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Air-Conditioning, Heating, and Refrigeration Institute

October 31, 2011

California Energy Commission (CEC)

Re: October 13, 2011 Nonresidential Staff Workshop - 2013 Building Energy Efficiency Standards (AHRI Comments on Measures Related to Unitary Air Conditioners and Heat Pumps)

Dear CEC Staff:

The Air-Conditioning, Heating and Refrigeration Institute (AHRI) is the trade association representing manufacturers of heating, cooling, water heating, and commercial refrigeration equipment. Over 300 members strong, AHRI is an internationally recognized advocate for the industry, and develops standards for and certifies the performance of many of the products manufactured by our members. In North America, the annual output of the HVACR industry is worth more than $20 billion. In the United States alone, our members employ approximately 130,000 people, and support some 800,000 dealers, contractors, and technicians.

We have several concerns with respect to the measures discussed at the October 13, 2011 CEC staff workshop. We urge CEC to reconsider its code change proposals with respect to fan control, integrated economizers and single zone VAV. Although we appreciate the opportunity to provide these comments, we recommend that CEC immediately convene a meeting with our industry either via teleconference or face-to-face in order to discuss and resolve our concerns.

The inclusion of efficiencies (i.e.; those effective January 1, 2015) in Tables 110.2-A that are more stringent than efficiencies listed in Tables 6.8.1A of ASHRAE 90.1-2010 is in violation of federal preemption. All HVAC products listed in tables 110.2 A are regulated by the Energy Policy and Conservation Act, Pub. L. 94-163, 89 Stat. 926 (1975) ("EPCA"), as amended by the Energy Policy Act of 1992, Pub. L. No. 102-486, 106 Stat. 2776 (codified at 42 U.S.C. § 6311 *et seq.*) ("EPACT"). EPCA created a uniform, national regulatory scheme that governs all aspects of appliance efficiency, including standards, regulatory terminology, testing requirements, labeling, and other disclosures of information. This regulatory scheme was intended to stop a patchwork of state regulations and to ensure that products meeting federal requirements can be sold in the entire U.S. without restrictions Under these federal regulations, states, cities and other jurisdictions are preempted from setting efficiency standards beyond those mandated by the Department of Energy and/or ASHRAE 90.1, Consequently, the California Energy Commission cannot require minimum EER levels that are greater than the requirements of ASHRAE 90.1 as proposed in Tables 110.2A for the January 1, 2015 effective date.

Furthermore, Tables 110.2-A and 110.2-B do not accurately capture all subcategories, rating conditions and size categories. Additionally, industry moved away from the IPLV metric on January 1, 2010 and is now using the Integrated Energy Efficiency Ratio (IEER) metric to capture the part load performance of unitary equipment. Consequently, we ask that CEC replace IPLV with IEER. All test procedure reference needs to be changed from ARI to AHRI. CEC should harmonize the information in Tables 110.2-A and 110.2-B with the efficiencies, subcategories, rating conditions, size categories and metrics in Tables 6.8.1A and 6.8.1B of ASHRAE 90.1-2010.

Attached is a presentation that summarizes an analysis of a similar proposal that has been made to ASHRAE 90.1 for the 2 speed fan and integrated economizer with some additional comments added to correlate with the Title 24 proposal. Unfortunately the proposal to Title 24 and ASHRAE 90.1 lumps the 2 speed fan and integrated economizer change into one change. The study shows that the 2 speed fan can be easily justified, but the minimum capacity requirement of 20% cannot be justified. The study also found serious issues with some of the assumptions used in the study used to justify the changes which are also documented in the report.

There has been very little work done with the manufacturers of the equipment by the developers of the Title 24 proposal and ASHRAE 90.1 proposal. The AHRI members feel that a proposal can be developed that can take advantage of the large energy savings of the 2 speed fan and also improve the issues with the integrated economizer that were identified in the Title 24 supporting documents in a cost effective approach that can be supported by equipment designs. AHRI is working on this proposal and very much would like to meet with the CEC to develop a consensus proposal that could be implemented in Title 24 as well as in ASHRAE 90.1 as a national requirement which would result in the lowest cost solution. Additionally, AHRI would like CEC to consider the following changes to the proposed code language:

§140.4(e)2:

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

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

      EXCEPTION to Section ~~144~~140.4(e) 2A: Systems that provide 75 percent of the annual energy used for mechanical heating from site-recovered energy or a site-solar energy source.

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

      i. Direct expansion systems with a cooling capacity < 65,000 Btu/hr and with an economizer shall have control systems, including two-stage or electronic thermostats, which cycle compressors off when economizers can provide partial cooling.

      ii. 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.~~ shall have mechanical capacity control that is interlocked with the economizer control such that the economizer does not begin to close until the unit leaving air temperature is less than 45 F. All constant volume units including 2 speed fan systems must have a minimum of 2 stages of mechanical cooling. Variable air volume units must have a minimum of 4 stages or variable capacity with a minimum capacity of 25%.

[a] See Tables 110.2-A and 110.2-B for rating standard and conditions.

§140.4(m).

(ml) **Fan Control.** ~~Each multiple zone system and single zone system listed in Table 140.4-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.~~ ~~**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.~~

Air-handling and fan-coil units with chilled-water cooling coils and supply fans with motors greater than or equal to 5 1/4 hp shall have their supply fans controlled by two-speed motors or variable-speed drives. At cooling demands less than or equal to 50% for proportionally controlled units and for 2 stage control units operating on the first stage, the supply fan controls shall be able to reduce the airflow to no greater than the larger of the following:

One half of the full fan speed, or

The volume of outdoor air required to meet the venti-lation requirements of Standard 62.1.

Constant volume units shall also have a minimum of 2 stages of capacity and shall be capable of operating the economizer, if required, with 2 stage fan speed control with operation at low speed when the economizer capacity is less than 50%. Variable air volume units shall have a minimum of 4 stages with a minimum stage of 25% or less.

All air-conditioning equipment and air-handling units with direct expansion cooling and a cooling capacity at AHRI conditions greater than or equal to 65,000 Btu/h shall have their supply fans controlled by two-speed motors or variable-speed drives. Constant volume units at cooling demands less than or equal to 50% for proportionally controlled units and for 2 staged controlled units operating on first stage, the supply fan controls shall be able to reduce the airflow to no greater than the larger of the following:

Two-thirds of the full fan speed, or

The volume of outdoor air required to meet the venti-lation requirements of Standard 62.1.

The requirements for 65,000 to 110,000 Btu/hr capacity are effective 1/1/2015 and greater than 110,000 Btu/hr are effective immediately.

We would appreciate an opportunity to discuss this study and our alternate proposals with CEC staff as soon as possible, so that staff can make an informed decision on the various measures. Also attached are the comments that AHRI submitted to CEC and its consultants earlier this year. If you have any questions or wish to discuss this further, please do not hesitate to call me at (703) 600-0383.

Sincerely,

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FAN CONTROL AND INTEGRATED ECONOMIZER PROPOSAL COMMENTS

ASHRAE Meeting 10-21-2011

Richard Lord

Introduction

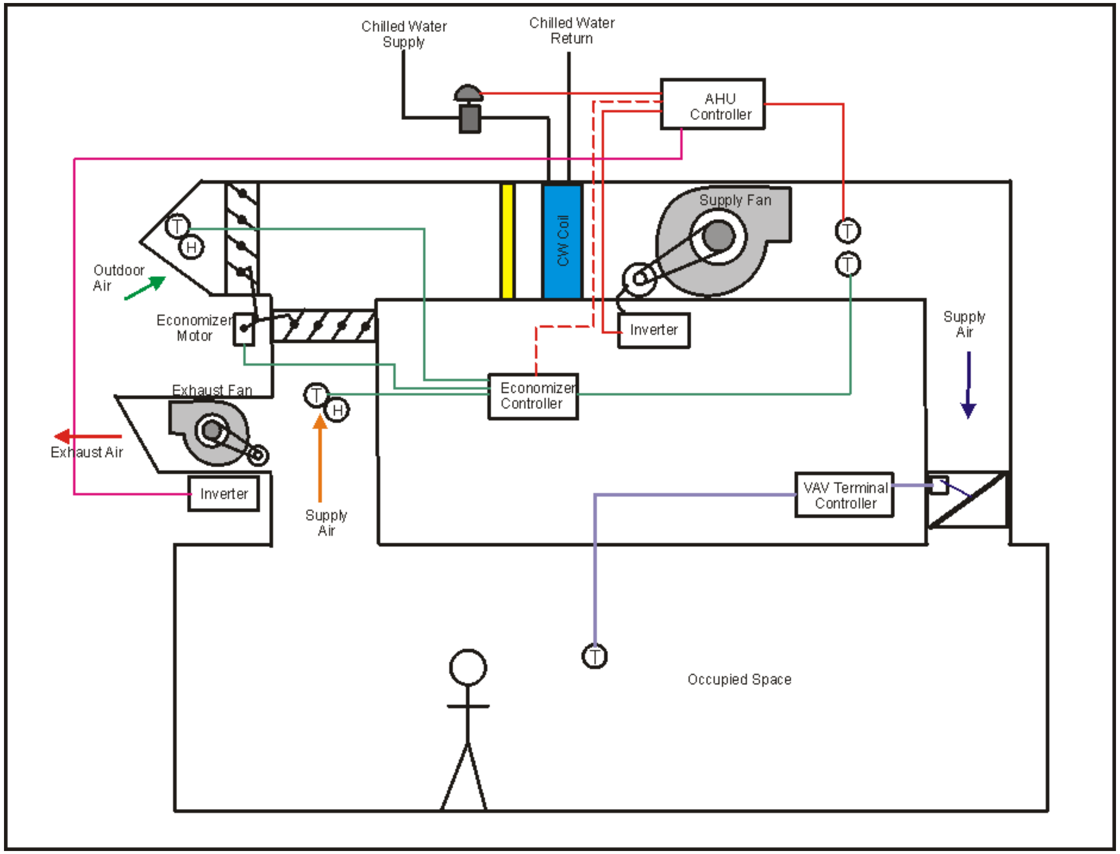
- Overall the industry does not support the proposal as currently written and this has been communicated thru AHRI
- It is likely that the industry will support the fan speed proposal, but do not support the variable capacity which has been proposed for integrated economizers
- We would prefer that the proposal be separated into two proposal as the fan speed reduction is driving most of the energy savings.

Economizer Cycling

- In the CMP justification for the integrated economizer proposal a plot of cycling problems with an economizer wad documented and this is driving the proposal for capacity modulation down to 20% actual capacity
- The unit in question was a VAV unit and already had capacity control down to 25% and moving this to 20% will not eliminate the issue.
- I believe the problem is that the economizer and capacity control are controlling to the same temperature sensor, but appear not to be link in software which results in one overriding the other and causing the cycling
- What would solve the problem is linking the economizer and compressor control such that the dampers are locked open during integrated compressor operation. There are many products on the market today that do not have the problem as mentioned.
- In the range of application for the proposal which is 5 to 10 tons the products are essential all constant volume and the economizer controls and the operating conditions are different.
- For constant volume units the airflow should be at full cfm when in economizer mode so the amount of capacity control is not as critical vs. the VAV units where the cfm is reduced during the economizer cycle.
- There are many control routines in use that limit the cycling and in fact these were simulated in the economizer proposal that we approved last year.
- Some of these are;
  - Lock the dampers open and only cycle the economizer when the leaving air temperature drops below 40-45 F
  - Lock the dampers open and then modulate them closed proportionally between 55 F and 45 F
  - Set the economizer set point low, 50-53 F and then when Y2 comes on it will not override the economizer

Background – Equipment Configurations

Typical Large VAV Chilled Water System



Typically supply air set point is 55 F and is used for both the economizer and chilled water coil

ASHRAE 90.1 Supply Reset

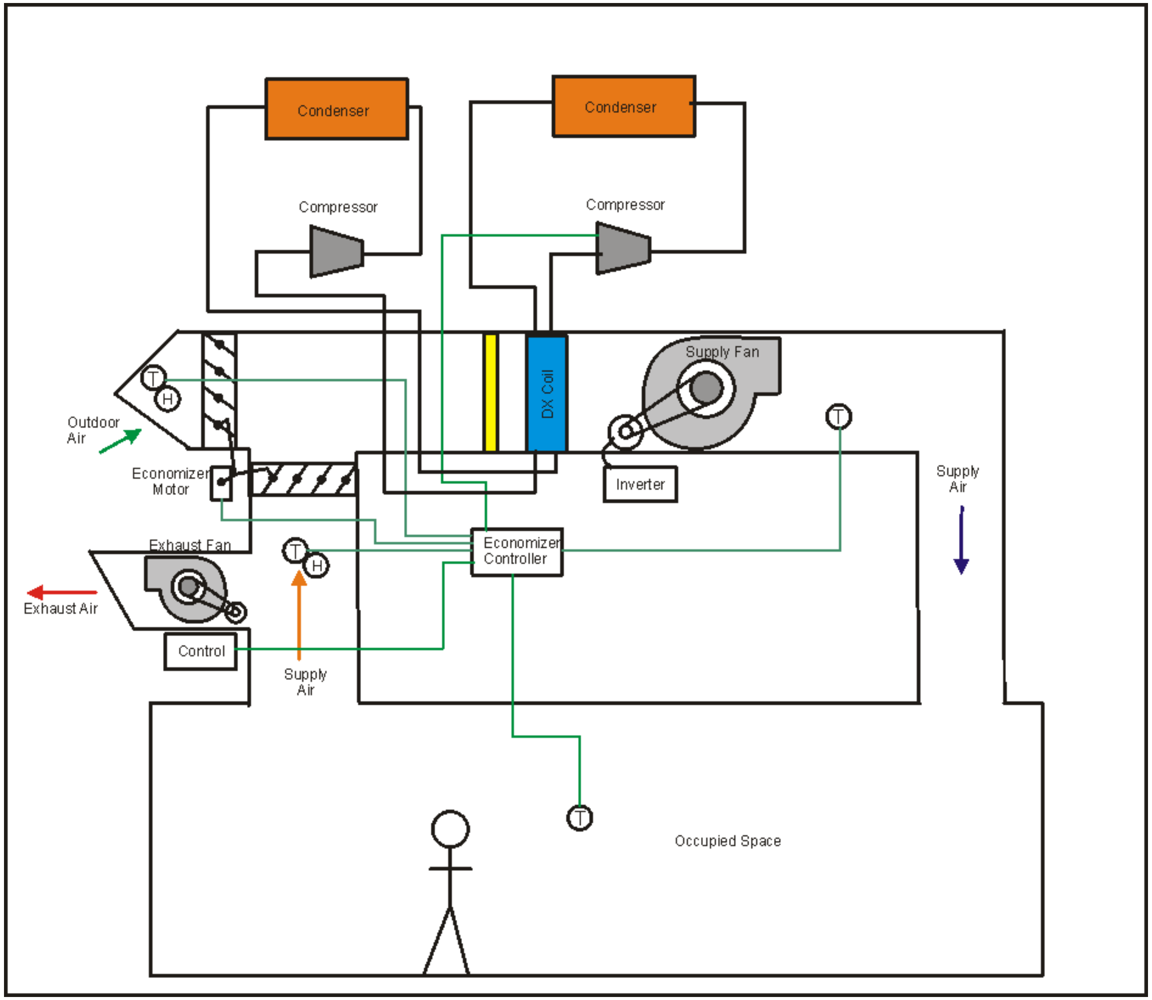
6.5.3.4 Supply-air temperature reset controls. Multiple zone HVAC systems must include controls that automatically reset the supply-air temperature in response to representative building loads, or to outdoor air temperature. The controls shall reset the supply air temperature at least 25 percent of the difference between the design supply-air temperature and the design room air temperature. Controls that adjust the reset based on zone humidity are allowed. *Zones* which are expected to experience relatively constant loads, such as electronic equipment rooms, shall be designed for the fully reset supply temperature.

Because the VAV system cfm is a function of the building load, I found that for the benchmark buildings the average maximum cfm during economizer operation is around 50% so the full benefit of economizers is not obtained. Reset helps.

Typical Reset

Supply air set point = 55 F
Space set point = 75 F
Max Reset = .25 x (75-55) = 5 F
Max Reset Temperature = 60 F

Background – Medium Packaged Rooftop



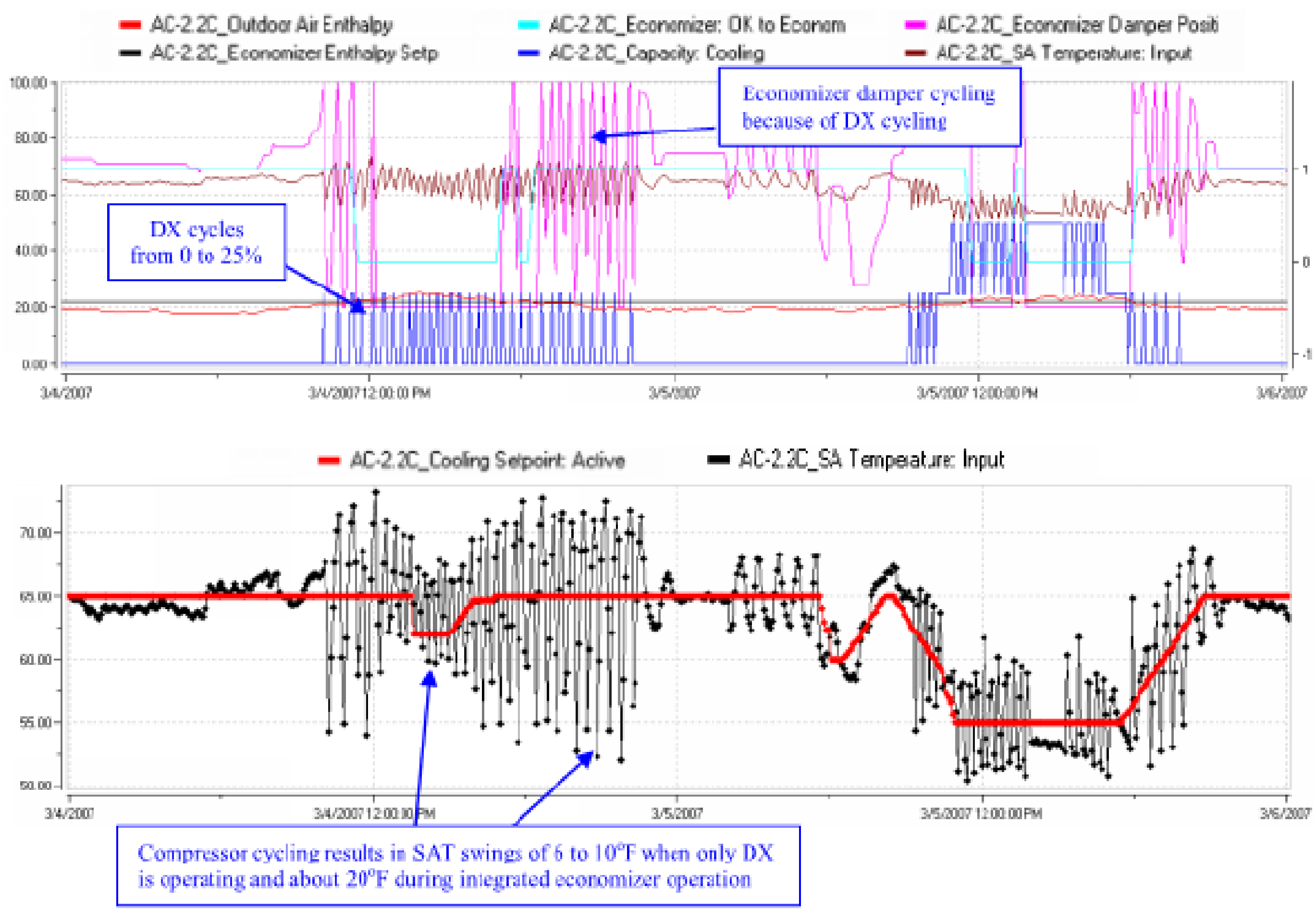
For the large units many units use a 2 compressor design, but each compressor is in a separate circuit with a face split coil

Again they are mostly constant volume and are controlled directly by a thermostat.

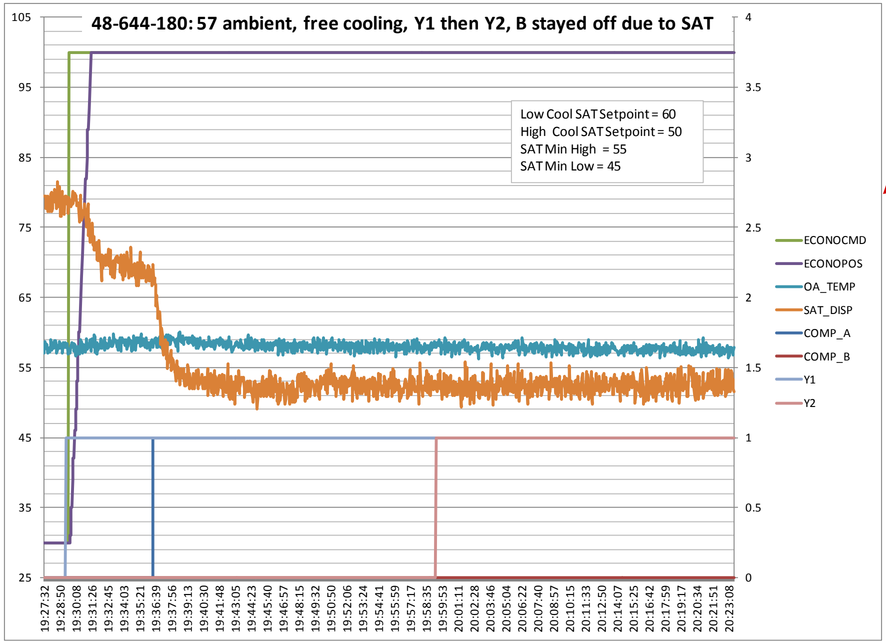
Reference Problem Equipment Data

This is a temperature trace from a VAV units that had a minimum capacity step of 25% where the dampers and compressor were fighting each other

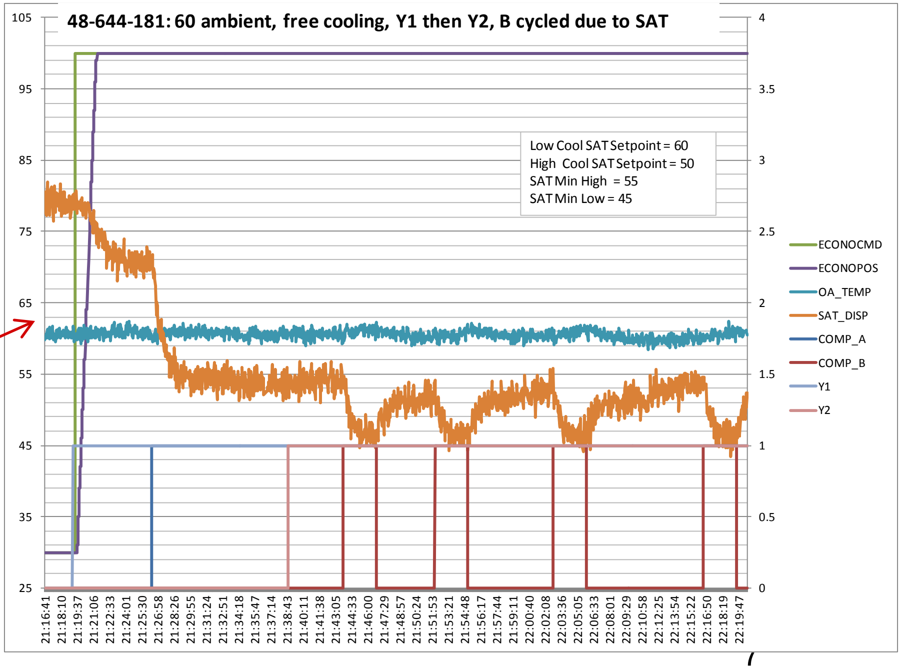
Figure 2 – AC Unit Control – M5 AC-2.2C



Typical 10 Ton Constant Volume 2 Stage Rooftop

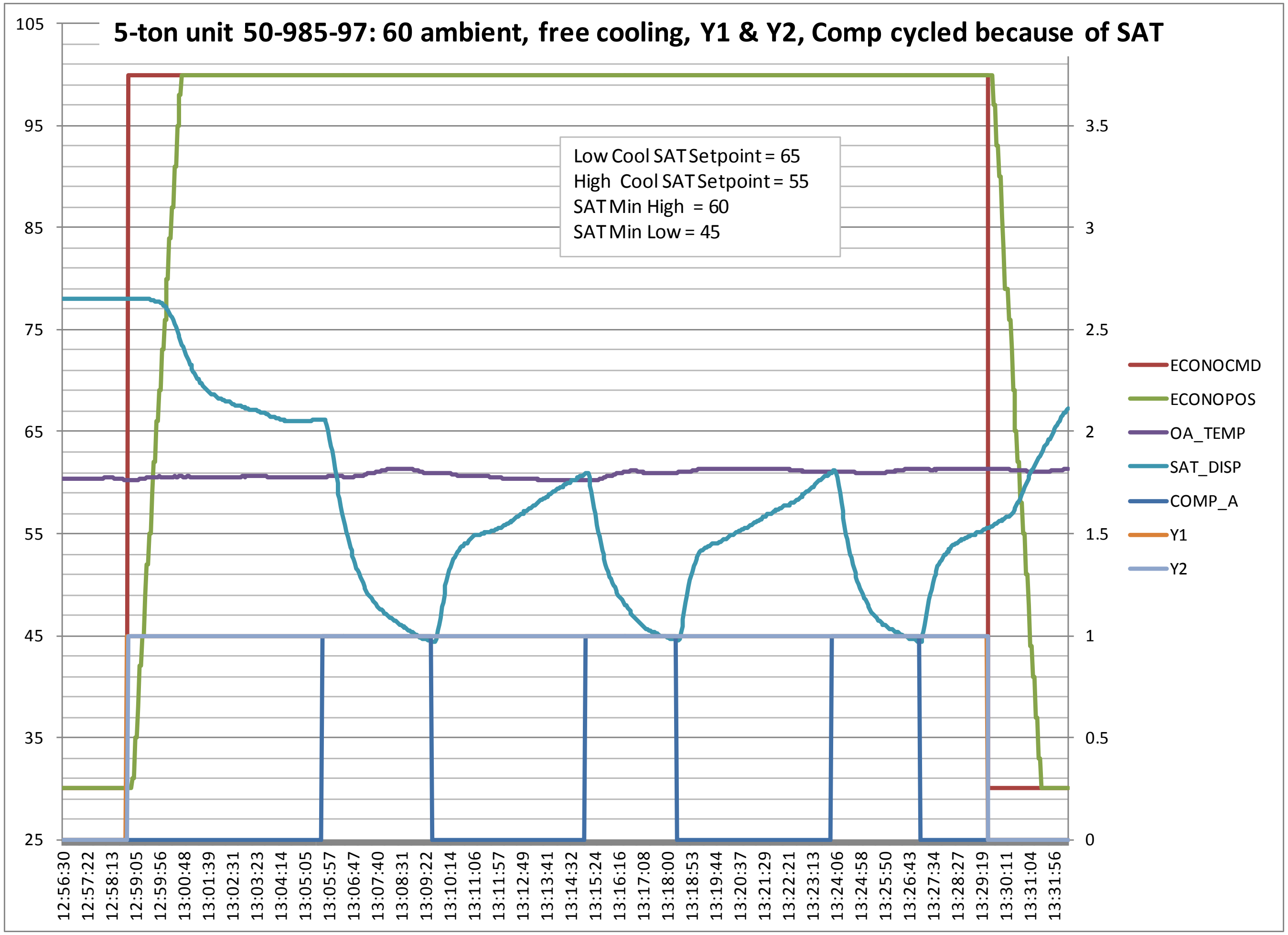


Standard Operating with economizer locked to 100% and stage 1 cooling on



Forced compressor 2 on even though typical loads would not require that much capacity but you can still see the economizer never closed

Typical 5 Ton Single Stage Rooftop



Again even with a single compressor the economizer on the constant volume units stays 100% open

Conclusions for Integrated Economizers

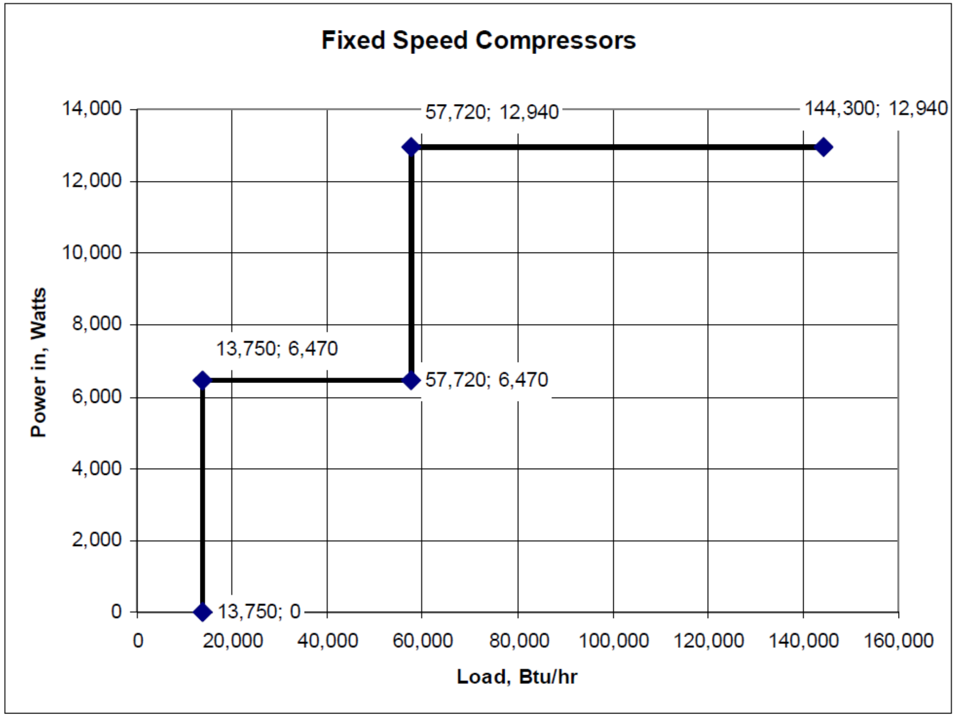
- Using modulating compressors on constant volume is not the best solution and will be a very expensive option compared to controls based solutions that are essentially very low to no cost options.

- Modulating capacity down to 25% or even lower on VAV units is important and in fact should be lower, but controls requirements are still needed to interlock the compressors and economizers

- I believe that a single compressor with an economizer is not ideal and would recommend to improve economizer integration that 2 compressor stages are used along with requirements on controls similar to what is required in ASHRAE 189.1

Compressor Efficiency

- In one of the reference papers a plot was shown that indicated variable speed compressors are significantly better at reduced load. **<u>This curve is totally wrong.</u>**

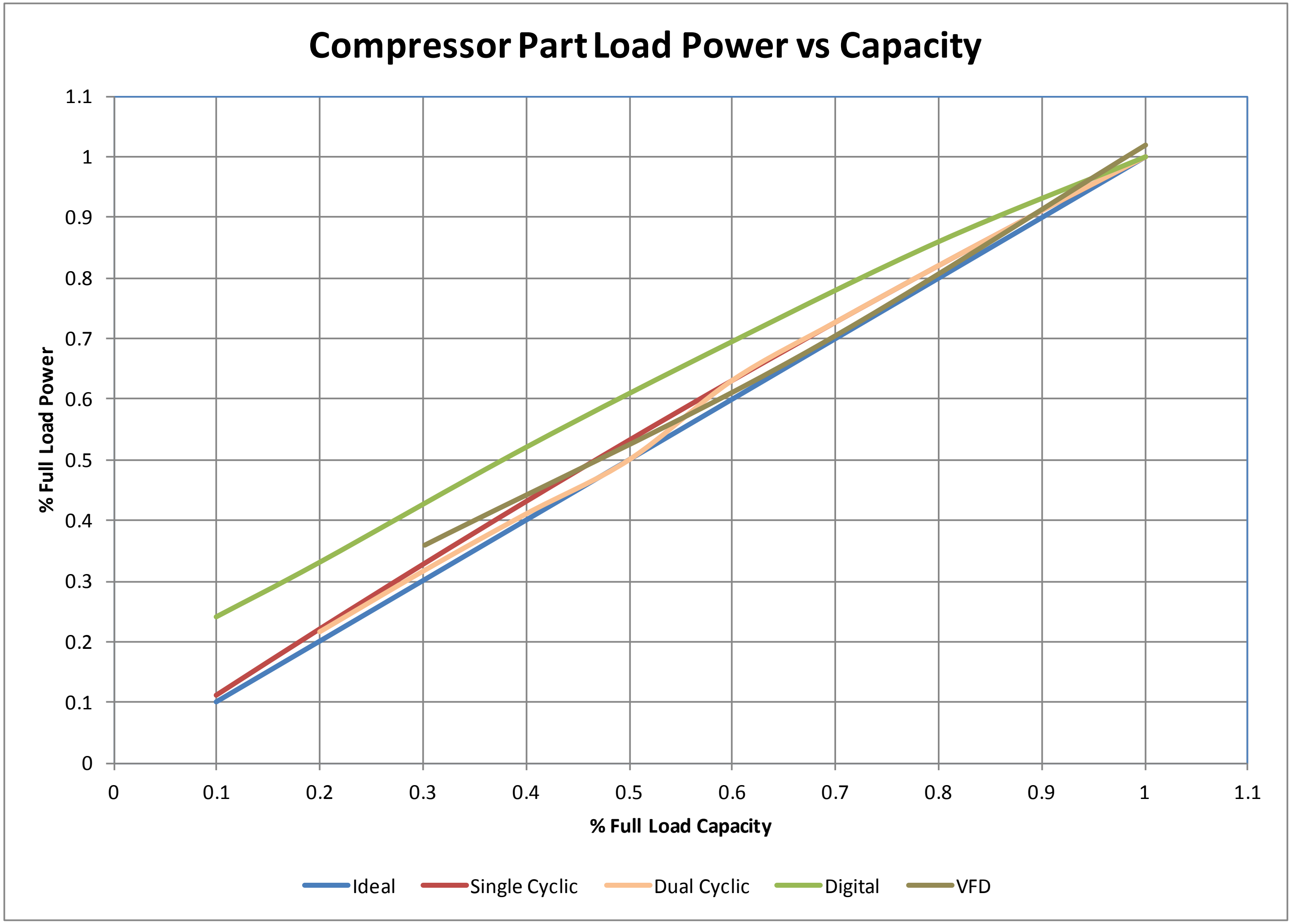


This curve is saying that at part loads the compressor power is constant which is wrong. A single compressor will cycle and the power will be the integrated sum of the on-off power plus a degradation for startup

The degradation coefficients are well defined and are test derived for residential systems or there is a conservative defaults can be used and is defined in AHRI 210/240 and AHRI 340/360 and is used in SEER and IEER ratings.

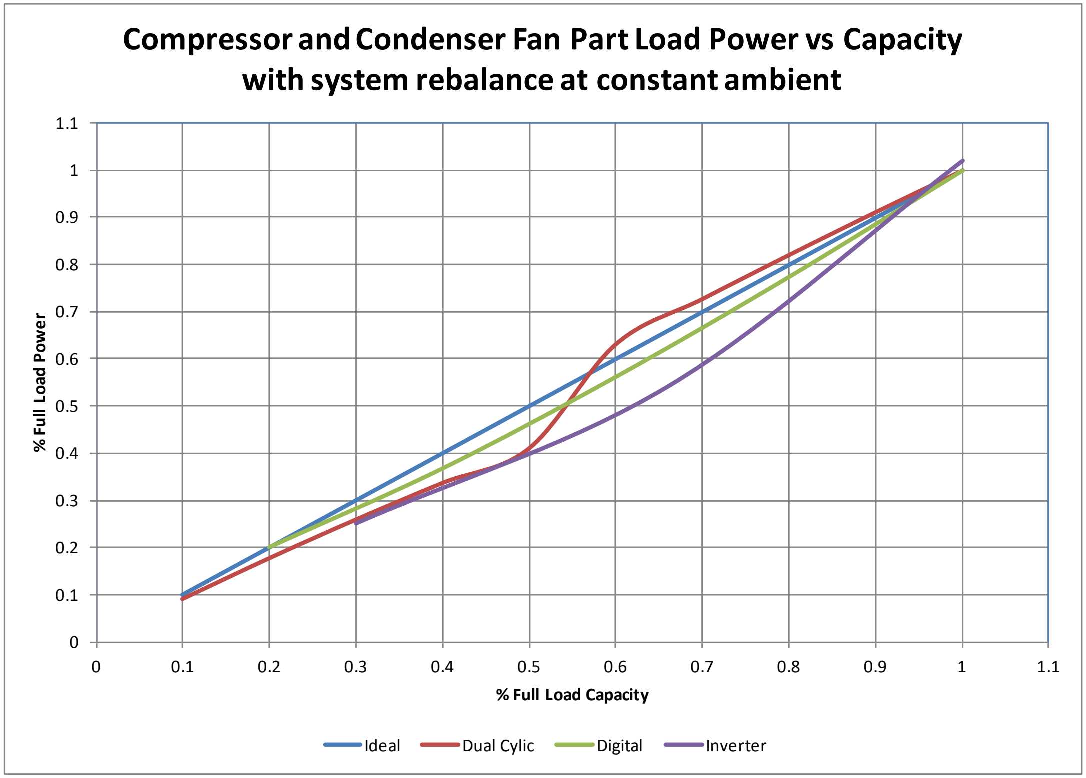
Real Compressor Efficiency Curve

This is a plot of the % full load compressor power vs the full load % capacity at a constant saturated suction and saturated discharge



Alternate Compressor Plot

- The prior plot is misleading as it is only a compressor plot and does not factor in the rebalance of the heat exchangers as they unload in a real system.
- I created a plot of various compressor options to show what a real system impact would be.
- This is a plot of compressor and condenser fan power at a fixed ambient and return air condition



The huge performance improvement is not there and dual compressors on a single circuit can very close with a much lower complexity and cost than variable speed and better in performance than Digital

This is actual confirmed in that most who use the digital have to limit the capacity unloading to get a good SEER

Also note that the variable speed can only get done to 30% and the requirements is 20% which is more like 15% displacement

Study Proposal Assumptions & Claims

Variable Capacity Compressors

- In the justification report it indicates that several manufacturers have products below 65K Btu/h capacity that are variable capacity and this is correct, but they are very high tier units with many high end features are very expensive
- But the proposal is for 65K and above and currently there is only 1 manufacturer who has a high tier products that was just introduced this month.
- Study claims that compressors are available but this is not totally correct
  - Copeland has the digital compressor thru 10 tons which as you saw is not very efficient at part load, and they only have variable speed compressors less than 5 tons
  - Danfoss has new variable speed compressors, but in this capacity range only have a minimum capacity of 33% and lose some efficiency at full load due to the inverter and over speeding of the compressor to insure oil pressure at low speed.
  - Combinations of variable and fixed capacity compressors could be used similar to VRF systems, but the cost estimates providing by AHRI and the industry were based on the use of a digital so the estimated costs would increase
- Only the digital compressor can get to 20% capacity which due to rebalance is more like 15% displacement. The current variable speed compressors are limited to around 40% actual capacity at economizer conditions unless multiple compressors are used
- Variable compressor technology is limited and likely could not support a full insertion in all products plus would take several years to develop and integrate into products

Study Proposal Assumptions & Claims

Integrated Economizers

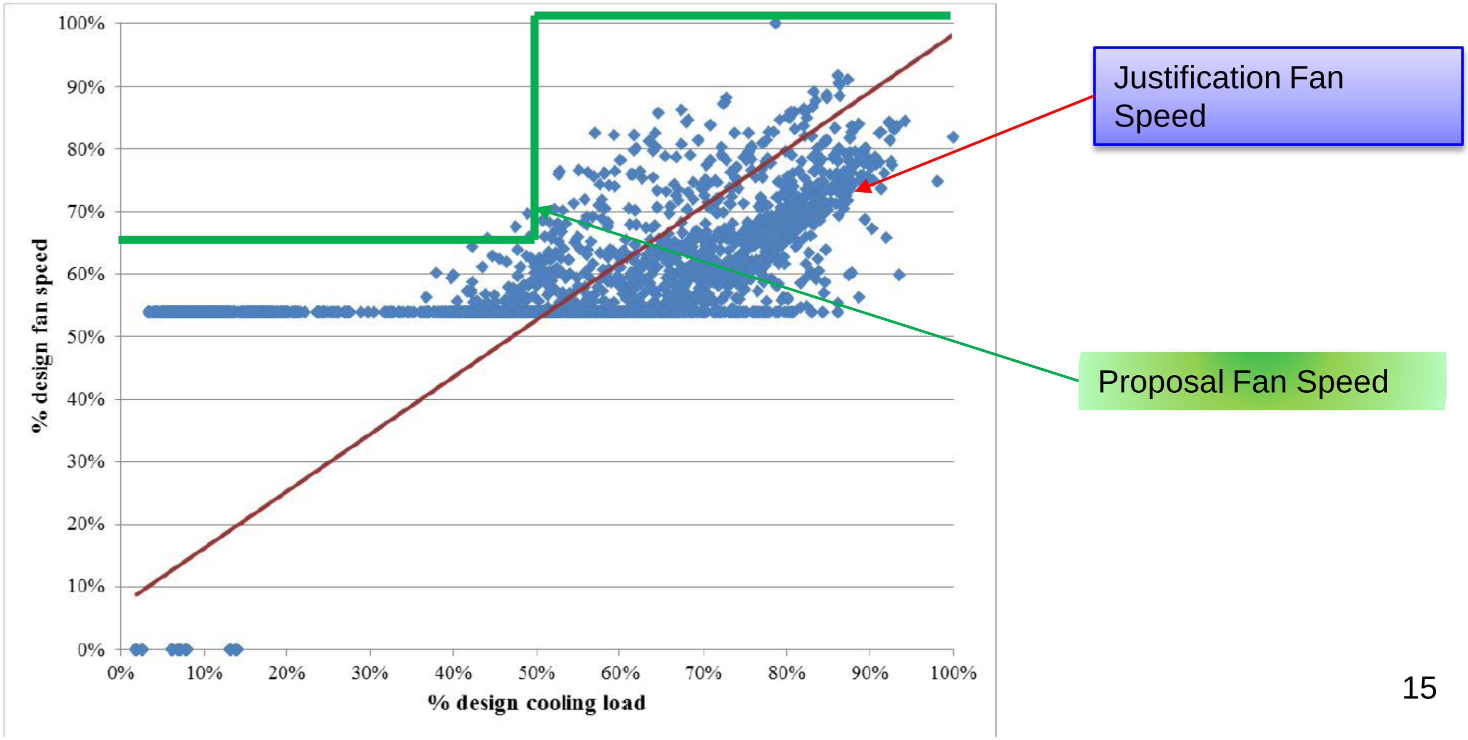
- The proposal justifications claims that the addendum CY economizer proposal assumed full integrated economizers.
- This is not correct and the economizers were de-rated when the supply air temperature went below 55 F which is a conservative estimate.
- For the CMP analysis is was assumed that the integrated cycle would result in the loss of all integrated economizer energy savings which is grossly overstated
- For the Title 24 analysis the it was assumed that the loss due to integrated would be 75% of the delta between full integrated and no economizer. This results in the removal of all integrated savings as well as de-rated economizer operation with no compressors.

Study Proposal Assumptions & Claims

Fan Speed Control

- The proposal requires for DX products a fan speed of 66% below a load of 50%
- The justification document assumed Variable speed fans starting at 100% load down to a speed of 50% at 50% load
- **Savings are overstated!**

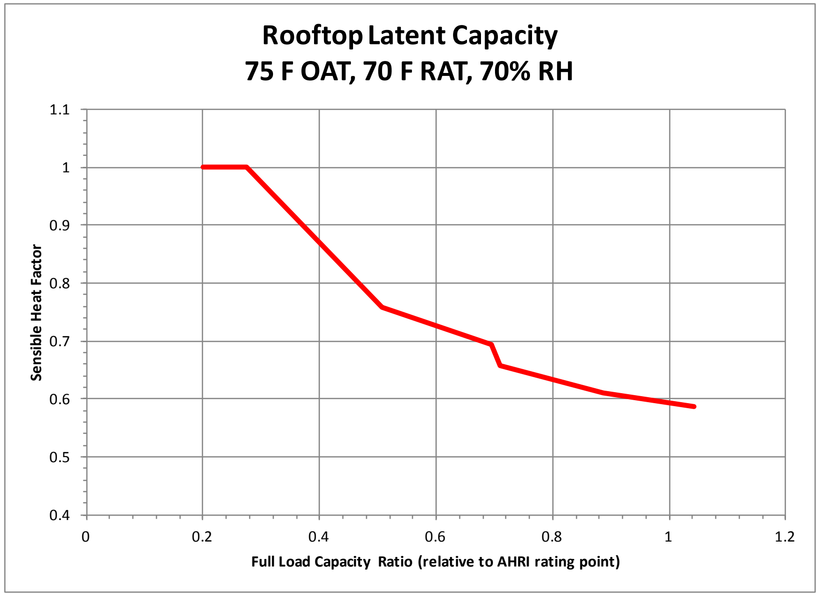
Figure 5. Combined Measures: Fan Control and Integrated Economizer – Single Zone DX:
% design fan speed versus % design cooling load for a representative zone



Study Proposal Assumptions & Claims

Humidity Control

- Study claims that better humidity control will be obtained with the variable capacity and variable speed fans.
- Variable speed fans will help part load humidity control during non-integrated low load operation, but during economizer operation the fan is at high speed to get full benefit of the economizer
- The variable capacity compressor will actual decrease humidity control as shown in the plot of operation at economizer integrated conditions



Due to the rebalance of the heat exchangers the saturated suction rises and the latent capability of the DX coils at a constant CFM decreases and below about 30% capacity the coils is only providing sensible cooling

Study Proposal Assumptions & Claims

- **Energy Analysis**
  - The study only looked at zones 2a, 2b, 3a, 4a, 5a, 5b and did not always use the ASHRAE standard work benchmark cities
  - For the ASHRAE 90.1 analysis it assumed a complete loss of all integrated economizer benefits by using 60 F dry bulb changeover as the base and differential drybulb for the proposal
  - Differential drybulb was used in a zones but is not allowed in current 90.1 high limits and in the proposal Taylor CMP
  - Study used a product with a 9.7 SEER which is far below the current 90.1 requirement of 13 SEER.
  - Also the SEER rated model likely was for a single stage product below 65 KBtu/hr capacity and the proposal is for products >65K
  - It was likely the default model for the DX product was used which is a residential single stage products and it does not properly model variable capacity or even 2 stage capacity
  - Unit was modeled as a VAV that throttles down to 50% fan speed which is not the proposal, but is a limit of Equest, DOE2 and EnergyPlus
  - Model was based on 2.5 inch total static which is about 1.3 inch external which is the high end of the application range for these products. Some units are applied down at more like .5 inches for concentric ducts. AHRI rating static is 0.35 to 0.40 inch external static. This makes the benefits of variable speed higher.

Study Proposal Assumptions & Claims

Cost Assumptions

- AHRI did provide data on costs as shown in the chart but some of the claims are not correct.
- These are not current product costs, and are projections based around the likely use of a digital scroll for variable capacity and assuming high volume national based volumes
- It was claimed that likely these products will drop, which is not likely to happen due to the price of copper, steel and rare earth magnets used in variable speed motors
- It claims that ECM motors can be used but the HP limits of these motors are around 1 to 2 HP and can not be used on the larger products
- Study claims that the AHRI cost include installation, but they were only the incremental product price from a distributor.

ASHRAE 90.1 65K to 110K Evaluation Consumer Price Information

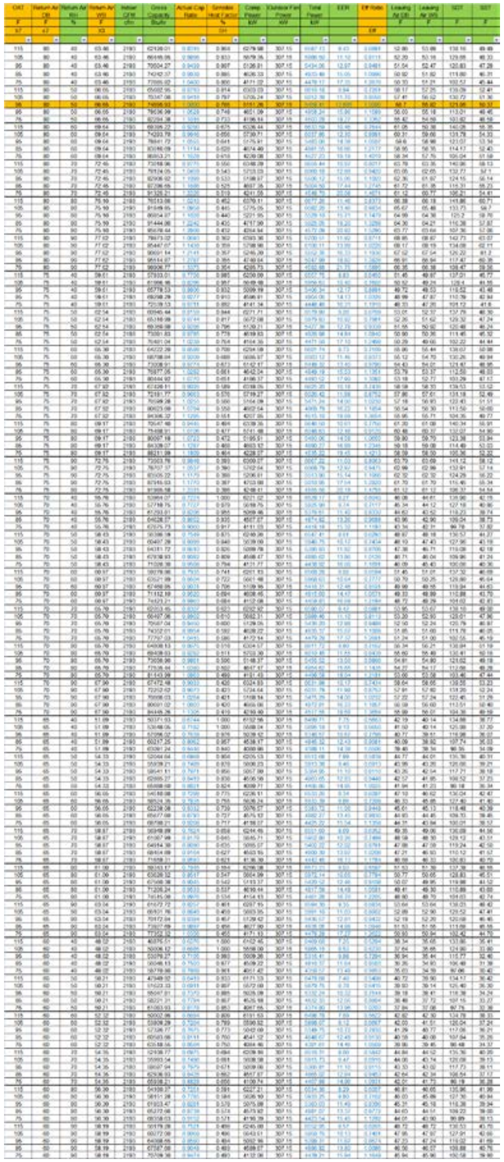
Option	Description	Incremental Cost			
		6 ton	7 ton	8 ton	10 ton
1	Single Stage Compression with 2 speed fan	$ 263.50	$ 214.00	$ 276.50	$ 270.67
2	2 Stage Compression with 2 speed fan	$ 496.00	$ 556.00	$ 655.67	$ 722.00
3	Variable capacity compressor, 2 speed fan	$ 1,190.33	$ 1,306.00	$ 1,484.33	$ 1,663.33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

Energy Analysis

- There are many issues with trying to model this in DOE2, Equest or EnergyPlus as noted in the justification report
- We have also found that the current modeling methods used in the building simulation programs do not properly model variable capacity and variable cfm products and are primarily based on full load single stage DX units that cycle at part load (Old Style Residential Equipment)
- To analysis this we created an expanded model of a typical 6 ton unit that meets the 2010 Efficiency requirements for EER and IEER. The product has an 11.0 EER and 11.2 IEER at AHRI rating conditions
- We used the building model output from the EnergyPlus models for the 5,000 ft2 small office for the 2004 ASHRAE code and then normalized it allow for analysis of the 6 ton unit.
- This was then post processed thru a large spreadsheet tool with Visual Basic models of the compressors, economizers and models of the psychometric properties
- This allows us to look at the details of the operation at each hour of operation
- We including models to simulation lower leaving air temperatures during integrated economizer operation
- Cyclic performance was degraded using the default cyclic coefficients from the AHRI340/360 standard which we know are conservative. When we test for them they are typically better

Typical Model Data



Outdoor Ambient – **70 F to 115 F** (head pressure control below 70 F)
Return Air Dry bulb – **60 F to 80 F**
Return Air Relative Humidity – **40 to 80%**

I also run this for each **stage** of capacity and each indoor **operating cfm**

Capacity is determine for each stage and cfm as a function of OAT, RAT, RWB.

Sensible Heat Factor is also determined at each stage and cfm as a function of RAT, RWB

Efficiency which does not include the indoor fan power is function of OAT, RAT, RWB for each stage and cfm combination

I use a separate model for indoor fan power and assumed 1 inch external static which is about 2.2 in total static for this unit

I also included a modulating exhaust fan as many units have exhaust fans with economizers

Industry Modeling Results

- The model was run for all 17 climate zones using the 5,000 ft2 office normalized hourly data and the benchmark cities
- We also ran the indoor fan as defined in the proposal where the fan is at high at loads above 50% and 2/3 speed at loads below 50%.
- We also assumed the fan would be on high speed during economizer operation
- Because we can get into the details for each hour of operation we were able to separate the 2 speed fan benefits from the variable capacity and integrated economizer benefits

Integrated Economizer Analysis

- In the CMP proposal the variable capacity and variable fan change benefits were lumped together and the full derate of the economizer was taken between integrated and non integrated
- Using the model that I developed, I separated out the integrated economizer savings result from the variable capacity compressor
- The justification document simulated the integrated economizer benefits by comparing full integrated savings vs non integrated savings which overstates the semi-integrated operation.

Zone	CITY	60 F Drybulb Changeover			Taylor Drybulb Changeover								
		Economizer	Integrated	hrs<55 F LAT	Economizer	Integrated	hrs<55 F LAT	Non-Ideal Incremental Power	Non-Ideal Incremental Power Cost	Incremental First Cost	Payback	Scalar Limit	Justified
		hrs	hrs	hrs	hrs	hrs	hrs	kW-h	$	$	yrs	yrs	
1A	Miami	92	0	0	332	53	53	4	0.36	1637	4491.5	8.86	No
1B	Riyadh	356	0	0	1039	307	86	6	0.58	1637	2801.6	8.86	No
2A	Houston	390	0	0	774	56	56	2	0.21	1637	7918.6	8.86	No
2B	Phoenix	495	0	0	1212	290	64	3	0.31	1637	5294.1	8.86	No
3A	Memphis	651	0	0	1134	106	106	8	0.76	1637	2146.8	8.86	No
3B	El Paso	907	0	0	1660	345	108	26	2.41	1637	680.4	8.86	No
3C	San Francisco	1413	0	0	2638	758	591	425	39.94	1637	41.0	8.86	No
4A	Baltimore	760	0	0	1194	131	131	18	1.66	1637	983.4	8.86	No
4B	Albuquerque	1259	0	0	1943	362	155	47	4.43	1637	369.5	8.86	No
4C	Salem	959	0	0	1652	404	273	102	9.55	1637	171.3	8.86	No
5A	Chicago	627	0	0	1001	109	109	20	1.92	1637	853.4	8.86	No
5B	Boise	1087	0	0	1622	345	192	42	3.97	1637	411.8	8.86	No
5C	Vancouver	1123	20	20	1811	525	491	1052	98.82	1637	16.6	8.86	No
6A	Burlington	693	0	0	1273	331	329	132	12.37	1637	132.4	8.86	No
6B	Helena	1060	0	0	1683	417	198	86	8.06	1637	203.0	8.86	No
7	Duluth	1006	0	0	1460	252	210	112	10.56	1637	155.0	8.86	No
8	Fairbanks	953	18	18	1390	379	272	314	29.46	1637	55.6	8.86	No

Non-Integrated Base Case

Semi-Integrated Results

As you can see the Variable capacity change by itself does not meet the Scalar limit for a 15 year design life

2 Speed Fan Benefit Analysis

- In the following two charts I show the metrics for a single speed, 2 stage cooling unit using the Taylor CMP dry bulb changeover temperatures

Taylor Drybulb Single Speed

Zone	CITY	Operating hours			Building	Energy Use				Economizer					
		Cooling	Mechanical	Economizer	Load	Total Power	Cost	Indoor Fan	Exh Fan	Ton-hrs	Non Integrated hrs>28 Btu/lb	Integrated	hrs<55 F LAT	hrs<50 LAT	hrs <45 LAT
		hrs	hrs	hrs	ton-hrs	kw-h	$	kw-h	kw-h	Ton-hrs	hrs	hrs	hrs	hrs	hrs
1A	Miami	3226	2911	332	8675	14734	1383.49	6991	86	298	36	53	53	13	0
1B	Riyadh	3434	2543	1039	7850	20300	1906.16	9845	476	1157	2	307	86	0	0
2A	Houston	2834	2074	774	6028	11670	1095.84	6142	122	609	117	56	56	21	0
2B	Phoenix	3134	2053	1212	6046	16394	1539.36	8985	479	1219	2	291	65	0	0
3A	Memphis	2654	1556	1134	4853	10170	954.92	5752	171	938	60	106	106	40	0
3B	El Paso	3031	1549	1660	5409	14548	1366.07	8690	550	1763	22	345	108	1	0
3C	San Francisco	2711	535	2638	3144	10354	972.27	7773	1113	3225	0	762	595	179	0
4A	Baltimore	2278	1136	1194	3693	8178	767.87	4937	181	1008	39	131	131	66	1
4B	Albuquerque	2881	1138	1943	4456	12739	1196.21	8260	547	2142	2	362	155	14	0
4C	Salem	2023	604	1652	2665	8197	769.66	5800	609	1904	2	405	274	89	0
5A	Chicago	1980	1028	1001	3260	7127	669.22	4291	160	852	27	109	109	44	1
5B	Boise	2235	794	1622	3180	9565	898.20	6408	484	1791	0	345	192	25	0
5C	Vancouver	1881	345	1811	2574	7224	678.30	5393	747	2477	6	525	491	229	12
6A	Burlington	1834	737	1273	2604	6181	580.41	3975	351	1139	32	331	329	136	3
6B	Helena	2008	575	1683	2717	8328	782.00	5757	596	1879	0	417	198	46	0
7	Duluth	1737	412	1460	2117	6629	622.47	4980	419	1803	15	252	210	62	0
8	Fairbanks	1444	296	1390	1973	5631	528.79	4140	531	1785	0	379	272	113	7

Taylor Drybulb 2 speed

Zone	CITY	Operating hours			Building	Energy Use				Economizer						2 Speed Energy Savings		
		Cooling	Mechanical	Economizer	Load	Total Power	Cost	Indoor Fan	Exh Fan	Ton-hrs	Non Integrated hrs>28 Btu/lb	Integrated	hrs<55 F LAT	hrs<50 LAT	hrs <45 LAT	Total Power	Total Cost	Indoor Fan Power
		hrs	hrs	hrs	ton-hrs	kw-h	$	kw-h	kw-h	Ton-hrs	hrs	hrs	hrs	hrs	hrs	%	%	%
1A	Miami	3226	2911	332	8675	9374	880.22	2538	86	298	36	53	53	13	0	-36.4	-36.4	-63.7
1B	Riyadh	3434	2543	1039	7850	13984	1313.11	4970	476	1157	2	307	86	0	0	-31.1	-31.1	-49.5
2A	Houston	2834	2074	774	6028	7849	737.00	2972	122	609	117	56	56	21	0	-32.7	-32.7	-51.6
2B	Phoenix	3134	2053	1212	6046	11454	1075.48	5073	479	1219	2	291	65	0	0	-30.1	-30.1	-43.5
3A	Memphis	2654	1540	1134	4853	5425	509.38	1668	29	480	70	69	69	25	0	-46.7	-46.7	-71.0
3B	El Paso	3031	1549	1660	5409	11065	1039.02	5899	550	1763	22	345	108	1	0	-23.9	-23.9	-32.1
3C	San Francisco	2711	535	2638	3144	10173	955.23	7624	1113	3225	0	762	595	179	0	-1.8	-1.8	-1.9
4A	Baltimore	2278	1136	1194	3693	6163	578.72	3269	181	1008	39	131	131	66	1	-24.6	-24.6	-33.8
4B	Albuquerque	2881	1138	1943	4456	10363	973.06	6351	547	2142	2	362	155	14	0	-18.7	-18.7	-23.1
4C	Salem	2023	604	1652	2665	7276	683.26	5045	609	1904	2	405	274	89	0	-11.2	-11.2	-13.0
5A	Chicago	1980	1028	1001	3260	5309	498.55	2785	160	852	27	109	109	44	1	-25.5	-25.5	-35.1
5B	Boise	2235	794	1622	3180	8015	752.58	5160	484	1791	0	345	192	25	0	-16.2	-16.2	-19.5
5C	Vancouver	1881	345	1811	2574	7053	662.32	5250	747	2477	6	525	491	229	12	-2.4	-2.4	-2.6
6A	Burlington	1834	737	1273	2604	5141	482.78	3111	351	1139	32	331	329	136	3	-16.8	-16.8	-21.7
6B	Helena	2008	575	1683	2717	7508	705.02	5095	596	1879	0	417	198	46	0	-9.8	-9.8	-11.5
7	Duluth	1737	412	1460	2117	5949	558.61	4416	419	1803	15	252	210	62	0	-10.3	-10.3	-11.3
8	Fairbanks	1444	296	1390	1973	5498	516.23	4030	531	1785	0	379	272	113	7	-2.4	-2.4	-2.7

As you can see the 2 speed 66% low speed fan option offers significant energy savings and cost reductions.   This will decrease with operation at lower statics with we should do a sensitivity study on

2 Speed Economic Analysis

- Assuming a 2 speed fan with a lower speed of 66% for compression operation below 50% and 100% during economizer and a 2 stage compression system you get the following economics

Zone	CITY	1 speed	2 speed						
		Total Power	Total Power	Power Savings	Cost Savings	First Cost Increase	Payback	Scalar	Justified
		kw-h	kw-h	kw-h	$	$	yrs	yrs	
1A	Miami	14734	9374	5360	503.27	496	0.99	8.86	Yes
1B	Riyadh	20300	13984	6316	593.05	496	0.84	8.86	Yes
2A	Houston	11670	7849	3822	358.84	496	1.38	8.86	Yes
2B	Phoenix	16394	11454	4940	463.88	496	1.07	8.86	Yes
3A	Memphis	10170	5425	4745	445.54	496	1.11	8.86	Yes
3B	El Paso	14548	11065	3483	327.05	496	1.52	8.86	Yes
3C	San Francisco	10354	10173	181	17.04	496	29.11	8.86	No
4A	Baltimore	8178	6163	2014	189.15	496	2.62	8.86	Yes
4B	Albuquerque	12739	10363	2376	223.15	496	2.22	8.86	Yes
4C	Salem	8197	7276	920	86.40	496	5.74	8.86	Yes
5A	Chicago	7127	5309	1818	170.67	496	2.91	8.86	Yes
5B	Boise	9565	8015	1551	145.61	496	3.41	8.86	Yes
5C	Vancouver	7224	7053	170	15.98	496	31.03	8.86	No
6A	Burlington	6181	5141	1040	97.63	496	5.08	8.86	Yes
6B	Helena	8328	7508	820	76.98	496	6.44	8.86	Yes
7	Duluth	6629	5949	680	63.86	496	7.77	8.86	Yes
8	Fairbanks	5631	5498	134	12.56	496	39.49	8.86	No

- Results show that in many zones it can be easily justified, but in Zones 3C, 5C, and 8 it does not met the scalar limit.
- The reason is that these are very high economizer operating zones and my model assumes the economizer is on high speed during all operation.
- This could be significantly improved by also operating with 2 speed fan operation in economizer mode when the economizer is less than 50-60%

DX Evaluation Conclusions

- Study shows that a variable capacity can not be economically justified.
- Although the technology of variable speed and capacity are advancing it is not a common production option in the 65K and larger capacities
- For constant volume the integrated economizer can be improved with good control logic and the use of a minimum of 2 stages of capacity
- The two speed fan can be justified in all zones assuming that we also require 2 speed fan operation in economizer mode, but this will require some controls development work.
- Products that can meet these requirements are not available and redesign to the units to have two stages as well as economizer controls will be required, which will take 2-3 years to develop at a minimum so an effective date of more 1/1/2015 would likely be something the industry might be able to support
- As this study has just been completed, it needs to be reviewed by the industry to see if they will support it so I would recommend we not push this forward until the January meeting.

Chilled Water Coil Proposal

- For the chilled water coils the CMP proposal is requiring 2 speed fans down to ¼ HP with a lower speed of 50%
- This will save energy and the first cost increase are not high assuming the units have modulating chilled water coils
- But the small fan coils, typically use 2 way on-off valves and only operation at 0 and 100% so they will not have to meet the proposed requirement as written.
- If we elect to go forward with this then an additional requirement for a minimum of 2 stages of chilled water capacity control would be required
- I have not looked into the availability of 2 stage water valves or the cost premium for modulating, but I suspect the modulating will be very expensive relative to these small fan coil costs
- We also need to check with the manufacturers of these products and get their feedback on the options for at a minimum 2 stage water control valves.

Proposal ASHRAE 90.1 Changes

6.4.3.10 ~~Single Zone Variable-Air-Volume~~ Fan Controls. HVAC systems shall have variable airflow controls as follows:

a. Air-handling and fan-coil units with chilled-water cooling coils and supply fans with motors greater than or equal to ~~5~~ 1/4 hp shall have their supply fans controlled by two-speed motors or variable-speed drives. At cooling demands less than or equal to 50%, the supply fan controls shall be able to reduce the airflow to no greater than the larger of the following:

One half of the full fan speed, or
The volume of outdoor air required to meet the venti-lation requirements of Standard 62.1.

b. ~~Effective January 1, 2012, all~~ air-conditioning equipment and air-handling units with direct expansion cooling and a cooling capacity at AHRI conditions greater than or equal to ~~110,000~~ 65,000 Btu/h ~~that serve single zones~~ shall have their supply fans controlled by two-speed motors or variable-speed drives. At cooling demands less than or equal to 50%, the supply fan controls shall be able to reduce the airflow to no greater than the larger of the following:

Two-thirds of the full fan speed, or
The volume of outdoor air required to meet the venti-lation requirements of Standard 62.1.

6.5.1.3 Integrated Economizer Control. Economizer systems shall be integrated with the mechanical cooling system and be 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 at AHRI conditions ≥ 65,000 Btu/hr 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.

Alternate Proposal

6.4.3.10 ~~Single Zone Variable-Air-Volume~~ Indoor Fan Controls. HVAC systems shall have variable airflow controls as follows:

a. Air-handling and fan-coil units with chilled-water cooling coils and supply fans with motors greater than or equal to ~~5~~ 1/4 hp shall have their supply fans controlled by two-speed motors or variable-speed drives. At cooling demands less than or equal to 50% for proportionally controlled units and for 2 stage control units operating on the first stage, the supply fan controls shall be able to reduce the airflow to no greater than the larger of the following:

One half of the full fan speed, or

The volume of outdoor air required to meet the venti-lation requirements of Standard 62.1.

Constant volume units shall also have a minimum of 2 stages of capacity and shall be capable of operating the economizer, if required, with 2 stage fan speed control with operation at low speed when the economizer capacity is less than 50%. Variable air volume units shall have a minimum of 4 stages with a minimum stage of 25% or less.

The requirements for ¼ to 5 HP will be effective 1/1/2015 and the requirements for greater than 5 HP will be effective immediately

Alternate Proposal

b. ~~Effective January 1, 2012~~, all air-conditioning equipment and air-handling units with direct expansion cooling and a cooling capacity at AHRI conditions greater than or equal to ~~110,000~~ 65,000 Btu/h ~~that serve single zones~~ shall have their supply fans controlled by two-speed motors or variable-speed drives. At cooling demands less than or equal to 50% for proportionally controlled units and for 2 staged controlled units operating on first stage, the supply fan controls shall be able to reduce the airflow to no greater than the larger of the following:

Two-thirds of the full fan speed, or

The volume of outdoor air required to meet the venti-lation requirements of Standard 62.1.

Constant Volume units shall also have a minimum of 2 stages of capacity and shall be capable of operating the economizer if required with 2 stage fan speed control with operation at low speed when the economizer capacity is less than 60%. Variable air volume units shall have a minimum of 4 stages of capacity with a minimum stage of 25% or less.

The requirements for 65,000 to 110,000 Btu/hr capacity are effective 1/1/2015 and greater than 110,000 Btu/hr are effective immediately

6.5.1.3 Integrated Economizer Control. Economizer systems shall be integrated with the mechanical cooling system and be capable of providing partial cooling even when additional mechanical cooling is required to meet the remainder of the cooling load. The mechanical capacity control shall be interlocked with the economizer control such that the economizer does not begin to close until the unit leaving air temperature is less than 45 F. All units with an economizer must have a minimum of 2 stages of mechanical cooling for constant volume units and 4 stages with a minimum of 25% for variable air volume effective 1/1/2015.

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AHRI Air-Conditioning, Heating, and Refrigeration Institute

July 12, 2011

California Energy Commission (CEC)

Re: AHRI Comments on December 9, 2010 Single Zone VAV Presentation at the Nonresidential HVAC Stakeholder Meeting #2 and April 21, 2011 CASE Report on Fan Control and Integrated Economizers (Docket Number 10-BSTD-01; April 27, 2011 Staff Workshop – 2013 Building Energy Efficiency Standards")

Dear CEC Staff:

The Air-Conditioning, Heating and Refrigeration Institute (AHRI) is the trade association representing manufacturers of heating, cooling, water heating, and commercial refrigeration equipment. Over 300 members strong, AHRI is an internationally recognized advocate for the industry, and develops standards for and certifies the performance of many of the products manufactured by our members. In North America, the annual output of the HVACR industry is worth more than $20 billion. In the United States alone, our members employ approximately 130,000 people, and support some 800,000 dealers, contractors, and technicians.

We have developed some comments with respect to the single zone VAV presentation given at the nonresidential HVAC stakeholder meeting #2 on December 9, 2010, and the CASE report on fan control and integrated economizers which was discussed at the CEC staff workshop April 27, 2011.

<u>AHRI Comments on the Single Zone VAV – Nonresidential HVAC Stakeholder Meeting #2 Presentation:</u>

1. The mechanical cooling requirements stated on slide 8 would require significant product modifications to non-residential rooftop units and split systems manufactured by the industry. Such requirements would have adverse impact on product planning and development. For example, most two speed single compressors are a 100% -66% split. Requiring mechanical cooling to modulate in increments of 50% would compel manufacturers to use digital (proprietary technology) or variable speed single compressor. Currently, the variable speed compressor technology is under development. There are very few sizes and voltages available at the moment.

   The analysis requiring mechanical cooling to modulate in increments of 20% does not seem to be accurate. Modeling has proven that operating this low with certain technologies results in much higher energy consumption, as compared to cycling.

2. The code change proposal on fan control (slide 7) is too stringent for discrete two-speed motors. We recommend power limitation be removed and that only the fan speed requirement corresponding to ASHRAE 90.1-2010 be specified.

   The code change proposal with respect to fan control on slide 7 should be modified as follows:

   Fan Control. Each unitary air conditioner and air-handling unit with mechanical cooling capacity listed in Table X shall be designed to vary the airflow rate as a function of actual load and shall have controls and/or devices (such as two-speed or variable speed control) that will result in fan motor demand of no more than ~~50 percent of design wattage at~~ 66 percent of design fan speed.

3. Slide 14 shows a saving of $2,880 during mass production. We do not believe that this proposed cost saving for true single zone VAV equipment due to mass production is an accurate value since the volumetric increase of the equipment would only apply to equipment sales in California. We believe that the cost savings should be a more conservative value, thereby increasing the average incremental cost which directly impacts the payback periods.

4. We have several concerns about the model that was used to show the savings and justify the code change proposal.

   Firstly, the damper position bin chart on slide 26 lays the foundation that only 75% of the economizer savings are currently being utilized. It appears from the damper position bin chart on slide 26 that the damper position and fresh air percentage entering the unit were assumed to be the same. This assumption is invalid because the damper position and fresh air percentage varies from one unit to another. The invalid assumption significantly impacts the economizer savings calculation. For example, if it is assumed that the damper position and percentage of fresh air entering the unit were the same, the calculations in Table 1 lead to an economizer savings of close to 75%.

Table 1

Percent of time at Damper Position	**Damper Position**	**Percent of Fresh Air**	**Percent of Economizer Savings**
47.00%	91-100%	95.00%	44.65%
5.00%	81-90%	85.00%	4.25%
10.00%	71-80%	75.00%	7.50%
13.00%	61-70%	65.00%	8.45%
12.50%	51-60%	55.00%	6.88%
7.50%	41-50%	45.00%	3.38%
4.00%	31-40%	35.00%	1.40%
0.00%	21-30%	25.00%	0.00%
1.00%	11-20%	15.00%	0.15%
100.0%			76.65%

Testing on actual units shows that relationship between damper position and fresh air percentage is not linear and actually varies from unit to unit depending on several factors, including static pressure in the return duct and the velocity of the fresh air coming through the outside air dampers. Table 2 recalculates the economizer savings using the actual amount of fresh entering the unit from one manufacturer at these different damper positions, and it changes the baseline from 75% to 87%, thereby cutting the savings opportunity almost in half.

Table 2

Percent of time at Damper Position	Damper Position	Percent of Fresh Air	Percent of Economizer Savings
47.00%	91-100%	100.00%	47.00%
5.00%	81-90%	100.00%	5.00%
10.00%	71-80%	95.00%	9.50%
13.00%	61-70%	84.00%	10.92%
12.50%	51-60%	69.50%	8.69%
7.50%	41-50%	55.00%	4.13%
4.00%	31-40%	41.00%	1.64%
0.00%	21-30%	32.00%	0.00%
1.00%	11-20%	12.00%	0.12%
100.0%			86.99%

<u>AHRI Comments on the April 21, 2011 CASE Report on Fan Control and Integrated Economizers:</u>

1. The proposed language in 144 (e) 2.B. (page 53) would require that every product above 5 tons have either five compressors, or mandate the use of a variable speed or digital compressor on the first stage compressor for prescriptive applications. As mentioned earlier, the variable speed compressor technology is currently in the development stages. There are very few sizes and voltages available at the moment.

2. In reviewing the CASE report, it appears this data was taken on a large unit with six compressors but only four stages of cooling. The data was taken over a two day period only. Our concern is that the data does not necessarily represent what happens in all units, especially because this unit is a multiple zone VAV unit, with a unique control system and unique refrigeration system. Since this proposal is for multiple zone and single zone VAV units, bin charts would need to be developed for both types of units. We also believe that data from a statistically significant number of different units would need to be gathered to conclude potential savings. These units would also need to have a different number of compressor stages in them since two, three or four compressor stages can be used, depending on the manufacturer and the capacity of the unit.

3. We also believe there are significant differences in the way units can be designed for multiple zone VAV applications and single zone VAV applications. The most significant difference is the way the evaporator coil can be designed when there is more than one compressor. For multiple compressors, the evaporator can be designed with face-split, row-split or intertwined circuits. For true multiple zone VAV units, the evaporators must be row-split or intertwined in order to work at airflow rates as low as 20-25% of full air flow. Multiple zone VAV units also are designed to maintain the supply air at a given setpoint.

   A single zone VAV system typically needs to work down to approximately 60% of the airflow, so that face-split evaporator coil designs still work well in the applications. These applications are typically controlled by a room thermostat, so the only reason the economizer dampers would start to close when the outside air is suitable is when the supply temperature goes below a given setpoint for comfort reasons, which is typically around 55 ^{o}F. However, the supply temperature could also be reset to 50 ^{o}F and not cause any problems in the way the unit operates, and would not present any comfort issues for most applications since such a condition usually exists for only a couple of minutes.

   Chart A shows what the supply temperature would typically be if one stage of mechanical cooling is running in a two stage compressor unit, and the economizer is fully open at different outdoor temperatures with 50-60% relative humidity in the air.

Chart A

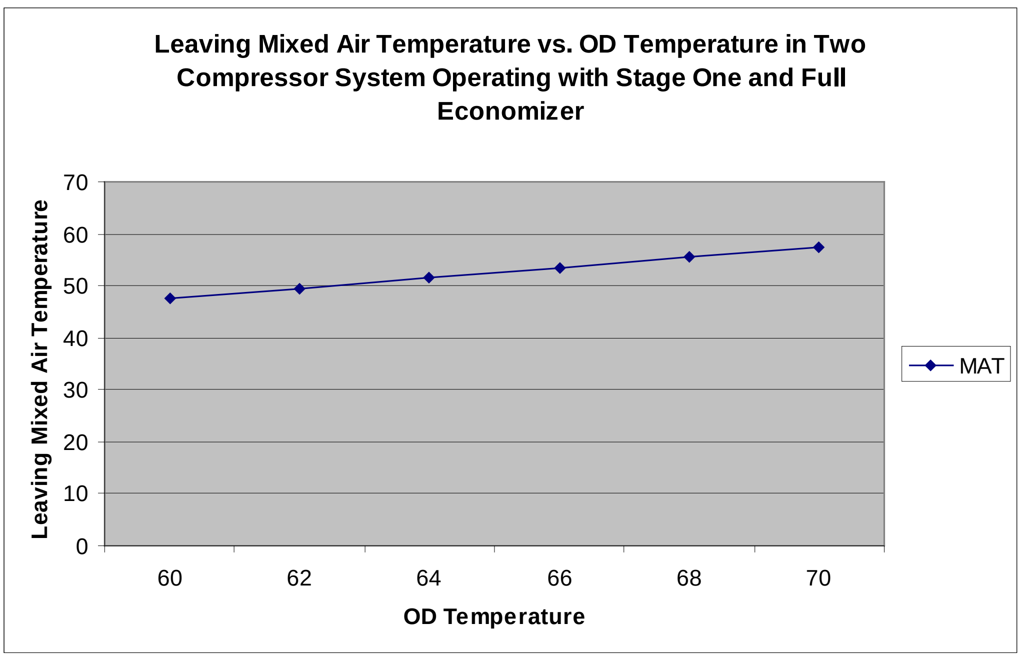


   Chart A suggests that the supply temperature could get below 50 ^{o}F when the outside air temperature is 60 ^{o}F. But there should be very few instances in typical applications, especially in the dry climate of California, when mechanical cooling is needed at an outside air temperature of 60 ^{o}F. We believe that most typical applications will not need mechanical cooling until the outside air gets closer to 65 ^{o}F. In this case, the economizer dampers do not need to close very often in order to maintain the supply air temperature above 55 ^{o}F.

Chart B shows the supply temperature versus outdoor temperature for three-compressor units. The supply temperature for a three-compressor system only gets down to about 52 ^{o}F at an outdoor air temperature of 60 ^{o}F. There should be very few instances where the economizer dampers would close, even if the setpoint for closing the economizer dampers is 55 ^{o}F.

Chart B

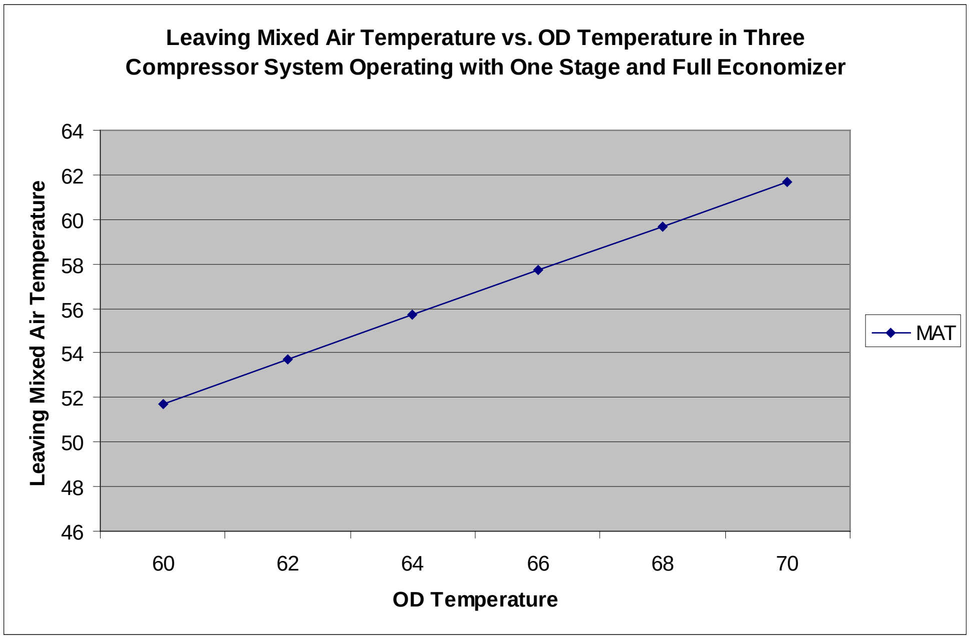
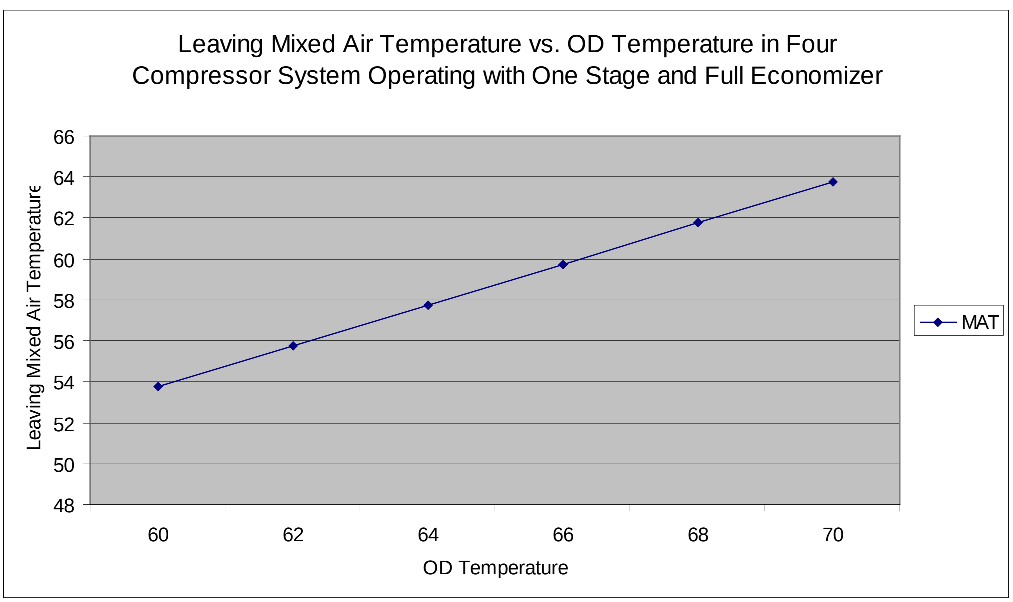


Chart C shows the supply temperature versus outdoor temperature for four-compressor units with face-split evaporator coils. At an outdoor air temperature of 60 ^{o}F, the supply is close to 54 ^{o}F. Therefore, the economizer dampers should seldom close when the outside air is 60 ^{o}F and suitable and the mechanical cooling comes on.

Chart C



Additionally, a single zone VAV unit that runs the airflow down to 60% of the total airflow will inherently bring in significantly more fresh air than a typical multiple zone VAV system which can operate down to 20-25% of the total airflow. This will allow the economizer of the single zone VAV unit to satisfy the load much longer before mechanical cooling will be needed, as compared to a typical multiple zone VAV system.

A unit with a row-split or intertwined evaporator coil applied in a single zone VAV application with a minimal airflow which equates to 60% of the total airflow will provide supply temperatures that are 2-3 ^{o}F cooler than units with face-split coils at the same airflow when one compressor is operating and is controlled by a room thermostat. This is because more of the coil is active at part-load in row-split or intertwined coil circuits, thereby resulting in more total capacity and more sensible capacity. The supply air temperatures will be significantly higher than a typical multiple zone VAV application where the airflow can run much lower.

Due to the significant differences in the way units can be designed and operated in multiple zone VAV applications versus single zone VAV applications, we believe that the potential economizer savings is significantly lower in single zone VAV units than in multiple zone VAV units. In the case of multiple zone VAV units, we believe that the savings is only half of what was presented during the December 9, 2010 stakeholder meeting since the amount of fresh air used in the damper position bin calculations appears to be incorrect. We believe that single zone VAV applications only need to have a maximum compressor capacity of 50% when the economizer air is suitable and mechanical cooling is needed. Since most units above 20 tons have three or more compressors in them, we would also have no problem if the maximum compressor capacity was 33% for units greater than 20 tons.

Although we understand that a multiple zone VAV should have a lower maximum compressor capacity than a single zone VAV, we believe that it does not need to be as low as 20% based on our concerns with respect to the economizer savings analysis. We believe that 25% is a more reasonable number.

AHRI believes that both the December 9, 2010 single zone VAV presentation and the April 21, 2011 CASE report on fan control and integrated economizers have serious flaws and, if implemented would not help CEC achieve its stated objectives. We recommend that CEC reconsider its code change proposals with respect to fan control, integrated economizers and single zone VAV. Although we appreciate the opportunity to provide these comments, we recommend that CEC convene a meeting with our industry in order to discuss and resolve our concerns. If you have any questions or wish to discuss this further, please do not hesitate to call me at (703) 600-0383.

Sincerely,

Aniruddh Roy
Regulatory Engineer

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AHRI Air-Conditioning, Heating, and Refrigeration Institute

March 30, 2011

Mr. Matthew Tyler
Portland Energy Conservation, Inc. (PECI)
1400 SW 5th Avenue, Suite 700
Portland, Oregon 97201

Re: AHRI Comments on Memorandum Issued by PECI on January 4, 2011

Dear Mr. Tyler:

The Air-Conditioning, Heating and Refrigeration Institute (AHRI) is the trade association representing manufacturers of heating, cooling, water heating, and commercial refrigeration equipment. Over 300 members strong, AHRI is an internationally recognized advocate for the industry, and develops standards for and certifies the performance of many of the products manufactured by our members. In North America, the annual output of the HVACR industry is worth more than $20 billion. In the United States alone, our members employ approximately 130,000 people, and support some 800,000 dealers, contractors, and technicians.

We have developed some comments in response to PECI's memorandum on proposed requirements for light commercial unitary HVAC equipment. The memorandum was issued on January 4, 2011 to ASHRAE's Technical Committees 8.11 and 7.5. Our comments are:

- Temperature sensor calibration—temperature sensors are not typically adjustable in the field. Temperature sensor calibration is already performed by the sensor manufacturers, and requiring original equipment manufacturers (OEMs) to re-perform the calibration seems to be a redundant process. Most OEMs are not equipped to calibrate temperature sensors as accurately as the sensor manufacturers are; the redundant task would be an unnecessary burden on OEMs since they will now be required to invest in instrumentation and labor to calibrate the sensors at their own facilities. Temperature sensor calibration is a process that should be done on a sampling basis using statistical process control and accurate measurement devices by the sensor manufacturer. The costs associated with calibration will be much less if done by the sensor manufacturer. Measurement of sensor accuracy in an OEM's production line assembly is not a practical approach.

- Maximum 10 cfm/sf damper leakage at 1.0 in w.g.—we feel that this a reasonable number for leakage since it agrees with ASHRAE 90.1 requirements for zone 3, 4, 5b, and 5c; however, in order to ensure that accurate measurements are recorded, testing should be conducted in a laboratory environment rather than a production assembly.

- Minimum 200,000 full damper cycles—this may be an unreasonable number for design life cycles and it is unclear what the requirement really means. A requirement to test dampers in a life cycle test with some degree of sampling might be an option.

- 5-year performance warranty of economizer assembly—currently, a 5 year warranty is an option that customers can select but the general standard is 1 year. We recommend against performance warranty requirements for an economizer within a unit. The majority of economizer related issues are caused due to misapplication in the field or lack of proper maintenance.

- Direct drive modulating actuator with gear driven interconnections (GDI)—although this is essentially the approach most of the industry has taken, it is a prescriptive design approach. Currently, poor quality GDI dampers and good quality non-GDI dampers exist in the marketplace. We feel that it is better to define the requirements associated with the reliability of the damper drive. Additionally, we feel that it would be more appropriate to set equipment performance goals and allow the manufacturer to determine the best way to attain those goals.

- Integrated economizer control—this is already required by ASHRAE 90.1-2010, so it should not be a problem with the industry.

- Economizer high limit control and deadband—high limit controls are commonly used in the industry, but several types exist. This is a prescriptive design approach, and we maintain that it would be more appropriate to set equipment performance goals and allow the manufacturer to determine the best way to attain those goals. Additionally, most high limit controls have a deadband. It is unclear as to what this requirement is trying to convey and we would like to see some clarification.

- Require economizer on smaller AC along with compliant T-stats (2-stage or electronic)—the 33,000 Btu/h requirements should not be an issue for the light commercial applications since the industry has access to smaller economizers; however, there should be an exemption for residential buildings where economizers are not justified. Requiring factory installed economizers on smaller equipment will drastically increase the overall cost of the equipment.

- Spaces required to have occupancy sensors are required to setback thermostat by 3°F or more or reduce VAV airflow when room is unoccupied—this will require developing new thermostats which are not available in the market today. It is unclear whether this requirement is overriding minimum ventilation requirements for VAV systems.

- FDD (fault detection and diagnostics) prescriptive requirement with signaling capability to t-stat or exterior gateway—it is unclear as to what is required. FDD is a broad subject and can cover many topics. We would like further specifics from PECI on the FDD requirements. This requirement seems to be very expensive, and the resulting energy savings do not justify the overall cost to meet this requirement.

Here are some additional comments on PECI's proposed requirements for light commercial unitary HVAC equipment:

- We feel that the discrepancies noted between the operation of factory installed economizers and field installed economizers is related to installation and maintenance practices. It would be more appropriate to promote quality installation and maintenance practices with regards to the economizer setup rather than require economizers to be factory-installed and comply with reliability certification.

- Overall, the proposed requirements represent a request for a California specific unit design which attempts to work around the current federal efficiency requirements through the incorporation of a factory installed economizer. It would be more appropriate for the State of California to implement building code requirements for economizers installed with units greater than or equal to 36,000 Btu/h.

As such, we urge PECI to postpone any further development on proposed requirements for light commercial unitary HVAC and to immediately convene a stakeholder meeting in order to receive the industry's input. If you have any questions or wish to discuss this further, please do not hesitate to call me at (703) 600-0383.

Sincerely,

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