Comparison of Measured and Modeled Economizer Operation

Bob Davis
Mike Kennedy
David Baylon

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ECOTOPE
CONSULTING
RESEARCH
DESIGN

4056 9TH AVENUE NE SEATTLE, WA 98105
(206) 322-3753 FAX: (206) 325-7270

Introduction

Over the past several months, Ecotope has provided technical advice and publications to PECI as part of its Air Care Plus Program for packaged rooftop units. Extensive simulations were run as part of this work, with the goal being to estimate the savings that can be attributed to proper economizer function under various control strategies. As these were only simulations, there was an additional verification step included in Ecotope's work that was intended to validate some part of the simulations with detailed field data. Various rounds of monitoring were carried out in late spring and extended into the summer; observations from this monitoring were summarized in an earlier memorandum.

Unfortunately, as our efforts through late spring and into summer have proven, it can be very challenging to verify simulated economizer savings if the verification commonly involves an unfavorable outdoor temperature regime for many commercial buildings. That is, outdoor air is too warm to offer substantial free cooling. Further, if a building is characterized by a shell-dominated cooling load, even as outdoor temperatures moderate, there will be limited opportunities for free cooling because the cooling load itself will moderate.

Despite problems with the monitoring, a number of issues have arisen which we believe are important in refining efforts to diagnose and repair economizer system components. These issues are the results of problems with economizer control settings (changeover settings, differential operation); equipment malfunction (dampers, thermostat logic, sensors); and equipment set-up (minimum outside air and return air bypass). In some cases, we may be repeating earlier concerns, but most of this is new material.

First, though, a review of economizer and compressor cooling theory is presented, and then results from recent monitoring are compared with modeled results in order to point out various factors that can influence cooling system performance.

Economizer Cooling Capacity and Thermostat Interactions

The amount of cooling delivered by an air handler with the compressor operating is relatively constant. It is determined by the airflow across the evaporator, outside air temperature, the minimum outside air fraction, the amount of fan heat, and the return air temperature. Assuming a constant airflow (essentially normalizing by air flow) a convenient measure of the delivered cooling is the temperature difference between the return and supply air flows. This would not work in humid climates but in northwest climates is a good first order approximation.

In a typical small roof top package unit there is a single fixed capacity compressor, and in some slightly larger units there are two compressors. With all cooling stages on, the air handlers are typically sized to provide a temperature difference of 24°F across the coil (the sensible component). As an example: if room air is 74°F, outside air is 84°F, minimum outside air is 20%, and the cooling coil is capable of providing 24°F of cooling, the maximum delivered temperature difference is 22°F. Air handler fan energy adds 1.5°F to the supply air temperature, reducing the maximum possible delivered temperature difference to 20.5°F.

In contrast, the amount of cooling delivered by the economizer is highly variable. It is determined by the air flow, outside air temperature, the maximum outside air fraction, the amount of fan heat, and the return air temperature. Assuming that the total airflow is comparable between various cooling modes, the amount of cooling can again be determined by examining temperature differences in the system. The maximum cooling potential is the difference between outside air and return air temperature, but most economizers operate with some fraction of return air even at "100 percent" outside air. In fact, this percentage can be considerable; the average was about 30% in the EWEB study. This reduces the cooling potential by the return air fraction. To follow the above example, if room air is 74° F, outside air is 64° F, and the maximum outside air fraction is 70%, the maximum possible delivered temperature difference is 7° F. The air handler fan energy adds 1.5° F to the supply air temperature, reducing the maximum possible delivered temperature difference to 5.5° F.

A major factor in occupant comfort related to economizer logic is the how the space thermostat handles the smaller amount of cooling delivered by the economizer under marginal outside temperature conditions. Typical controls use the economizer as stage one cooling and only engage the compressor if conditions change so that the economizer is locked out or the thermostat calls for stage two cooling. As the economizer operates in warmer conditions, there are more hours where it cannot deliver adequate cooling by itself. How and when the thermostat signals for stage two cooling determines the occupant comfort level.

The monitored data (discussed in detail below) show that this system, when run in differential mode, (where the economizer continually compares the enthalpy of the return and outdoor air and selects the most favorable air for the majority of the flow across the evaporator) found the thermostat held stage cooling off until the space temperature was 2° F *above* the setpoint; the first second stage cycle was generally very short and had little impact on the space temperature; after an hour or two of short cycles, the compressor finally runs for big blocks of time, in many cases taking the space temperature substantially below the set point. This conservative behavior seems designed to minimize over-cooling and manage unit cycling. The result is the space spends 2-3 hours at temperatures warmer than desired.

Thus a potentially major comfort issue related to economizers is intimately related to the thermostatic control logic used to control the equipment.

Maximum Outside Air.

The maximum outside air fraction sets the upper bound on the amount of cooling the economizer can deliver at a given outdoor temperature. Outside air dampers typically open fully but return air dampers rarely close completely. This helps to keep building pressurization under control but also impacts the economizer savings potential and exacerbates issues related to small amounts of delivered cooling during warmer conditions.

Most calculations assume economizers are capable of delivering 100% outside air, and code requires this capability. The substantial deviation from this standard for outside air means that economizer cooling is smaller than anticipated. This results in more mechanical cooling and the more potential comfort problems as stage 2 controls are employed.

To maximize savings and minimize problems the maximum outside air fraction should be as high as possible. In most cases this involves adjusting the damper linkage so that return dampers close completely while keeping the outside air dampers in their full open position. A few cases should be explored to see what building pressurization issues arise from moving to 100% from 70% or some lower outside air fraction. It is possible that some duct configurations would require some sort of pressure compensation. Units with minimal return ducts should not have this problem, while buildings with high pressure drop returns will have pressurization problems.

Discussion of September Monitoring

Doe-2 predictions of energy savings from various economizer logic models have been presented in earlier work. One goal of this work has been to verify the savings predictions of those models. There are two aspects to the savings predictions which have a great deal of uncertainty: the overall cooling load prediction, and the economizer performance.

The cooling load prediction abilities of the computer models have been proven time and again for a specific building with modeling specific conditions. The uncertainty here is whether the average conditions modeled are representative of the region. To validate this point fully, electricity consumption for a large number of facilities would need to be examined to determine the typical cooling loads. Depending upon the system type, this might require sub-metering. Average cooling energy and operating conditions could then be established to make robust predictions.

As such, the gross cooling load and economizer savings are uncertain. Even the economizer savings fractions have uncertainty related to this issue, as the cooling load profile is critical to determining economizer opportunities. A building that only has cooling in July and August will have very low savings fractions. A building with cooling year round will have very high fractions. For our purposes we have used operating conditions derived from the ELCAP monitoring project in 1990, combined with envelope, lighting and HVAC characteristics from the NEEA baseline study of 1996-vintage buildings.

The second uncertain aspect is whether, for a given set of conditions and cooling load, the model correctly predicts economizer savings for various economizer logic assumptions.

We attempted to use detailed monitoring of the economizer to shed light on this later point. Instead, we mostly have determined issues of importance to successfully monitor a space. The monitoring did provide some data on energy savings. Two separate projects were monitored. The first project was monitored during August and extremely hot weather made the results unusable, as there was very limited economizer action. The second project was monitored for 10 days during September. Again, remarkably hot weather prevailed for much of the test, but some economizer operations were observed.

Economizer Savings

Economizer energy savings are difficult to quantify from the monitored data. The lowest daytime temperature in the period is 60° F and 50% of the hours are over 72° F. The coldest daytime temperature with a cooling call is 62° F. As such the hours of availability and the potential cooling delivered are severely limited.

The average temperature difference between the return air and supply air with compressors off and the economizer damper open is 3.1° F in the changeover test and -0.3° F in the differential test. The actual delivered cooling must account for the fan heat temperature rise (1.58° F) when the fan is in the air stream. If the fan is assumed to run continously to provide ventilation then the cooling that offsets fan heat should be credited as delivered cooling, and the fan temperature rise can be added to the delivered return-supply delta temperature. If the fan is assumed to run only to deliver heating and cooling then the fan heat should not be credited as delivered cooling and the fan energy must be considered.

Since the predominate condition is continuous operation the fan heat is credited as delivered cooling. With fan heat added, the average delta t is 4.7° F in the changeover test and 1.3° F in the differential test. From this it is obvious that a 1° F or 2° F monitoring error would result in very different predictions of economizer savings. Given various sensor location issues it is very possible to have a 2° F error in this data set.

A very crude calculation would assume that the temperature difference during economizer operation is completely attributable to the economizer. This is not completely true since coils take a significant length of time to cool down. Economizer operation following a compressor run can show mixed air temperatures below either the return or outdoor air temperatures. In this work, the supply to return temperature difference is limited to the calculated mixed air temperature. Any supply temperatures below the theoretical mixed air temperature are set to the theoretical mixed air temperature.

The table below presents savings estimated using this modified delivered temperature difference. The "Average Drybulb" is the average ambient temperature during all periods with either economizer or compressor cooling. The "Delivered Cooling" column is the difference between the return air sensor and the supply air sensor. The supply air sensor is downstream of the fan so the fan temperature has been added in so this represents the cooling delivered assuming continuous fan operation. Duct losses are ignored.

Economizer Logic	Average Outdoor Drybulb Temperature(° F)	Delivered Cooling	Cooling Load (deg-hours)	Compressor Fraction	Economizer Fraction

Changeover 65 F("C+")	72.6	4.66	340	0.92	0.08
Differential	71.0	1.27	415	0.88	0.12

Comparison of the economizer savings between the two tests is problematic. Measurement issues led to the lack of ambient temperature data except when cooling is being requested. The absence of valid ambient data outside of cooling cycles make it difficult to compare the relative warmth of the periods. This is exacerbated by the change in economizer logic which increases the number of cooling hours by 50%.

A simple comparison of the two periods finds the differential period is cooler than the changeover period. So economizer savings should likely be larger during the differential period even if the logic is not changed. A qualitative comparison finds that the differential logic operates that economizer in many conditions where the changeover logic does not operate.

The modeling of the small office prototype predicts no difference in energy savings between these two cases (assuming non-integrated differential operation).

Comparison to Modeled Results

The monitored data was for two 4 day periods, one with a 65°-67° F change-over and one with differential enthalpy control logic. The system had 20% minimum outside air and 50% maximum outside air (with flow across the evaporator measured at 1675 SCFM).

The original simulations used a set of assumptions that were meant to represent the "average" building/system combination. Energy savings of 35% were predicted for an economizer with 65°F changeover without integrated operation in the Seattle climate (with 20% minimum outside air). The non-integrated differential logic is predicted to produce identical results[bd1].

A major difference between the project monitored here and the modeling assumptions is the maximum outside air fraction. Using a maximum air fraction of 50% yields a savings estimate of 10% rather than 35%.

A second difference is in the definition of integrated economizer operation. The monitored system allows integrated operation but it is compromized by having no compressor modulation. The economizer modulates to keep supply air about 50F. As such within a few minutes of the start of compressor operation the economizer generally goes to minimum air defeating integrated operation. This is intermediate to the modeled conditions of integrated and non-integrated economizer and we believe the closest approximation is the non-integrated economizer. However it is clear from the monitored data that there is some integrated operation.

While the estimate 8% to 12% savings agree well with the simulation estimate of 10%, monitored data from a few days cannot be compared with annual simulations directly. A look at the temperatures during the monitoring period indicate that it was substantially hotter than a typical September and not very diverse (either in terms of diurnal variation or overall daily averages) making comparison to monthly data inappropriate without real-time weather data.

The only reasonable approach is to compare economizer operation at given temperature bins. If the monitored data covers an adequate range of temperatures then a valid comparison can be made. The tables below present measured and modeled data summarized by outdoor temperature bin. The modeled data is for a complete year; the measured data is for the 8 day monitoring period. The model predicts economizer action over a relatively narrow band of outdoor temperature. Significant cooling load develops at the 54° F temperature bin. The economizer meets most of this load. The economizer fraction tapers off quickly, so that by the 66° F bin, all of the cooling load is met by the compressor. This same result occurs whether changeover or differential logic is modeled.

The second table presents the monitored data. Nothing can be said about when cooling load develops but at the coolest temperature bin with cooling, 61° F, the economizer meets all of the cooling load. The economizer fraction holds steady at 100% until 67° F and then it quickly drops to near 0% at 70° F. The differential test shows a significant difference above 70° F with small amounts of cooling load being met by the economizer up to the 75° F temperature bin.

Interestingly, the modeled results only show an economizer fraction of 15% at 61° F, compared with 100% in the monitored data. This indicates that the model brings the compressor on to keep up with load most of the time when conditions are in the sixties while the Honeywell thermostat holds off the compressor. This could be a difference in the cooling loads of the modelled versus monitored spaces, or it could be a difference in the logic that initiates second stage cooling.

The discrepancies between the monitored and measured data seem to indicate that monitored economizer savings are larger than those predicted. Economizer operation extends to warmer temperatures in the monitored data. This is very possibly a result of thermostat logic. The model keeps tight rein on the space temperature and quickly brings in the compressor to meet load. The observed delay in the second stage onset seen in the monitored data would allow the economizer to continue operating at elevated temperatures.

As pointed out earlier, there is large uncertainty in the measured results. The uncertainty could easily explain all of the difference between modeled and measured results. Error could also be responsible for the difference between the changeover and differential tests. However, a review of basic system operation has proven the economizer controller does perform basic functions properly (changeover, compressor operation, modulation of outside air fraction in response to input from the discharge air sensor).

Modeled Economizer Performance by Ambient Temperature Bin[bd2]

Ambient Temperature (F)	Total Hours	Cooling Hours	Total Cooling	Economize Cooling	Cooling Degree-Hours (base 60° F)	Economizer Fraction	Compress Fraction
47	213	1	0.5	0.4	0	0.783	0.21
48	194	3	1.1	0.8	0	0.738	0.26
49	177	1	1.7	1.6	0	0.941	0.05
50	179	1	0.2	0.1	0	0.583	0.41
51	179	3	3.2	3.2	0	1.000	0.00
52	180	2	3.1	3.1	0	1.000	0.00
53	191	5	4.6	4.6	0	1.000	0.00
54	181	9	21.2	8.0	0	0.378	0.62
55	177	23	42.8	35.5	0	0.830	0.17
56	202	34	73.0	62.5	0	0.856	0.14
57	204	28	67.7	43.6	0	0.644	0.35
58	219	56	128.2	86.7	0	0.676	0.32
59	185	55	168.8	73.0	0	0.433	0.56
60	164	51	175.5	49.4	0	0.282	0.71
61	149	54	184.3	28.4	149	0.154	0.84
62	125	48	191.0	16.6	250	0.087	0.91
63	91	40	154.4	8.9	273	0.058	0.94
64	101	56	276.9	4.6	404	0.017	0.98
65	105	56	281.6	1.3	525	0.005	0.99
66	80	44	246.1	0.0	480	0.000	1.00
67	85	39	224.4	0.8	595	0.004	0.99
68	71	32	181.0	-0.1	568	0.000	1.00
69	59	36	212.2	0.5	531	0.003	0.99
70	75	45	287.3	-0.1	750	0.000	1.00
71	62	35	232.1	0.0	682	0.000	1.00
72	47	29	198.5	0.0	564	0.000	1.00
73	36	19	136.5	0.0	468	0.000	1.00
74	26	14	104.9	0.0	364	0.000	1.00
75	34	16	131.8	0.0	510	0.000	1.00
76	23	13	104.8	0.0	368	0.000	1.00
77	8	4	33.3	0.0	136	0.000	1.00
78	16	9	79.2	0.0	288	0.000	1.00
79	18	13	117.2	0.0	342	0.000	1.00
80	16	10	89.6	0.0	320	0.000	1.00
81	10	8	74.7	0.0	210	0.000	1.00
82	9	6	59.9	0.0	198	0.000	1.00
83	14	12	119.7	0.0	322	0.000	1.00
84	5	4	42.1	0.0	120	0.000	1.00
85	4	2	21.2	0.0	100	0.000	1.00
86	1	0	0.0	0.0	26	.	.
87	1	1	11.0	0.0	27	0.000	1.00

Measured Economizer Performance by Ambient Temperature Bin

Ambient Temperature (°F)	Total Hours	Delivered Temperature Difference (F)	Cooling Degree-Hours (base 60° F)	Compressor Fraction	Economizer Fraction
Changeover Test					
61	1.0	5.8	0.1	0.00	1.00
63	3.0	5.4	1.0	0.00	1.00
64	1.0	5.1	0.8	0.00	1.00
66	7.9	4.4	24.3	0.18	0.82
67	2.0	4.1	9.1	0.00	1.00
68	2.0	4.4	8.8	0.78	0.22
69	2.0	3.0	6.9	0.97	0.03
70	1.0	.	5.3	1.00	0.00
71	1.0	.	3.1	1.00	0.00
73	3.0	.	9.1	1.00	0.00
74	2.0	.	5.7	1.00	0.00
75	2.0	.	11.7	1.00	0.00
76	3.0	.	18.1	1.00	0.00
77	1.0	.	10.6	1.00	0.00
78	4.0	.	39.1	1.00	0.00
79	2.0	.	22.6	1.00	0.00
81	3.0	.	43.5	1.00	0.00
82	0.6	.	7.9	1.00	0.00
83	5.0	.	92.7	1.00	0.00
84	1.0	.	20.0	1.00	0.00
.	51.0	.	0.0	.	.
Differential Enthalpy Test					
64	2.0	4.8	1.5	0.00	1.00
66	3.0	4.4	2.3	0.00	1.00
67	2.0	3.8	8.6	0.22	0.78
68	5.0	3.2	10.6	0.00	1.00
69	5.0	2.3	33.0	0.54	0.46
70	0.3	0.3	2.6	0.96	0.04
71	7.0	1.2	72.2	0.83	0.17
72	3.0	0.8	36.0	0.92	0.08
73	3.0	0.1	36.1	0.94	0.06
74	2.0	0.4	28.1	0.93	0.07
75	2.0	-0.6	29.4	0.96	0.04
76	3.0	-1.1	47.6	0.99	0.01
77	2.0	-1.1	34.6	0.99	0.01
78	4.0	-2.4	72.1	1.00	0.00
	53.0				

<Insert comparison table (with compatible outdoor Ts overlapped and linking text>

Considerations for Future Monitoring

In future monitoring, the following items are notable:

- Extreme care should be exercised in placing the ambient sensor to minimize solar effects. It should be noted that the true incoming air temperature and the economizer control sensor measured temperature are probably different. Important questions include, how does the solar/roof temperature impact the ambient sensor of the economizer? Does the Architectural Energy Corporation MDL sensor react similarly to the Honeywell control sensor? If so, then the hours of economizer operation will be significantly reduced from what a model would predict. [OK3]Second, how does the measured temperature relate to the actual air temperature of air sucked into the unit? This has serious consequences regarding any measures of heat balance. Additionally, air movement in the hood will impact the reading of any sensor located there.

- The damper sensor (string potentiometer) worked quite well for determining whether the damper was open or closed. It did exhibit significant drift over time and with ambient temperature, slowly stretching as the temperature rose even without any damper arm displacement. For determining "full open" and "full closed" damper position, this is not a problem. For situations where dampers are modulating through a full range of conditions, this may be slightly problematic, but overall the sensor is a vast improvement over using mixed air temperature to determine air fractions.

- Results for a brief initial monitoring period should be examined to determine whether the units are appropriate for monitoring. In the first project, one of the units monitored had no cooling even though the average daytime exterior temperature was 80° F.

- Monitoring in the middle of summer is less than desirable for several reasons. Generally high temperatures severely reduce the opportunity for economizer operation, and the small temperature differences during warm whether make quantification of the economizers contribution very error prone. Units that only cooling in the middle of summer are poor candidates for economizer savings and for monitoring. Units with cooling in spring and fall are ideal candidates for economizer potential and monitoring.

Installation and Troubleshooting Considerations

A number of issues deserve careful consideration when troubleshooting economizers and examining thermostat selection and performance. These factors have central importance in delivering savings and maintaining occupant comfort.

Sensors

Sensor calibration and settings are very important to economizer function especially as more aggressive control strategies are implemented. Sensor calibration takes on a heightened importance when differential economizer logic is employed. Being able to verify sensor output to the nearest degree is important.

Assessing sensors depends both on the manufacturers documentation and the characteristics of the specific sensor:

The Honeywell sensors produce an output current in the milliamp range. While this range is well-documented, the signal is not easily detectable by many commonly available multi-meters. With the available information for Honeywell sensors, it is only possible to verify sensor readings to within several degrees (inadequate to assess sensors used in differential logic).

The check-out procedure for the Trane sensors is not as clear and there is no detailed table of expected outputs, only response curves. The Trane literature provides specific tolerances for how much hysterisis can be expected from their solid state sensors. For example, if using differential enthalpy, a change in choice of air stream will occur when there is a difference of 3.0 Btu/lb or more between return air enthalpy and outdoor air enthalpy. The tolerance for dry bulb sensors is set at 0.5° F [OK4]

Sensor outputs have been presented by Honeywell as linear (or nearly linear) over the range of interest, or close to it. Especially with enthalpy sensors, it is difficult to determine if this is an accurate simplification. Our experience, and that of others, has suggested enthalpy sensors are notoriously unreliable, even when new, and their accuracy degrades with age. This has led us to lean toward strongly toward dry bulb sensors since there is only one degree of freedom to consider in sensor accuracy. Also, in differential mode, the simulated savings for dry bulb vs. enthalpy sensors are very similar, and the solid state dry bulb sensor costs half of what a solid state enthalpy sensor costs. If the system is to be run only in changeover mode, a snap-disc control is extremely inexpensive and, assuming it operates correctly, is the simplest and most straightforward choice.

Summary of Thoughts on Thermostats and Aggressive Economizer Logic.

The use of aggressive economizer logic allows economizer operation at warmer outdoor air temperatures, thereby adding smaller amounts of cooling. This has implications for occupant comfort and energy savings.

Monitoring data from the differential economizer operation show interior space temperatures peaking at 1, 2, or even 3° F warmer than the nominal setpoint. On a typical summer day, the first call for cooling will be in the morning, when differential economizer logic would allow economizer operation. On many mornings, the delivered economizer cooling is very minimal and not adequate to meet the cooling load. The space temperature rises until the thermostat signals for stage two, which finally starts the compressor. Some thermostats will handle this better than others. A thermostat that quickly figures out the space is getting warmer and triggers the second stage rather than waiting for it to get 1 or 2 degrees warmer would seem to be a superior choice.

Second, while the economizer always delivers cooling (in properly operating units), it doesn't always save energy. In cases where the air handler fan is not operated continuously, the efficiency of the economizer (including fan energy consumption) must be compared with the efficiency of the compressor (including fan energy). Each economizer system will have a crossover point, at which time it is better to operate the compressor. If the economizer is operated as stage one cooling and is delivering 4° F of cooling, it will have to operate 5 times longer than the compressor to deliver the same amount of cooling and the added fan energy negates any energy savings. As such, systems that are likely to have fans run only when heating or cooling is required (that is, in "Auto" mode) should be limited to a 60-65° F changeover point, or to differential logic that requires at least a 5-10° F temperature difference between return and outside air temperature depending upon fan motor heat and the maximum outside air fraction[db5].

In summary, in typical small commercial applications, there are several issues which need to be addressed to insure occupant acceptance and energy savings.

- The Honeywell T7300 series commercial thermostat appears to let space temperatures ride too high while the economizer simultaneously delivers a small amount of cooling. Stage two is used too sparingly. The latest T7300 (series F) includes settings for changing the second stage logic to a straight temperature difference logic with adjustable set point (deadband). This needs further exploration, as does the use of thermostats utilizing derivative control. A thermostat that looks at the rate of temperature change (using the temperature derivative, the "D" in a P + D control[db6]) should behave more acceptably in handling the variable capacity of the economizer when economizers are allowed to operate at warmer temperatures.

- The changeover setpoint on the Honeywell controller is very crude, which suggests differential could be a better choice when using Honeywell components[db7]. The changeover setpoint on Trane systems, in our experience, is much better.

- Differential control is very sensitive to sensor calibration. With available materials the level of accuracy and precision of Honeywell sensors (especially enthalpy sensors) can not be verified adequately to ensure differential control will work as the manufacturer claims.

Occupant comfort is a subject in and of itself. At the one site, when the system was changed so that the compressor was actually off for some part of the time during a stage 1 cooling call (before it was jumpered to always come on, even when conditions were favorable for exclusive economizer operation), indoor temperatures did increase slightly. However, the perception of the supervisor at the site was that indoor temperatures had increased "at least 5° F". She had not received any specific complaints from her co-workers, however. We did not receive any comfort complaints during the time the control strategy was changed, nor did the HVAC company that coordinated our activities. We are uncertain we would have learned anything about this issue if we had not asked the supervisor directly. But it is likely that if an aggressive economizer setpoint were instituted in many instances, occupants would notice the difference in comfort and would act to reduce the benefits of the economizer through complaints and HVAC technician adjustments.

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[bd1]Mike – is this really true, or is there something I'm missing here? Or maybe you are just saying the cut-in point is the same for compressor operation, but oveall savings are still different. This is what the simulations showed. One general issue is that the simulations consider integral econo operation to be as it is in large equipment where the cooling modulates, as opposed to the equipment we have where the econo modulates (generally closed). The savings are identical for non-integrated operation which I chose as being closest to what we see.

Page: 8

[bd2]What are the units (kWh, etc) for total cooling and economizer cooling? They need to be inserted into the table.

Page: 10

[OK3]This seems to cross over to being a economizer savings statement rather than a monitoring statement like the rest of the paragraph. It is confusing to me.

Page: 11

[OK4]Is this talking about the sensors or the differential econo logic? If logic then the first sentence needs to be clearer. If sensors then the second sentence needs to be clearer.

Page: 12

[db5]If we are going to talk about enthalpy here we need to explain what the differential setting would be that would be equivilent to the "5-10°F" mentioned for dry-bulb sensors.

Page: 12

[db6]I think we have to explain what a P+D control is. I think that this is an important point that needs to be expanded. As I understand I the P+D control would bring the compressor on when the mixed air and the return air get to close together and not based on shifts in the return air temp toward the cooling setpoint?

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[db7]If you get to adjust the deadband as series F or is there another variable?