.2 SECTION 144 – PRESCRIPTIVE REQUIREMENTS FOR SPACE CONDITIONING SYSTEMS

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

1. ***Single zone systems*** **and constant volume ~~fan~~ systems.** The total fan power index at design conditions of each fan system with total horsepower over 25 hp shall not exceed 0.8 watts per cfm of supply air.
2. **Multiple zone v*ariable air volume (VAV)* systems.**

   A. The total fan power index at design conditions of each fan system with total horsepower over 25 hp shall not exceed 1.25 watts per cfm of supply air; and

   ~~B. Individual VAV fans with motors 10 horsepower or larger shall meet one of the following:~~

   ~~i. The fan motor shall be driven by a mechanical or electrical variable speed drive.~~

   ~~ii. The fan shall be a vane-axial fan with variable pitch blades.~~

   ~~iii. For prescriptive compliance, the fan motor shall include controls that limit the fan motor demand to no more than 30 percent of the total design wattage at 50 percent of design air volume when static pressure set point equals 1/3 of the total design static pressure, based on certified manufacturer's test data.~~

B. Static Pressure Sensor Location. Static pressure sensors used to control variable air volume fans shall be placed in a position such that the controller set point is no greater than one-third the total design fan static pressure, except for systems with zone reset control complying with Section144(c)2D. If this results in the sensor being located downstream of major duct splits, multiple sensors shall be installed in each major branch with fan capacity controlled to satisfy the sensor furthest below its setpoint.

C. Set Point Reset. For systems with direct digital control of individual zone boxes reporting to the central control panel, static pressure set point shall be reset based on the zone requiring the most pressure; i.e., the set point is reset lower until one zone damper is nearly wide open.

144 (e) Economizers.

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

B. Capable of providing partial cooling even when additional mechanical cooling is required to meet the remainder of the cooling load. Effective January 1, 2015, direct expansion systems with a cooling capacity ≥ 65,000 Btu/hr[a] shall be capable of staging or modulating capacity in increments of no more than 20% of total cooling capacity. Controls shall not false load the mechanical cooling system by limiting or disabling the economizer or any other means, such as hot gas bypass, except at the lowest stage of cooling capacity.

[a]See Tables 112-A and 112-B for rating standard and conditions

~~**144 (l) Variable air volume control for single zone systems.** Effective January 1, 2012 all unitary air conditioning equipment and air-handling units with mechanical cooling capacity at ARI conditions greater than or equal to 110,000 Btu/hr that serve single zones shall be designed for variable supply air volume with their supply fans controlled by two-speed motors, variable speed drives, or equipment that has been demonstrated to the Executive Director to use no more energy. The supply fan controls shall modulate down to a minimum of 2/3 of the full fan speed or lower at low cooling demand.~~

144 (l) Fan Control. Each *multiple zone system* and *single zone system* listed in Table 144-D shall be designed to vary the airflow rate as a function of actual load. *Single zone systems* 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. *Multiple zone systems* shall include controls that limit the fan motor demand to no more than 30 percent of the total design wattage at 50 percent of design air volume when static pressure set point equals 1/3 of the total design static pressure.

Table 144-D – Effective Date for Variable Airflow Control of Fan Systems

Cooling System Type	**Fan Motor Size**	**Cooling Capacity[a]**	**Effective Date**
Direct Expansion	any	≥ 110,000 Btu/hr	January 1, 2012
Direct Expansion	any	≥65,000 Btu/hr and <110,000 Btu/hr	January 1, 2015
Chilled water	≥1/4 hp	any	January 1, 2012

Evaporative	≥1/4 hp	any	January 1, 2012

[a]See Tables 112-A and 112-B for rating standard and conditions

EXCEPTION 1 to Section 144(l): Systems that supply 100% outdoor air and are required to be constant volume in order to maintain minimum ventilation or makeup air rates.

5.3 Appendix NA7 – Acceptance Requirements for Nonresidential Buildings

NA7.5 Mechanical Systems Acceptance Tests

NA7.5.4 Air Economizer Controls

NA7.5.4.1 Construction Inspection

Prior to Functional Testing, verify and document the following:

- Economizer lockout setpoint complies with Table 144-C of §144(e)3.
- Economizer lockout control sensor is located to prevent false readings.
- System is designed to provide up to 100 percent outside air without over-pressurizing the building.
- For systems with DDC controls lockout sensor(s) are either factory calibrated or field calibrated.
- For systems with non-DDC controls, manufacturer's startup and testing procedures have been applied
- For DX systems 65,000 Btu/hr and less, thermostats (e.g. two stage or electronic) and control system has capacity to modulate compressor or cycle compressor off during periods where economizer cooling can partially meet the cooling load as per §144(e)2.B i.
- For DX: equipment submittal specifies compressor capacity steps and/or compressor capacity modulation complying with the stages or modulation required in §144(e)2.B ii

NA7.5.4.2 Functional Testing

Step 1: Disable demand control ventilation systems (if applicable)

Step 2: Enable the economizer and simulate a cooling demand large enough to drive the economizer fully open. Verify and document the following:

- Economizer damper is 100 percent open and return air damper is 100 percent closed.

- For systems that meet the criteria of §144(e)2.B ii, verify that the economizer remains 100 percent open when the cooling demand can no longer be met by the economizer alone.
- For systems that meet the criteria of §144(e)2.B i, verify that the economizer is 100 percent open part of the time and the compressor cycles on and off when the cooling demand can no longer be met by the economizer alone.
- All applicable fans and dampers operate as intended to maintain building pressure.
- The unit heating is disabled.

Step 3: Disable the economizer and simulate a cooling demand. Verify and document the following:

- Economizer damper closes to its minimum position.
- All applicable fans and dampers operate as intended to maintain building pressure.
- The unit heating is disabled

Step 4: Simulate a heating demand and set the economizer so that it is capable of operating (i.e. actual outdoor air conditions are below lockout setpoint). Verify the following:

- The economizer is at minimum position

Step 5: Restore demand control ventilation systems (if applicable) and remove all system overrides initiated during the test.

NA7.5.6 Supply Fan Variable Flow Controls

NA7.5.6.1 Construction Inspection

Prior to Functional Testing, verify and document the following:

- Supply fan includes device(s) for modulating airflow, such as variable speed drive or electrically commutated motor.
- For multiple zone systems:
  - Discharge static pressure sensors are either factory calibrated or field-calibrated.
  - The static pressure location, setpoint, and reset control meets the requirements of §144(c)2~~C~~B and §144(c)2~~D~~C.

NA7.5.6.2 Functional Testing

Step 1: Simulate demand for design airflow. Verify and document the following:

- Supply fan controls modulate to increase capacity.
- For multiple zone systems: Supply fan maintains discharge static pressure within +/-10 percent of the current operating set point.
- Supply fan controls stabilize within a 5 minute period.

Step 2: Simulate demand for minimum airflow. Verify and document the following:

- Supply fan controls modulate to decrease capacity.
- Current operating setpoint has decreased (for systems with DDC to the zone level).

- For multiple zone systems: Supply fan maintains discharge static pressure within +/-10 percent of the current operating setpoint.
- Supply fan controls stabilize within a 5 minute period.

Step 3: Restore system to correct operating conditions

5.4 Nonresidential ACM Manual

Table N2-16 – System #1 and System #2 Descriptions

System Description:	Packaged Single Zone with Gas Furnace/Electric Air Conditioning (#1) or Heat Pump (#2)
Supply Fan Power:	See Section 2.5.3.5 for design power. Fan power ratio at part load for variable speed = speed ratio ^3 (e.g. 12.5% of design power at 50% speed).
Supply Fan Control:	Constant volume < ~~10~~6 tons ~~proposed~~ calculated cooling capacity Variable ~~Volume with 2~~ speed motor > ~~10~~ 6 tons ~~proposed~~ calculated cooling capacity
Min Supply Temp:	50 $\leq$ T $\leq$ 60 DEFAULT: 55
Cooling System:	Direct expansion (DX)
Cooling Efficiency:	Minimum SEER or EER based on equipment type and output capacity of proposed unit(s). Adjusted EER is calculated to account for supply fan energy.
Maximum Supply Temp:	85 $\leq$ T $\leq$ 110 DEFAULT: 100
Heating System:	Gas furnace (#1) or heat pump (#2)
Heating Efficiency:	Minimum AFUE, Thermal Efficiency, COP or HSPF based on equipment type and output capacity of proposed unit(s).
Economizer:	Integrated drybulb economizer, when mechanical cooling output capacity of the proposed design as modeled in the compliance run by the compliance software is over 65,000 Btu/hr and fan system volumetric capacity of the proposed design as modeled in the compliance run by the compliance software is over 2500 cfm Partially integrated drybulb economizer. For DX systems with a cooling capacity of 65,000 Btu/hr or less. When economizer can meet load it provides entire load. When economizer can partially meet load, compressor cycles on and off, when compressor on economizer is closed and when compressor off economizer sis fully open. (Note: Not currently modeled by DOE-2.1E or DOE-2.2)
Ducts:	For ducts installed in unconditioned buffer spaces or outdoors as specified in §144(k), the duct system efficiency shall be as described in Section 2.5.3.18.
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 N2-1 – System #5 Description

System Description:	Four-Pipe Fan Coil With Central Plant
Supply Fan Power:	See Section 2.5.3.5 for design power. Fan power ratio at part load for variable speed = speed ratio ^3 (e.g. 12.5% of design power at 50% speed).
Supply Fan Control:	Constant speed < ¼ bhp calculated fan power Variable speed fan >= ¼ bhp calculated fan power.
Minimum Supply Temp:	50 $\leq$ T $\leq$ 60 DEFAULT: 55
Cooling System:	Chilled water
Chilled Water Pumping System	Variable flow (2-way valves) with a VSD on the pump if three or more fan coils. Constant volume flow with water temperature reset control if less than three fan coils. Reset supply pressure by demand if proposed system has DDC controls.
Cooling Efficiency:	Minimum efficiency based on the proposed output capacity of specific equipment unit(s)
Maximum Supply Temp:	90 $\leq$ T $\leq$ 110 DEFAULT: 100
Heating System:	Gas boiler
Hot Water Pumping System	Variable flow (2-way valves) riding the pump curve if three or more fan coils. Constant volume flow with water temperature reset control if less than three fan coils. Reset supply pressure by demand if proposed system has DDC controls.
Heating Efficiency:	Minimum efficiency based on the proposed output capacity of specific equipment unit(s)
Economizer:	Integrated dry bulb economizer, when mechanical cooling output capacity of the proposed design as modeled in the compliance run by the compliance software is over 75,000 Btu/hr and fan system volumetric capacity of the proposed design as modeled in the compliance run by the compliance software is over 2500 cfm
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")

6. Bibliography and Other Research

John Dieckmann and James Broderick, 2011, “AC Capacity Modulation”, ASHRAE Journal, February 2011

Dieckmann, J., K. McKenney, and J.Broderick. 2010. “Variable Frequency Drives, Part 2: VFDs for Blowers.” ASHRAE Journal 52(5): 58-62.

Copeland Corporation, “Digital Capacity Control for Refrigeration Scroll Compressor.” Application Engineering Bulletin. November 2003

7. Appendices

7.1 Non-Residential Construction Forecast details

7.1.1 Summary

The Non-Residential construction forecast dataset is data that is published by the California Energy Commission's (CEC) demand forecast office. This demand forecast office is charged with calculating the required electricity and natural gas supply centers that need to be built in order to meet the new construction utility loads. Data is sourced from Dodge construction database, the demand forecast office future generation facility planning data, and building permit office data.

All CASE reports should use the statewide construction forecast for 2014. The TDV savings analysis is calculated on a 15 or 30 year net present value, so it is correct to use the 2014 construction forecast as the basis for CASE savings.

7.1.2 Additional Details

The demand generation office publishes this dataset and categorizes the data by demand forecast climate zones (FCZ) as well as building type (based on NAICS codes). The 16 climate zones are organized by the generation facility locations throughout California, and differ from the Title 24 building climate zones (BCZ). HMG has reorganized the demand forecast office data using 2000 Census data (population weighted by zip code) and mapped FCZ and BCZ to a given zip code. The construction forecast data is provided to CASE authors in BCZ in order to calculate Title 24 statewide energy savings impacts. Though the individual climate zone categories differ between the demand forecast published by the CEC and the construction forecast, the total construction estimates are consistent; in other words, HMG has not added to or subtracted from total construction area.

The demand forecast office provides two (2) independent data sets: total construction and additional construction. Total construction is the sum of all existing floor space in a given category (Small office, large office, restaurant, etc.). Additional construction is floor space area constructed in a given year (new construction); this data is derived from the sources mentioned above (Dodge, Demand forecast office, building permits).

Additional construction is an independent dataset from total construction. The difference between two consecutive years of total construction is not necessarily the additional construction for the year because this difference does not take into consideration floor space that was renovated, or repurposed.

In order to further specify the construction forecast for the purpose of statewide energy savings calculation for Title 24 compliance, HMG has provided CASE authors with the ability to aggregate across multiple building types. This tool is useful for measures that apply to a portion of various building types' floor space (e.g. skylight requirements might apply to 20% of offices, 50% of warehouses and 25% of college floor space).

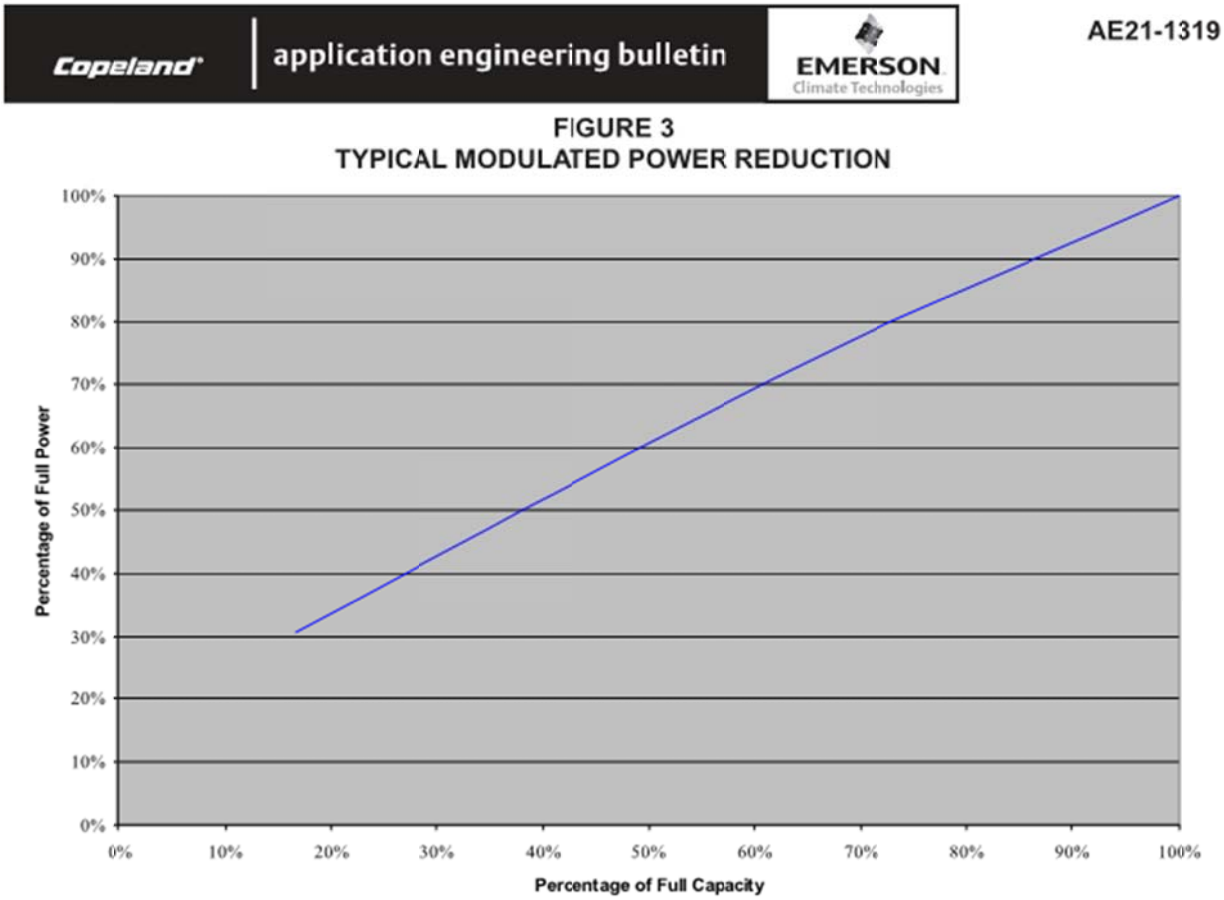
The main purpose of the CEC demand forecast is to estimate electricity and natural gas needs in 2022 (or 10-12 years in the future), and this dataset is much less concerned about the inaccuracy at 12 or 24 month timeframe.

It is appropriate to use the CEC demand forecast construction data as an estimate of future years construction (over the life of the measure). The CEC non-residential construction forecast is the best publicly available data to estimate statewide energy savings.

7.1.3 Citation

"NonRes Construction Forecast by BCZ v7"; Developed by Heschong Mahone Group with data sourced August, 2010 from Abrishami, Moshen at the California Energy Commission (CEC)

7.2 Part Load Efficiency of a Variable Capacity Compressor



Source: Copeland Corporation, "Digital Capacity Control for Refrigeration Scroll Compressor." Application Engineering Bulletin. November 2003

7.3 Case Study in Economizer Cycling for Large Packaged Unit

Taylor Engineering
1080 Marina Village Parkway, Suite 501 ▪ Alameda, CA 94501-1142 ▪ (510) 749-9135 ▪ Fax (510) 749-9136

To:	Engineers
From:	Hwakong Cheng & Reinhard Seidl
Subject:	Economizer and SAT Resets on Packaged AC Units
Date:	April 23, 2007

In performing a trend review for KLA-Milpitas, we observed "normal" operational patterns in the packaged AC units that deviated significantly from the expected operation and that may result in far greater energy consumption than we would otherwise assume. In particular, it appears as if the supply air temperature reset and integrated economizer in DX units may provide significantly less energy savings than would be found with similar controls in chilled water systems, and may not even provide savings relative to controlling to a fixed SAT.

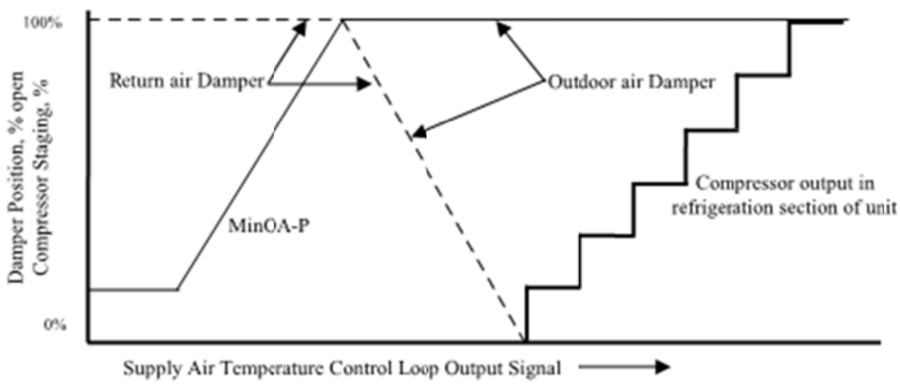
The pattern is as follows: Unit is in economizer, and

- Calls for compressors to come on in integrated mode to help handle the load
- This load is small, but the minimum unit capacity step is 25%
- When compressors come on, they "overshoot" and immediately cause supply temp to drop far below setpoint
- In response, the outside air damper closes (!) to bring supply temp back up, thereby effectively eliminating fully integrated economizer operation

The KLA-Milpitas campus is partially served by large Trane Intellipak units (up to 90 to 120 tons capacities), each of which contain 6 compressor stages. At part loads, however, the internal Trane controllers actually stage multiple compressors together in 25% capacity increments so that there are effectively only 4 stages of operation. The smaller compressors are not utilized individually for operation at low part loads. The large capacity of each effective stage results in excessive short-cycling whenever the load is significantly below the capacity of the next compressor stage. This cycling is observed with particular frequency when the units are operating with the SAT reset, and when operating in integrated economizer mode (cooling load partially met by economizer with DX providing the remainder of the necessary cooling to meet the SAT setpoint). The resulting short cycling of the DX stages frequently involves overshooting the SAT setpoint by 10°F or more (for total SAT swings of up to 20°F) during periods as short as 20 minutes. The overshooting then causes the cooling loop outputs to swing down to 0, thereby forcing the economizer dampers closed (see control diagram in Figure 1). The DX stages operate for the minimum programmed on-time and then wait the minimum off-time before cycling on again, all the while, dragging the economizer damper into the cycle.

April 23, 2007
Page 2

Figure 1 - Supply Air Temperature Control for Packaged AC Unit



The net result is that the economizer damper is shut to the minimum position for a significant fraction of time when it could provide at least partial cooling. The effectively oversized compressor stages may prevent these packaged units from achieving compressor energy savings through SAT reset. Rather, the increased probability of cycling due to reduced coil loads during SAT reset may actually result in an increase in energy consumption. Furthermore, the excessive short-cycling may result in increased wear on the damper actuators and compressors and negatively impact thermal comfort in the associated spaces.

For example, Figure 2 shows trends for one unit over a two day period. Over the course of the first day, the compressor cycles between 0 and 25% and causes similar cycling in the economizer damper position. The associated SAT swings are worse as the SAT is fully reset up to the maximum setpoint and when the unit operates in integrated economizer mode. When the economizer is locked out and the SAT setpoint is only partially reset to 62°F, the SAT swings are about 6°F. The swing increases to about 8 to 11°F when the SAT is fully reset and increases even further to a maximum of over 20°F when the unit operates in partial economizer mode.

April 23, 2007
Page 3

Figure 2 – AC Unit Control – M5 AC-2.2C

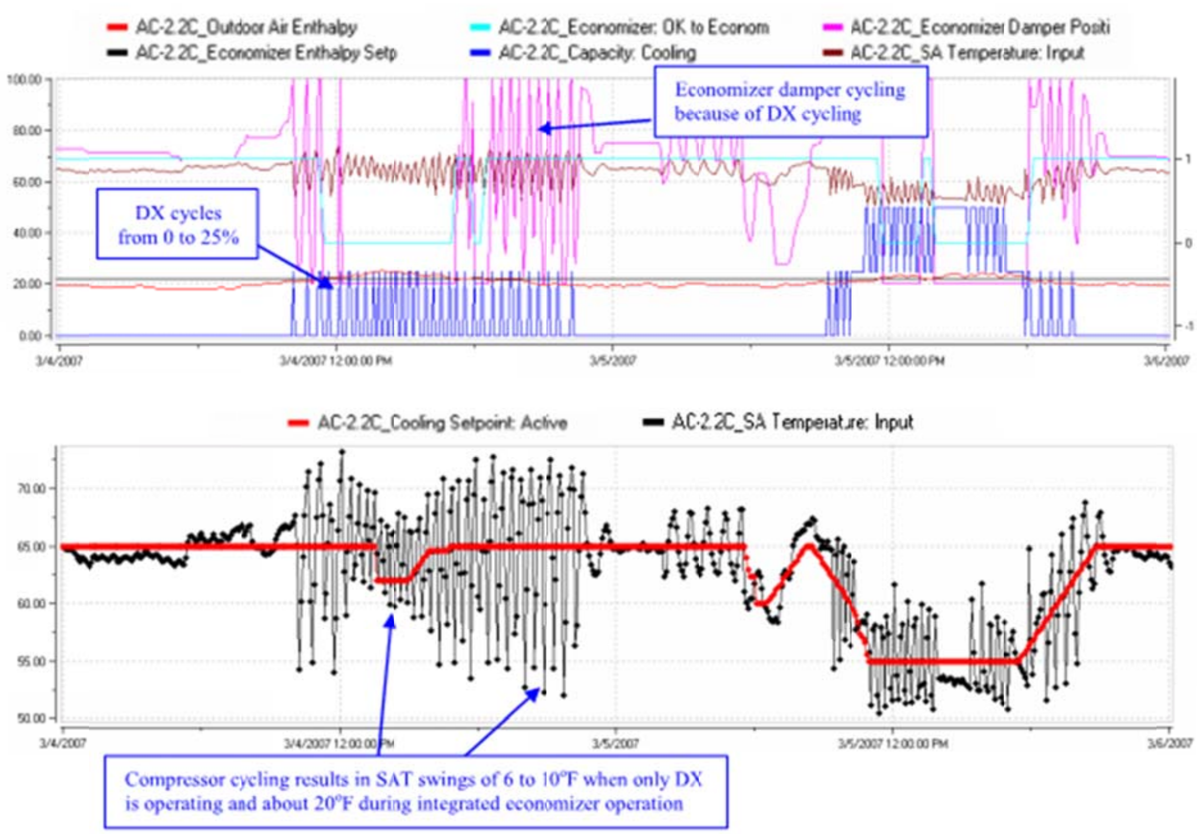


Figure 3 shows a bin analysis of damper positions for an AC unit over a 48-hour period in early March for hours during which the economizer is enabled. This unit operates in 24/7 cooling mode here but yet the economizer damper is fully open for less than half of the available hours because of the excessive cycling. The corresponding trends during the same period are shown Figure 4.

April 23, 2007
Page 4

Figure 3 – Binned Damper Position – M5 AC-1.2C

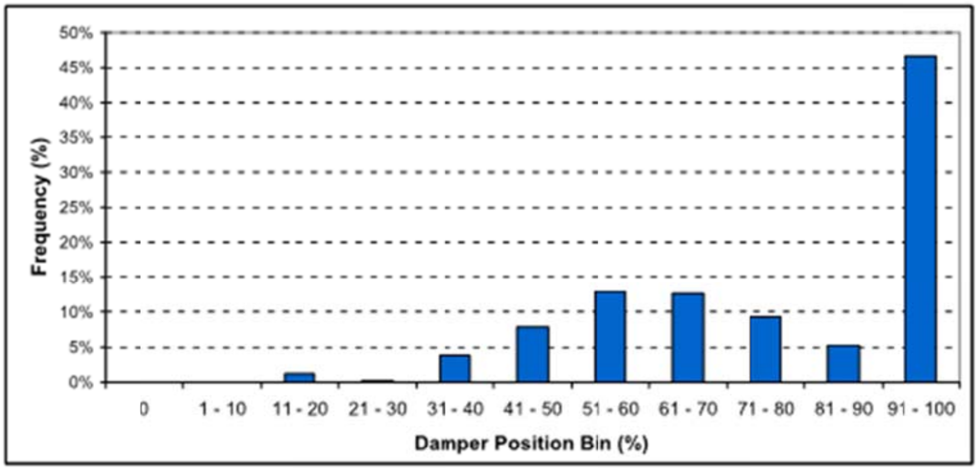
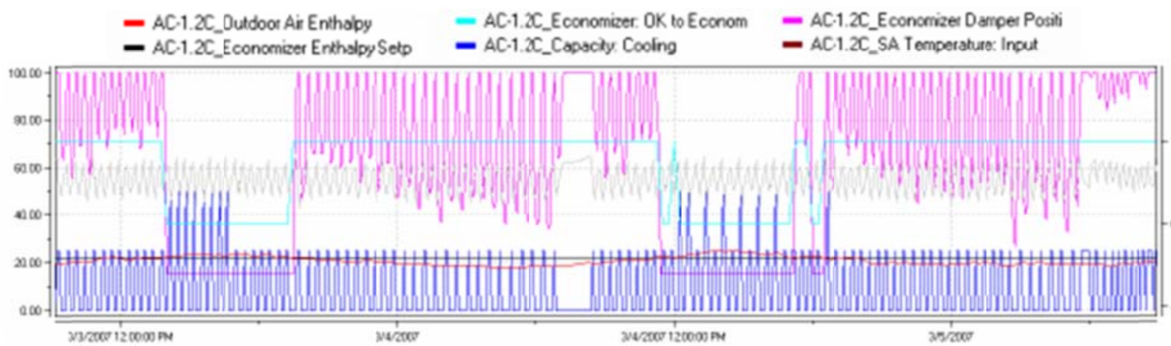
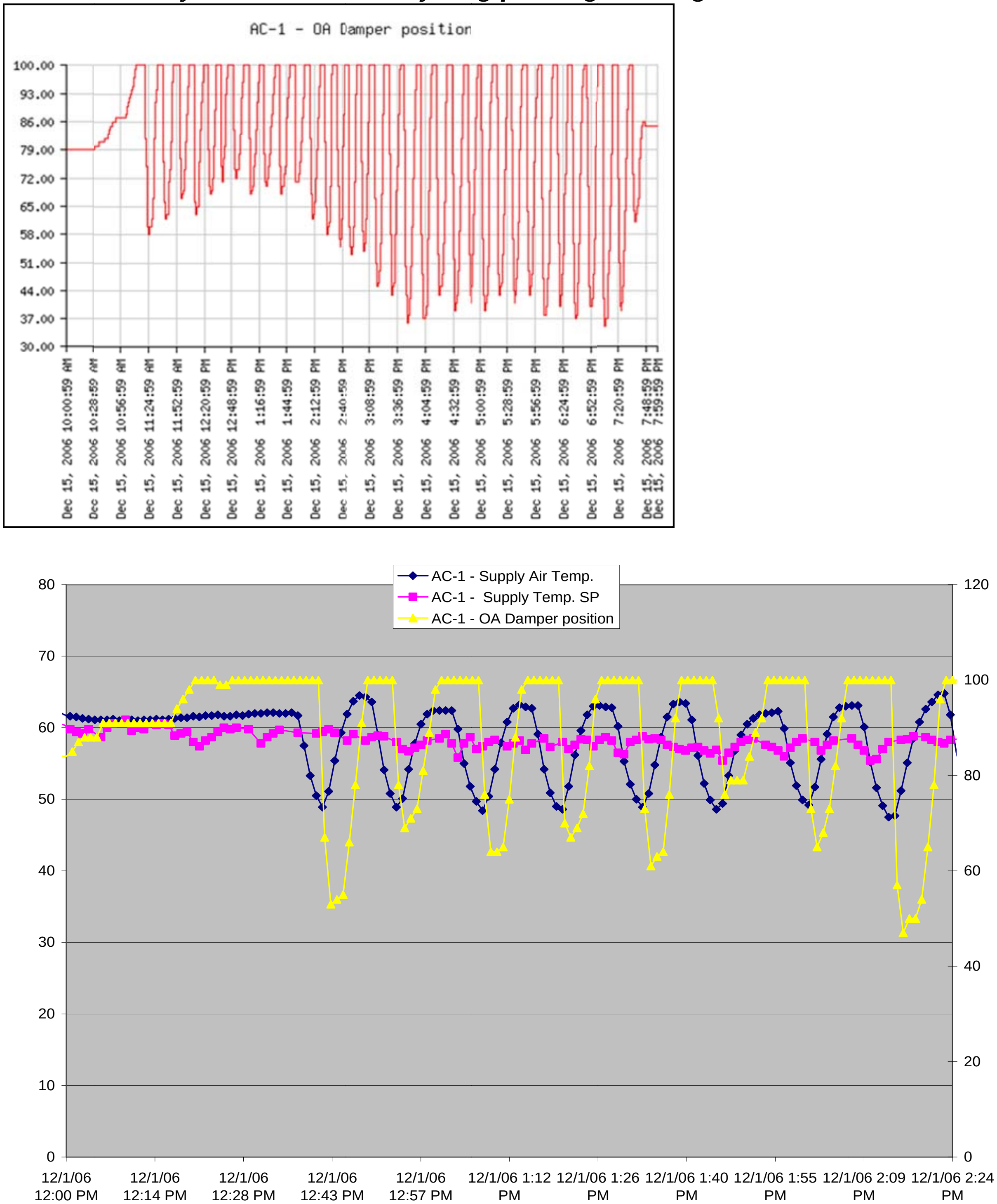


Figure 4 - AC Unit Control - M5 AC-1.2C



Trane has indicated that there may be adjustable internal controls within the Intellipak units that forces the economizer damper to partially close with the intent to limit excessive compressor cycling (but this may be prohibited by T24?).

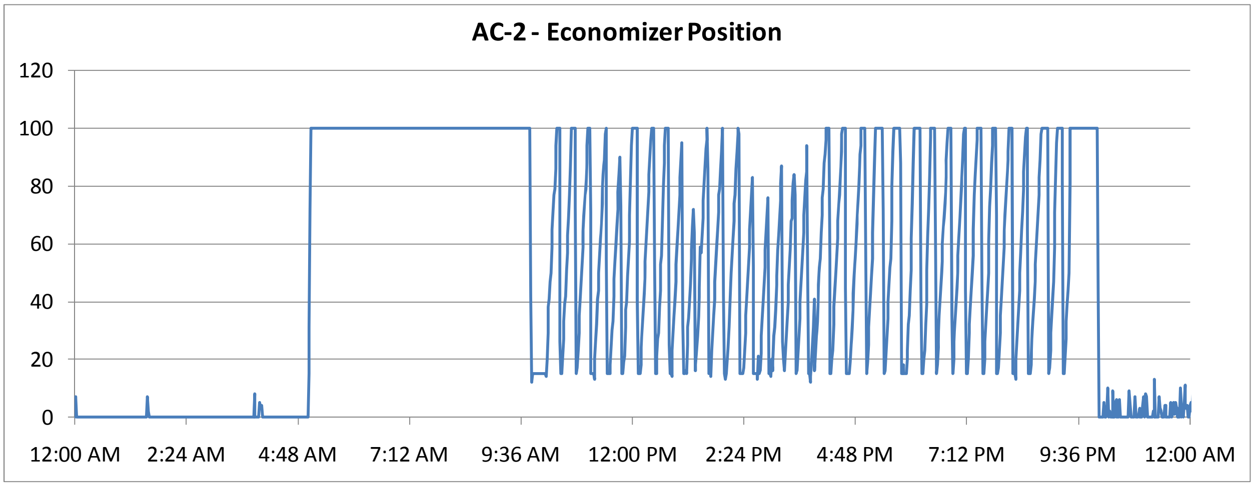
7.4 Case Study 2 in Economizer Cycling for Large Packaged Unit



Source: MHIRC trend log, 75 ton packaged VAV unit.

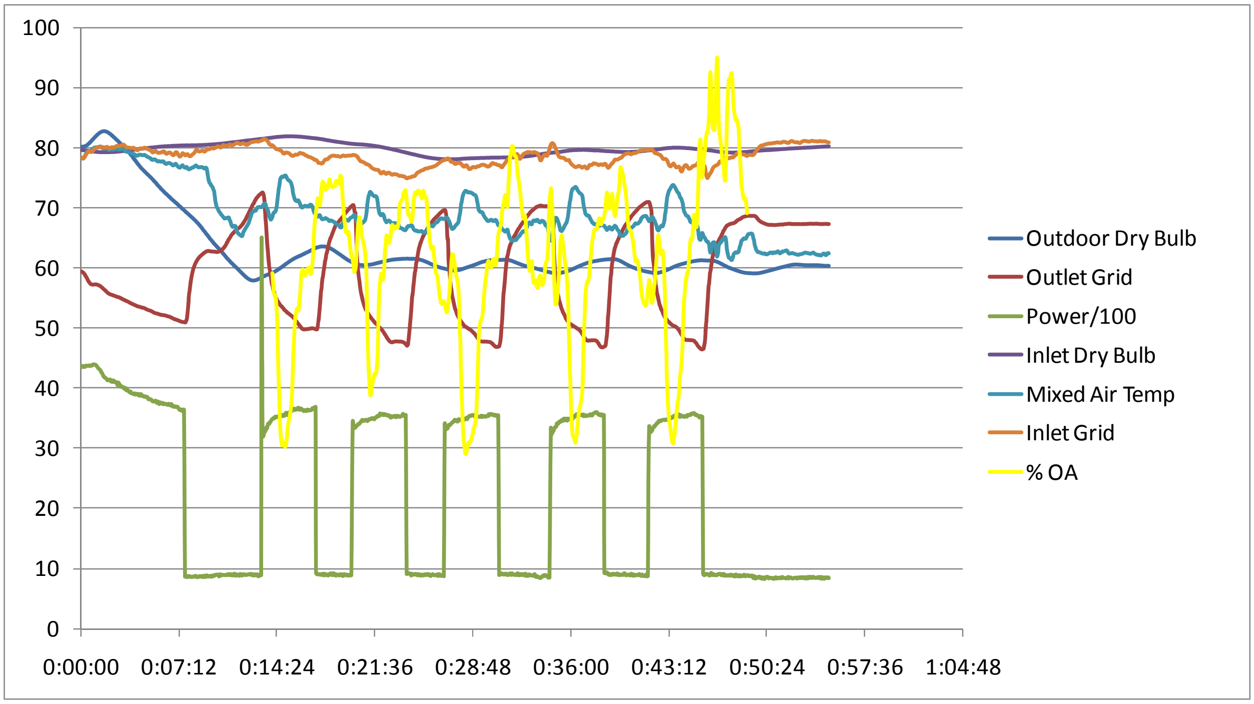
7.5 Typical Economizer Cycling for Small Packaged Unit

The average economizer position while the economizer is cycling during the day is 68%.



Source: MHIRC trend log, 15 ton packaged single zone unit.

The following data is for a 5-ton Lennox Strategos unit. The economizer is cycling as the compressor cycles and the average outdoor air fraction is 61% (based on outside air, return air and mixed air temperatures).



Source: PECI

7.6 Meeting Minutes for ASHRAE 90.1 Mechanical Subcommittee Packaged Working Group Conference Calls

90.1 Packaged Unit Working Group

11/3/10 Meeting Minutes

Attendees:

	7/14/10	9/9/10	11/3/10
Jeff Stein	X	X	X
Bing Liu	X		X
Dick Lord	X	X	X
David Handwork	X		
Randall Higa	X		
Steve Taylor	X	X	X
Mark Hydeman	X	X	X
Susanna Hanson		X	X
Dave Grassl		X	
Karim Amrane		X	
Deirdre McShane		X	X

1. Added a revamped table with hours in low-speed, high-speed, off in each zone for case
2. Added an explanation about 90.1 vs T24
3. Dick Lord said to steer clear of 2 speed fan requirement for <65,000 Btu/h because it would affect the SEER which is preempted by federal code (?)
4. Changed fan control language from “low speed” to “no more than 50 percent of design wattage at 66 percent of design fan speed.”
5. Rejected the idea (from Susanna) of having an IEER improvement instead of prescriptive requirement for the integrated economizer requirement
    a. Initially only changing the IEER would not account for integrated economizer performance
    b. Eventually it will be encompassed in the IEER when it becomes standard practice
6. Dick Lord suggested aiming for <3 year payback for industry approval of the measure
7. Dick Lord will continue to work on gathering cost data

90.1 Packaged Unit Working Group

9/9/10 Meeting Minutes

Attendees:

	7/14/10	9/9/10
Jeff Stein	X	X
Bing Liu	X	
Dick Lord	X	X
David Handwork	X	
Randall Higa	X	
Steve Taylor	X	X
Mark Hydeman	X	X
Susanna Hanson		X
Dave Grassl		X
Karim Amrane		X
Deirdre McShane		X

1. Dick's Analysis:
   a. Developing performance models for the following system types:
      i. Base case would be an 11.0 EER with an 11.2 IEER (gas fired unit requirement for 2010)
      ii. More accurate and industry typical single compressor unit
      iii. 2 stage 2 compressor equal size and 2 circuits
      iv. 2 compressor with 40/60 split and 2 circuits
      v. 2 compressor equal size and a single circuit
      vi. 2 compressor with 40/60 split and single circuit
      vii. Digital compressor with a single circuit
      viii. Variable speed compressor with a single circuit.
   b. The fan side will remain the same and we should look at 2/3 reduction 2 speed and a variable speed.
   c. Dick to develop 7 DOE-2 equations for each system type
      i. Separate out indoor fan?
   d. Dick to model different options in his spreadsheet tool
   e. Dick to provide DOE-2 equations for use by Taylor

2. Taylor Analysis
   a. Taylor is using 2 analysis approaches to model the different system types (this will provide 3 sets of results: 2 from Taylor and one from Dick)
      i. A high speed and a low speed run with results spliced together
      ii. The new equest Staged Capacity model
   b. We reviewed the Taylor modeling assumptions and provided feedback such as
      i. Use 2/3 fan speed for low speed
      ii. Use 30% fan power for 2/3 flow
3. IEER vs prescriptive?

   a. IEER set at worst case now
   b. Dick has done studies showing how fan speed and variable capacity compressors show up in IEER – share?
   c. Discuss after we get the results
2. ARI meeting in November 14-16 – results before then
3. Cost data
   a. Taylor and Dick to provide first cost thresholds to meet 8.8 scalar
      i. ARI to poll members to see if they feel they can meet the first cost threshold when accounting for mass production
   b. Blind ARI survey?
   c. Dave providing single point of reference
4. Next meeting: late October / early November
   a. Jeff to send Doodle poll

Assignments:
1. Dick
   a. Provide DOE-2 equations
   b. Share IEER studies of fixed vs variable speed fans, variable capacity compressors
   c. Provide list of benchmark cities
   d. Provide feedback on TSP (currently Taylor using 2.5”), and supply fan efficiency (currently using 53%)
2. Dave Handwork
   a. Provide pricing data

90.1 Packaged Unit Working Group

7/14/10 Meeting Minutes

Attendees:

Jeff Stein	X
Bing Liu	X
Dick Lord	X
David Handwork	X
Randall Higa	X
Steve Taylor	X
Mark Hydeman	X

Consensus: focus only on >= 65k Btu/h

DOE is working on regional requirements starting in 2016 but only for single phase. 3 phase will still be covered by 90.1. so we need to track what DOE is doing.

Cost Data

- Dick to discuss with ARI ULE group to get average incremental cost as a function of size, efficiency, technology, etc.
- Desired options for costing:
    1. Basecase – partial integration
    2. 2 speed fan (single stage compressors) – partial integration
    3. 2 speed fan + 2 stage compressor – partial int
    4. Variable speed fan + variable capacity compressors (50% capacity min) – fully integrated economizer
    5. Variable speed fan + variable speed compressors
- Cost data for each option to account for future mass production assuming that option is effectively required by code
- Incremental does not include extra bells and whistles like bacnet gateway that are typically included in Tier 2 and 3 but not included in basic models
- Incremental maintenance costs?
- Incremental costs beyond equipment costs? – electrical? Controls? GC?

Energy Analysis - Taylor Engineering to do the energy modeling in equest

- Building models
    - Small office is most conservative so only model small office
    - Retail, classroom are less conservative
    - Use Bing's scorecard to develop building model
    - Same baseline cities
- 2 speed fan analysis
    - run single speed, then run 2 speed fan, then back calculate incremental IEER
    - Requires hourly post processing to pick hrs when low speed works
    - Ask Hirsch if equest can incorporate 2 speed

- Compressor analysis
    - Dick to help develop DOE2 curve for variable capacity and variable speed compressors
- Economizer modeling:
    - Partial integration: model integrated and non-integrated and split the difference for non variable capacity compressors.
    - Variable capacity and variable speed compressors will be modeled as fully integrated

7.7 Load Profiles for Integrated Economizer

The Realistic office occupancy load profiles were generated. Five different schedules were generated and randomly assigned to each zone on different days. This was done to accurately model the net effect on a multiple zone air handler. For simplicity the same schedules were used for lights, people, and equipment.

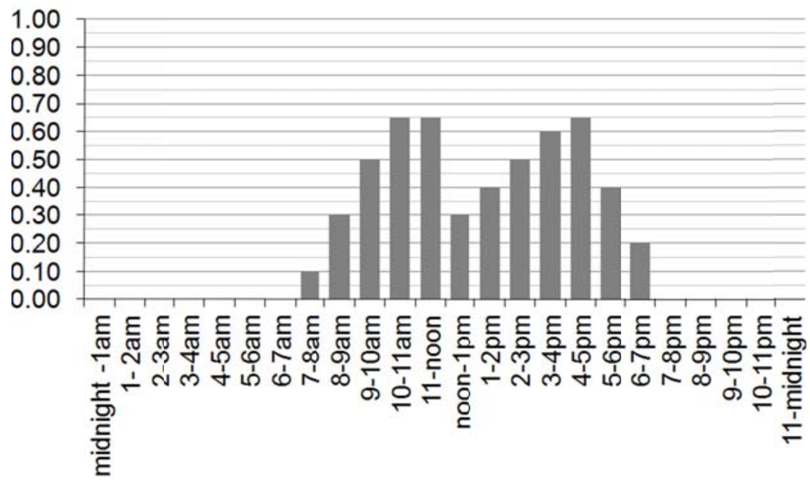


Figure 36. Integrated Economizer – Multiple Zone DX: Load Profile 1

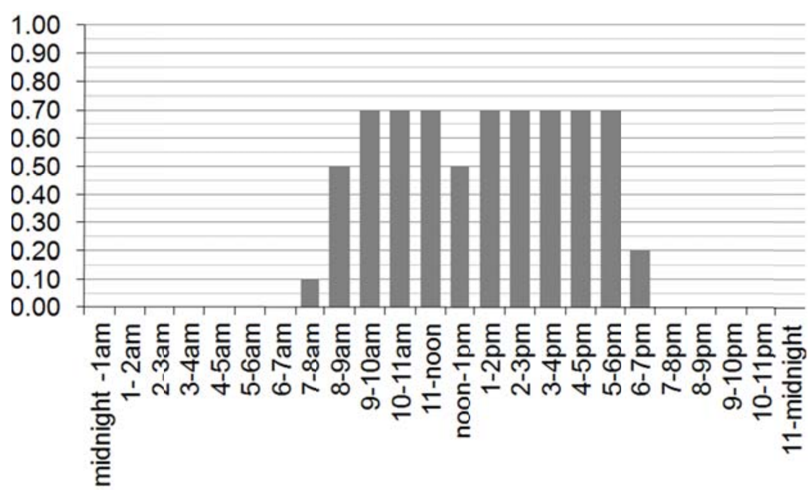


Figure 37. Integrated Economizer – Multiple Zone DX: Load Profile 2

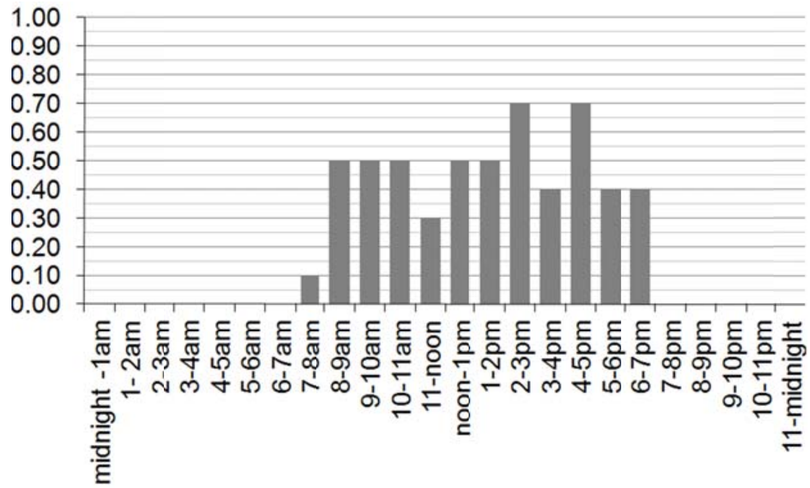


Figure 38. Integrated Economizer – Multiple Zone DX: Load Profile 3

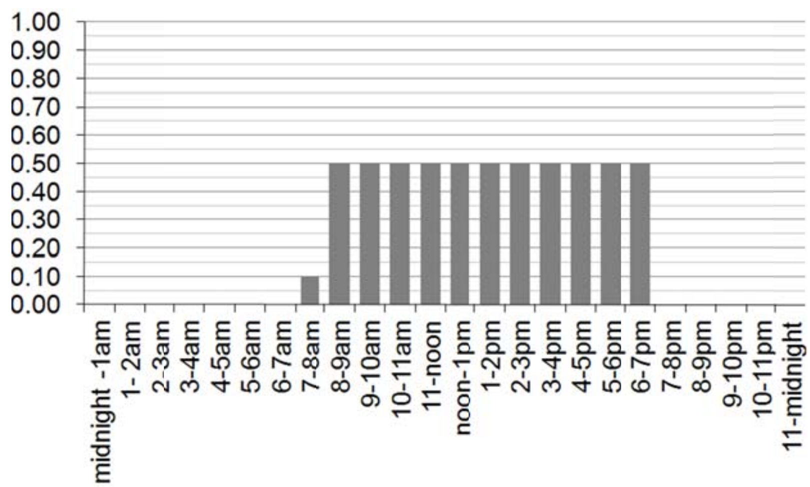


Figure 39. Integrated Economizer – Multiple Zone DX: Load Profile 4

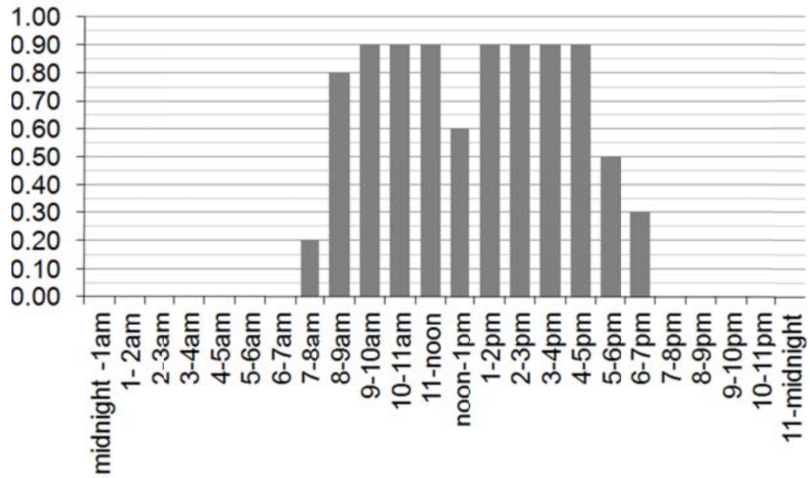


Figure 40. Integrated Economizer – Multiple Zone DX: Load Profile 5

7.8 Email Exchange with Richard Lord – Cost data from AHRI

From: Lord, Richard CAR [mailto:Richard.Lord@carrier.utc.com]
Sent: Tuesday, February 01, 2011 3:44 PM
To: Jeff Stein
Subject: RE: fan control and integrated economizer

The full AHRI ULE section was polled and then voted on this. I don't have the details on who responded and how many manufacturers supplied data, but they all were given the opportunity to vote and it did pass by a majority vote.

VRF is not part of this section and technically will not fall under the requirements new requirement as the capacity of the fan coils will fall below the capacity threshold that you have defined at 65K in almost all cases. Not that the typically interpretation is that the requirements apply to each fan coil and not to the central condensing unit.

These are current cost and not attempt has been made to project 2015 costs. They are also assuming that they are based on a national requirement and due not reflect the cost that would be if the requirement was just unique to California.

I will try to see if I can get more details on the votes.

Also if you can seen me the presentation, I have a meeting next week with the Engineering committee and would like to share this with them.

From: Jeff Stein [mailto:JStein@taylor-engineering.com]
Sent: Tuesday, February 01, 2011 5:27 PM
To: Lord, Richard CAR
Subject: RE: fan control and integrated economizer

Dick,

Thank you for making this happen. Can you provide some more details? How many manufacturers responded? Which manufacturers were polled? Were the VRF manufacturers included, for example?

Are these current costs or do they reflect the expected incremental cost in 2015 assuming 2 speed/variable speed fan control and integrated economizers are required?

--Jeff Stein

From: Lord, Richard CAR [mailto:Richard.Lord@carrier.utc.com]
Sent: Monday, January 31, 2011 9:56 AM
To: Jeff Stein
Subject: RE: fan control and integrated economizer

Here is the cost information.

ASHRAE 90.1 65K to 110K Evaluation Consuner Price Information

Option	Description	Incremental Cost			
		6 ton	7 ton	8 ton	10 ton
1	Single Stage Compression with 2 speed fan	$ 263.50	$ 214.00	$ 276.50	$ 270.67
2	2 Stage Compression with 2 speed fan	$ 496.00	$ 556.00	$ 655.67	$ 722.00
3	Variable capacity compressor, 2 speed fan	$ 1,190.33	$ 1,306.00	$ 1,484.33	$ 1,663.33
4	Variable capacity compressor, variable speed fan	$ 2,133.00	$ 2,374.00	$ 2,708.00	$ 3,148.00

Note 1 2 speed fans will have a lower speed of 2/3 or lower
Note 2 Cost data should be the incremental price to a customer for the feature
Note 3 Although we have provided cost for option 1, it is not a viable option and will cause operational problems

Here is the information on the compressor. Not quite as bad as I indicated, but in many of our applications we have found that at light loads it is better to cycle the compressor off vs running at very low capacities and we use this technique to improve SEER and IEER

Can you send me the presentation that you reviewed at the meeting.

We will get back to you with some alternatives on the economizer integration logic.

From: Jeff Stein [mailto:JStein@taylor-engineering.com]
Sent: Sunday, January 30, 2011 9:36 PM
To: Lord, Richard CAR
Subject: fan control and integrated economizer

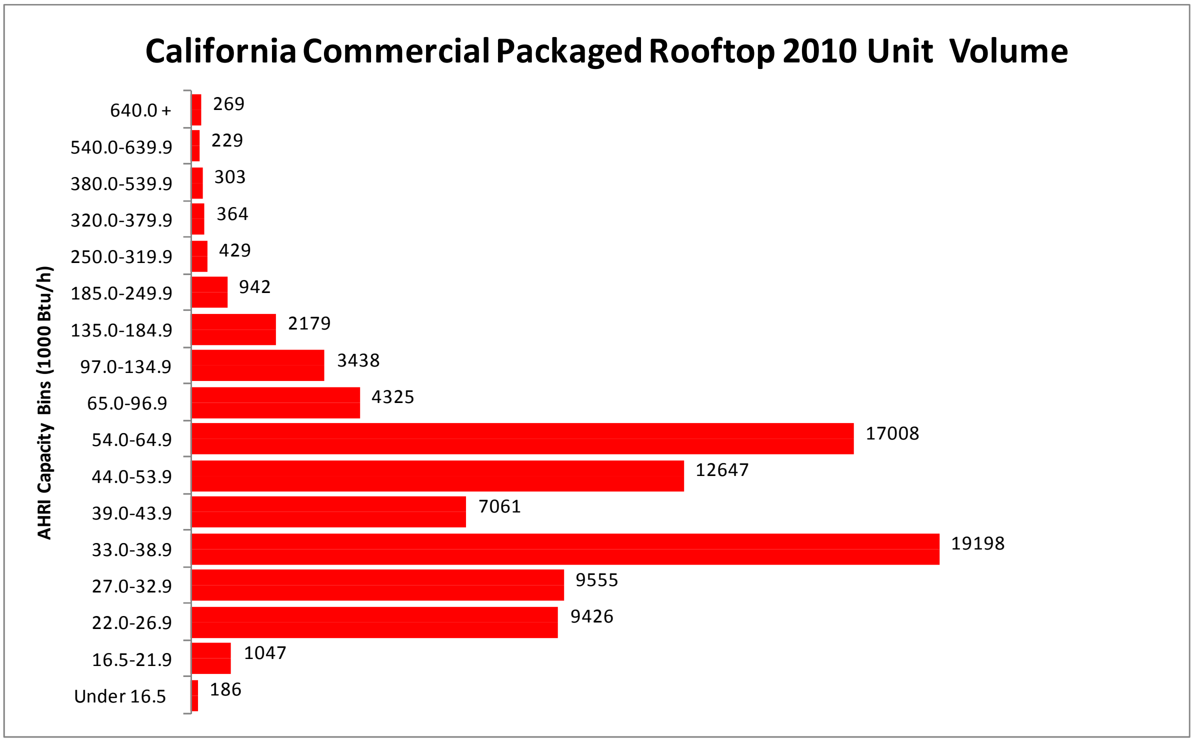
Dick,

Can you send me the AHRI cost data and the digital scroll part load power curve?

Sincerely,
Jeff Stein, P.E.
Principal, Taylor Engineering, LLC
1080 Marina Village Parkway, Suite 501, Alameda CA 94501-6427
(510) 263-1547 direct, (510) 749-9135 main, (510) 749-9136 fax, (510) 220-3932 mobile
jstein@taylor-engineering.com
www.taylor-engineering.com

7.9 California Commercial Packaged Rooftop Unit Volume Added in 2010

Bar graph provided by Dick Lord of AHRI.



AHRI Capacity Bins (kBtu/h)		**Number of Units**	**Average size**	**Total Capacity per bin**	
Min	**Max**		**(kBtu/h)**	**(kBtu/h)**	**(Tons)**
0.0	16.4	186	8.2	1525	127
16.5	21.9	1047	19.2	20102	1675
22.0	26.9	9426	24.45	230466	19205
27.0	32.9	9555	29.95	286172	23848
33.0	38.9	19198	35.95	690168	57514
39.0	43.9	7061	41.45	292678	24390
44.0	53.9	12647	48.95	619071	51589
54.0	64.9	17008	59.45	1011126	84260
65.0	96.9	4325	80.95	350109	29176
97.0	134.9	3438	115.95	398636	33220
135.0	184.9	2179	159.95	348531	29044
185.0	249.9	942	217.45	204838	17070
250.0	319.9	429	284.95	122244	10187
320.0	379.9	364	349.95	127382	10615
380.0	539.9	303	459.95	139365	11614
540.0	639.9	229	589.95	135099	11258
640.0	n/a	269	640	172160	14347
				5,149,671	**429,139**

Calculations used in the Annual Statewide Savings in Section 1.4 are displayed in the table below. Average Savings of 965 kWh/ton/yr are the average savings for the six climate zones modeled in Section 3.5. An even distribution of units between 97.0 and 134.9 kBtu/h was assumed and 34% of the units in this capacity range were included accordingly, ie: (110-97)/(134.9-97) = 34%.

Capacity (kBtu/h)		Number of Units	Average Unit Size	Average Savings	Total Tonnage	Annual Statewide Savings
Min	**Max**		**(kBtu/h)**	**(kWh/ton/yr)**	**(tons)**	**(kWh/yr)**
65.0	96.9	4325	81	965	29176	28,157,496
97.0	110	1179	104	965	10171	9,816,156
					39347	**37,973,652**