2111 Wilson Boulevard, Suite 500
Arlington, VA 22201, USA
www.ahrinet.org
PH 703.524.8800
FX 703.528.3816

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

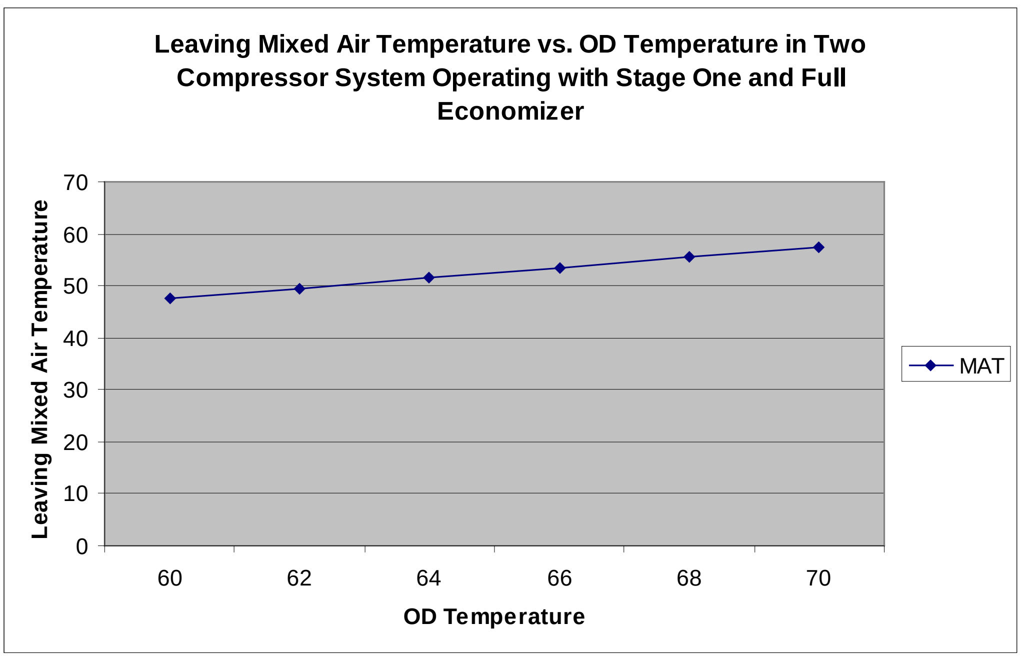


   Chart A suggests that the supply temperature could get below 50 ^{o}F when the outside air temperature is 60 ^{o}F. But there should be very few instances in typical applications, especially in the dry climate of California, when mechanical cooling is needed at an outside air temperature of 60 ^{o}F. 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

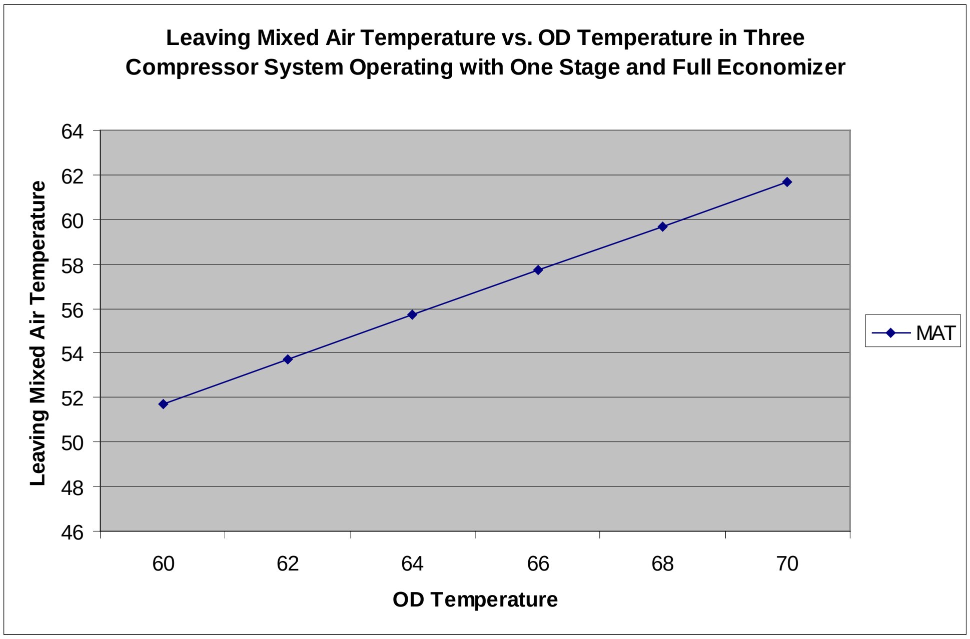
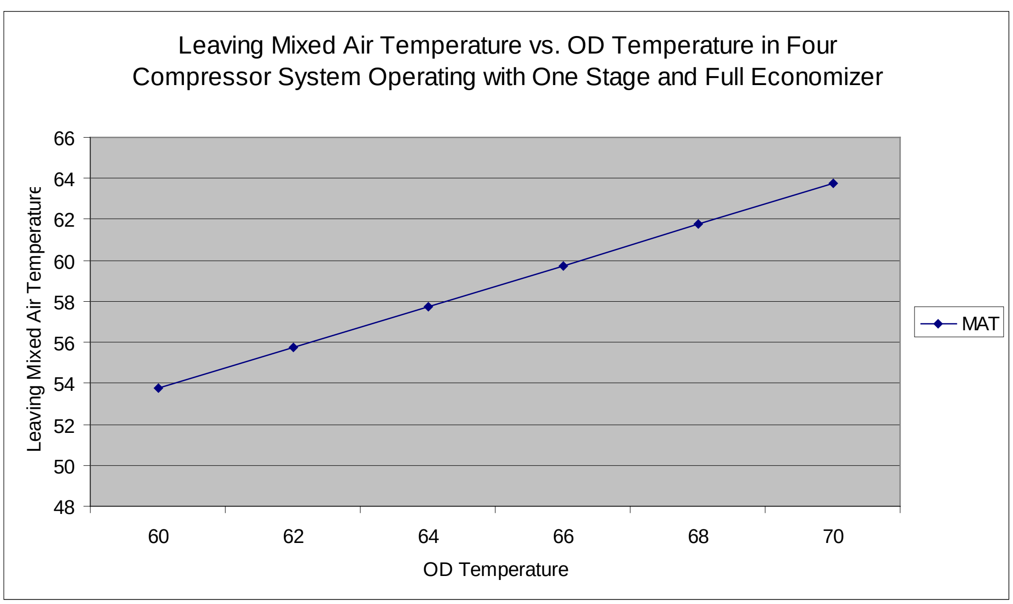


Chart C shows the supply temperature versus outdoor temperature for four-compressor units with face-split evaporator coils. At an outdoor air temperature of 60 ^{o}F, the supply is close to 54 ^{o}F. Therefore, the economizer dampers should seldom close when the outside air is 60 ^{o}F and suitable and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

Air-Conditioning, Heating, and Refrigeration Institute
2111 Wilson Boulevard, Suite 500
Arlington, VA 22201-3001, USA
703-600-0383 Phone
703-562-1942 Fax
aroy@ahrinet.org