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Energy and IAQ Implications of Residential Ventilation Cooling

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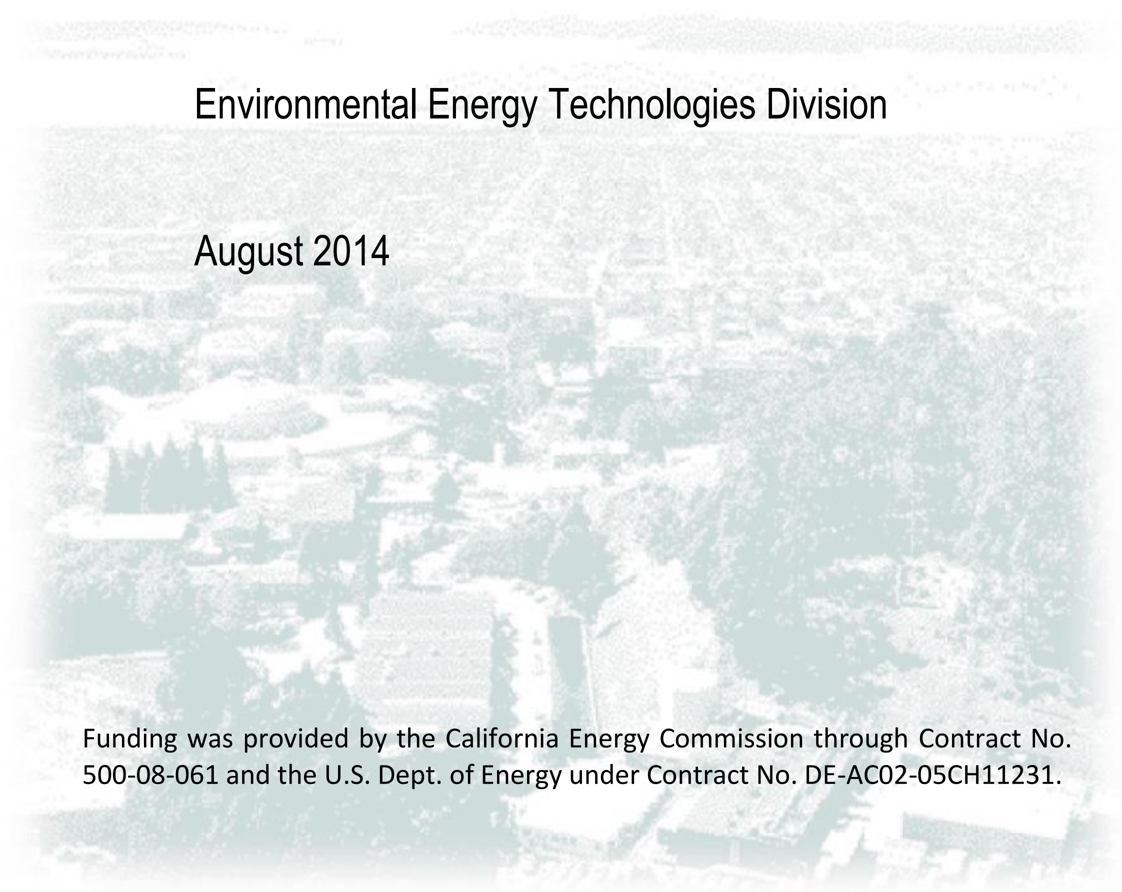

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ABSTRACT

This study evaluates the energy, humidity and indoor air quality (IAQ) implications of residential ventilation cooling in all US IECC climate zones. A computer modeling approach was adopted, using an advanced residential building simulation tool with airflow, energy and humidity models. An economizer (large supply fan) was simulated to provide ventilation cooling while outdoor air temperatures were lower than indoor air temperatures (typically at night). A typical new construction, low-mass, timber frame home with ASHRAE Standard 62.2 compliant ventilation was considered. The simulations were performed for a full year using one-minute time steps to allow for scheduling of ventilation systems and to account for interactions between ventilation and heating/cooling systems.

The results showed that energy savings from a residential economizer are moderate (up to 200 kWh of cooling energy per year) using a high performance brushless permanent magnet (BPM) air handler. A Permanent split capacitor (PSC) air handler led to less cooling energy savings, and in some climates a small (approximately 1%) cooling energy increase. The cooling energy savings were greatest in climate zones 3, 4, and 5. Economizers were found not to contribute to excess indoor humidity in hot, humid climates, but did lead to increased indoor humidity in more moderate climates (but still not to the level found in hot, humid climates). Economizer operation reduced annual relative exposure by about 5 to 15% indicating a significant potential for IAQ improvement. The improvements were greatest in climate zones 3B and 4B due to longer economizer operation time. One caveat with the ventilation cooling recommendations is that they were for a lightweight timber frame construction and may change for heavier brick/block structures not included in this study.

KEYWORDS

Night Ventilation, Air-Side Economizer, Air Conditioning, Ventilation Cooling, Peak Demand

Introduction

Air conditioning (AC) accounts for over 6% of residential energy use in the United States, with nearly 100 million homes using 186 billion kilowatt hours (or 0.635 quads) of energy per year (US EIA 2001). As of 2009, approximately 87% of US homes had air conditioning – up from 68% in 1993. This rising demand for mechanical cooling is an increasing burden on homeowners due to increased energy bills. The additional AC use also contributes to higher peak electricity demand loads and associated grid reliability issues; increased fossil fuel consumption (as most peak electricity is from natural gas power plants operated during peak events) and an increased national carbon output.

One approach to reduce residential AC use is ventilation cooling using an *economizer* – a large supply fan. The economizer is used to supply the home with cool outdoor air at an airflow rate much higher than minimum indoor air quality (IAQ) ventilation requirements. This usually happens during the night when outdoor air temperatures are cooler than indoor air temperatures. In commercial buildings energy standards such as ASHRAE Standard 90.1 (ASHRAE 2010b) require the installation of economizers to provide ventilation cooling. Some electric utility residential energy conservation programs (e.g., Sacramento Municipal Utility District's SMUD Advantage Program) feature residential economizers, but they remain rare in residential applications. Therefore, the opportunity for large-scale energy savings in residences could be large.

While most literature on ventilation cooling is in the domain of commercial buildings, some studies have been performed on residential buildings. An early report by Kammerud et al. (1983) concluded that ventilation cooling in the US had the potential to substantially reduce the use of residential mechanical cooling. They noted that buildings with higher thermal mass would be able to take greater advantage of ventilation cooling, but in order to realize fully the energy benefits, more effective means of dealing with the introduction of moist air (a drawback of ventilation cooling in humid climates) had to be developed. Givoni (1998) found that night ventilation was particularly effective in high-mass buildings. A theoretical analysis by Golneshan and Yaghoubi (1990) of residential buildings in hot, arid regions of Iran demonstrated that night ventilation could be used to reduce the indoor daytime temperature to an acceptable range of thermal comfort, without the use of mechanical cooling. Santamouris et al. (2010) analyzed energy data from 214 mechanically cooled residential buildings that also used night ventilation, and found that night ventilation could reduce the cooling load by up to 40 kWh/m^2/y. The average cooling load reduction was 12 kWh/m^2/y. They also concluded that the effectiveness of night ventilation increases with the cooling demand of the building. Springer (2007) combined ventilation cooling strategies with mechanical pre-cooling using the house AC system, while monitoring test residential buildings in Sacramento. Springer found that peak-period AC energy consumption could be reduced by up to 88%.

Ventilation cooling and economizer operation are controlled based on indoor and outdoor temperature conditions. The times throughout the year when these conditions are met greatly depend on climate. Therefore, the climate limits the energy saving potential of economizers and ventilation cooling, and in some cases may even prohibit any energy savings at all. Additionally, economizers have the potential to raise indoor humidity in climates with high outdoor humidity by supplying large quantities of moist, outdoor air. To avoid this problem economizer use is usually not recommended in humid climates because elevated indoor humidity can result in an uncomfortable indoor environment and possible health issues for the occupants, unless supplemental dehumidification is provided (as is done for commercial systems that independently control temperature and humidity).

Because economizers move much more air than infiltration or minimum mechanical ventilation required by standards (such as ASHRAE 62.2) there is the potential to significantly improve indoor air quality due to the additional dilution of indoor pollutants.

This study uses a computer modeling approach to examine the energy, humidity and IAQ implications of using an economizer in residential buildings. Fifteen different US climates were considered, covering the full range of dry and humid, and hot and cold climatic conditions.

Economizers

Economizers are large supply fans which reduce the cooling load of a building by supplying outside air to the occupied zone while outdoor air temperatures are cooler than indoors. The use of economizers is common in large commercial buildings where cooling loads and ventilation rates are often higher than in residential single-family homes. Using economizers can significantly reduce the energy demand on the mechanical cooling system while also delivering outside ventilation air, thus improving IAQ (EPA 2000). The airflow associated with economizers is usually 20 times (or more) than the minimum airflow required in residential ventilation standards such as ASHRAE Standard 62.2 (2010a). This high airflow is required in order to provide significant cooling capacity using small indoor-outdoor temperature differences. Outdoor nighttime temperatures generally result in delivered air temperatures between 2 and 4 K lower than indoors. For comparison, a typical AC unit delivers air approximately 10 K lower in temperature than indoor air.

Residential applications of economizers are less widespread. Unlike their commercial counterparts, residential building loads are dominated by heat exchange through the building envelope (conduction, air infiltration, solar gains, etc.) and require fewer hours of cooling at low ambient temperatures (Ueno & Straube 2011). However, there is still potential for energy savings by using night ventilation to reduce the use of mechanical cooling during the day.

In typical residential applications the forced air system air handler is used as the economizer fan. A damper system is used to open an air inlet from outside in place of the normal return air flow pathway from the house. This allows the economizer to distribute outside air to the occupied zone via the supply ducts. To avoid over-pressurizing the house a pressure relief mechanism is needed, e.g., a motorized skylight in the ceiling or pressure relief dampers in the return ducts. The main purpose of economizers is to provide cooling. However, because they supply outside air they also provide ventilation. In the future, with the use of the principle of *ventilation equivalence* (Sherman et al. 2011a) it is expected that there will be a mechanism to take credit for improved IAQ from economizer use in ventilation standards. In this study the principle of equivalent ventilation was used to estimate the change in IAQ due to economizer operation.

A potential drawback to using economizers is poor humidity control when outdoor humidity is high relative to indoors. During humid weather, the increased ventilation rate from the economizer can increase the indoor humidity, potentially leading to comfort, health, and moisture problems. Kubota et al. (2009) conducted field experiments on residential buildings in the humid climate of Malaysia. They showed that night ventilation provided good daytime indoor temperatures, but at the expense of higher indoor humidity. To maintain indoor comfort additional dehumidification may be required to control this extra moisture load. Traditionally the use of an economizer is not recommended in humid climates such as Miami, Florida for this very reason. Commercial buildings have systems that control both humidity and temperature. They also have higher outdoor air ventilation requirements compared to single-family residences, so economizers are more acceptable in these humid climates.

If improperly installed, the economizer components can lead to increased envelope leakage, especially through the dampers when the economizer is not operating (McWilliams & Walker 2005). This can cause higher levels of infiltration leading to larger space-conditioning loads during regular heating and cooling operation. Similarly, sealed and well-insulated ducts are necessary for effective use of an economizer to avoid warmer or contaminated air from the attic, crawlspace or garage being passed into the occupied zone. Therefore, a good duct system is a prerequisite for energy efficient economizer operation.

Indoor Humidity

The U.S. Environmental Protection Agency (EPA) recommends keeping indoor relative humidity below 60%, and ideally between 30% and 50% (EPA 2010). A comprehensive guide on the effects of indoor moisture and humidity on health and the indoor environment has been published by the World Health Organization (2009). Fungal growth is expedited by dampness in houses (Gallup et al. 1987; Waegemaekers et al. 1989; Douwes et al. 1999) which can lead to the production of harmful fungal spores and allergens. Fungi can also lead to the structural damage of buildings e.g., dry rot (Singh 1999).

ASHRAE Standard 160 (ASHRAE 2009) uses 70% as the design criteria for relative humidity (RH) based on various building failure criteria (mostly related to mold growth) summarized in Viitanen and Salonvaara (2001). High humidity can become a problem when RH is above 70% for timescales of the order of days. Condensation will form on windows and the indoor environment can be unpleasant for occupants. When RH is above 70% for timescales on the order of weeks, it can become a serious health and structural problem. There will be increased dust mite populations (Lstiburek & Carmody 1994) and mold growth with potentially serious health implications; porous parts of the structure can start to rot. Therefore, in this study, the time that indoor RH exceeded 70% was considered as the humidity evaluation criterion.

A sophisticated humidity model was used to analyze the humidity issue in depth. The model accounts for moisture removal by mechanical AC (including moisture storage on coils and other equipment cycling effects), sources of moisture in the home due to occupants, and moisture coupling between air in the home and other home components that act to store moisture. This model has been used successfully in previous studies and shown to produce indoor humidity levels (and rates of change of indoor humidity) that match measured values (see Walker and Sherman (2007) and Lstiburek et al. (2007) for more details).

In the absence of occupants, indoor humidity will eventually equal outdoor humidity assuming some level of ventilation. The presence of occupants can only increase internal humidity due to respiration, perspiration, and internal loads from occupant activities such as cooking, showering etc. Therefore, indoor humidity will be higher than outdoor humidity due to the presence of occupants, and ventilation should decrease indoor humidity. The use of AC (or dehumidifiers) removes moisture from indoor air. When this rate of removal is faster than the rate of addition from occupants and indoor sources, the indoor humidity will be lower than outdoors. In this case, ventilation, particularly at the high rates provided by economizers, can increase indoor humidity. This implies that economizer use should be limited to heating/shoulder seasons in humid climates when the outdoor humidity and air temperature are lower and AC is rarely used.

Low sensible load, energy efficient homes are particularly sensitive to humidity issues because their small AC systems lack the capacity or operating time to control humidity. Energy efficient homes have the same latent loads as conventional homes, but lower sensible loads which lead to reduced moisture removal capacity. Thus, energy efficient homes may perform differently from a humidity standpoint. Another possible climate-related consideration is in climates where it does not cool down enough at night for economizer operation. Coincidentally, this is more likely in humid climates, and so offsetting some humidity increase while limiting the potential for energy savings. The simulations in this study examine these humidity issues in detail.

PSC and BPM Air Handler Motors

Generally, residential economizers use the heating/cooling system air handler to deliver the ventilation cooling air to the house. Studies by the Canada Housing and Mortgage Corporation (1993) and Walker (2008) have shown that residential air handlers are almost an order of magnitude less efficient than their larger commercial counterparts.

A typical fan in a residential forced-air heating and cooling system has a permanent split capacitor (PSC) type motor. In residential furnaces, PSC motors usually have between two and four fixed speeds. Different speeds are necessary to match the different airflow requirements for heating and cooling operation (airflow rates for cooling are generally about 25% greater than for heating in cooling dominated climates).Typical residential PSC air handler performance is approximately 1 L/s/W (2 cfm/W) and can be reduced further by poor design of the duct system and the cabinet where the air handler is housed (Walker 2008).

An alternative to PSC air handlers are brushless permanent magnet (BPM) air handlers. At lower airflows or with well-designed low-pressure duct systems, BPM air handlers have the capability to better the performance of their PSC counterparts by a factor of five, to 5 L/s/W (10 cfm/W) (Walker 2008).

The speed of BPM motors are electronically controlled and can be set specifically to match the airflow requirements for each application. They are often used with controls that are designed to speed up or slow down the air handler in an attempt to preserve airflow regardless of the static pressure across the fan, e.g., when filters become dirty and increase the airflow restriction. This self-moderation helps maintain an airflow range through the heat exchanger close to the optimal flow rate for which the heat exchanger was designed, maintaining air conditioner efficiency. The drawback is cost - BPM air handlers are more expensive than PSC air handlers and are less common (roughly 25% of the market) in residential HVAC systems, though Raymer (2010) notes that this trend is changing. It should also be noted that the potential advantages of BPM air handlers are often negated in residential systems that have high airflow resistance because the BPM motor increases its power consumption in order to maintain airflow (Walker 2008).

This study includes consideration of the energy implications of switching from a standard PSC motor to a BPM motor when using the air handler for economizer operation.

Indoor Air Quality (IAQ)

IAQ was determined by calculating the indoor 'relative dose' and 'relative exposure' of occupants to indoor contaminants, based on a constant indoor emission rate (see Sherman (2008) and Sherman and Wilson (1986)). A relative dose of unity is what an occupant would receive if they lived in a house ventilated using a continuously-operating mechanical ventilation system (in this case an exhaust fan) sized to meet the ASHRAE Standard 62.2 (2010a) minimum whole-house mechanical ventilation rate. Relative dose and exposure levels above unity indicate a greater occupant exposure to pollutants and poorer IAQ. Relative dose and exposure levels below unity indicate less occupant pollutant exposure and improved IAQ. In this study, relative exposure was calculated every minute, while relative dose was calculated as a 24-h running average of the relative exposure. The default infiltration credit of 10 L/s per 100 m^2 (2 cfm per 100 sq. ft.) of floor area, as per the 2010 edition of ASHRAE Standard 62.2, is included in the dose and exposure calculations.

Relative dose and exposure were calculated for occupied hours only. The house was assumed to be unoccupied between the hours of 8 am and 4 pm every weekday, and then occupied for the rest of the time.

Simulations

Climate Zones

Simulations were performed for all DOE climate zones (1 to 8, and A to C) using TMY3 weather data (Wilcox & Marion 2008) for their representative cities (see Figure 1 and Table 1) (Briggs et al. 2003).

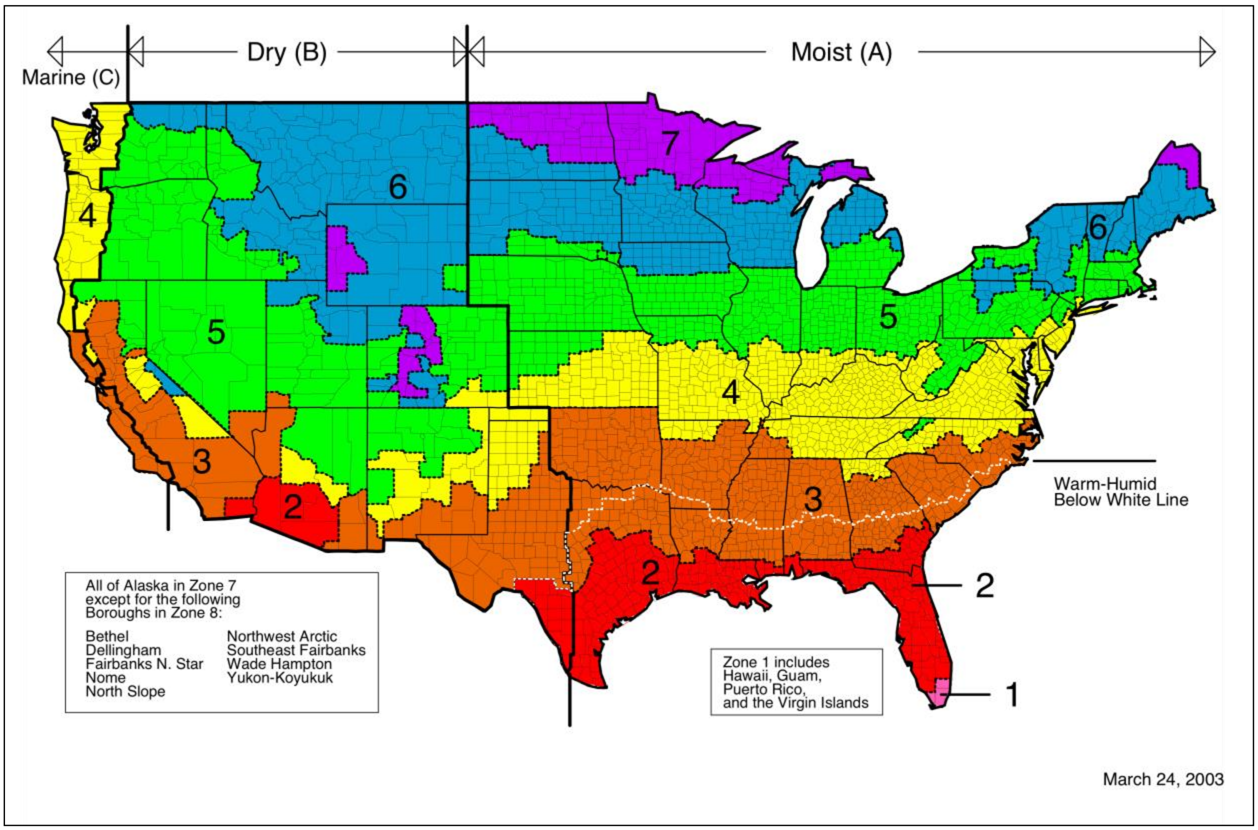


Figure 1: IECC Climate zones for the United States (Briggs et al., 2003)

Table 1: IECC Climate zones with definitions (Briggs et al., 2003)

Climate Zone	Representative City	State	Temp	Moisture	Köppen Classification Description
1A	Miami	FL	Very Hot	Humid	Tropical Wet-and-Dry
2A	Houston	TX	Hot	Humid	Humid Subtropical (Warm Summer)
2B	Phoenix	AZ	Hot	Dry	Arid Subtropical
3A	Memphis	TN	Warm	Humid	Humid Subtropical (Warm Summer)
3B	El Paso	TX	Warm	Dry	Semiarid Middle Latitude/Arid Subtropical/Highlands
3C	San Francisco	CA	Warm	Marine	Dry Summer Subtropical (Mediterranean)
4A	Baltimore	MD	Mixed	Humid	Humid Subtropical/Humid Continental (Warm Summer)
4B	Albuquerque	NM	Mixed	Dry	Semiarid Middle Latitude/Arid Subtropical/Highlands
4C	Salem	OR	Mixed	Marine	Marine (Cool Summer)
5A	Chicago	IL	Cool	Humid	Humid Continental (Warm Summer)
5B	Boise	ID	Cool	Dry	Semiarid Middle Latitude/Highlands
6A	Burlington	VT	Cold	Humid	Humid Continental (Warm Summer/Cool Summer)
6B	Helena	MT	Cold	Dry	Semiarid Middle Latitude/Highlands
7	Duluth	MN	Very Cold	-	Humid Continental (Cool Summer)
8	Fairbanks	AK	Subarctic	-	Subarctic

House Construction and HVAC Equipment

The effects of economizer operation were determined by the difference between a reference case with whole-house mechanical ventilation complying with ASHRAE 62.2 (2010) (i.e., a continuously-operating mechanical exhaust fan with an airflow rate of 28 L/s (60 cfm) with a power consumption of 11.8 W) and an identical home with an economizer. For both cases the relative dose and exposure were calculated relative to a home with ASHRAE 62.2 whole-house mechanical ventilation only, i.e., no natural infiltration or other mechanical ventilation systems.

House geometry was based on the California State Energy Code Title 24 Prototype C (Nittler & Wilcox 2008) (referred to as Title 24). The modeled home had an occupied living area of 195 m^2 (2,100 ft^2) with uniform 2.5 m (8.2 ft) ceilings. There were four occupants within the house, three bedrooms, three bathrooms and one kitchen. Envelope leakage was 4.8 ACH_{50}, typical of new construction in California, based on recent studies by Offermann (2009) and Wilcox (2011). The simulations used this envelope leakage combined with assumptions about leakge distribution and weather data to determine the natural infiltration rate for every time step. The reference case used the ASHRAE 62.2 (2010) default infiltration credit which is 20 L/s(42 cfm) for the modeled home, added to the ASHRAE 62.2 continuous mechanical ventilation rate for a total of 48 L/s (102 cfm) that is the same for every time step.

House insulation and fenestration values were based on the IECC 2009 values (IECC 2009). The home in this study had thermal characteristics representing lightweight, timber-frame construction. While this is typical of the majority of new home construction in the US, homes built from brick or block with higher thermal mass may have different results due to the longer time constants associated with heating and cooling the structures. The issue of higher mass homes should be addressed in future work.

Heating and cooling equipment was sized according to ACCA Manuals J & S (ACCA 2006). For heating, an 80% AFUE natural gas furnace was used. For cooling, a SEER 13 split-system air conditioner with a TXV refrigerant flow control was used. Heating and cooling ducts were located in the attic. The total duct air leakage was 6%, evenly split with 3% supply leakage and 3% return leakage, in order to meet the limit for the tight duct credit in Title 24. Six percent represents a reasonable level of duct air leakage that can be achieved in practice and is representative of a well-installed duct system.

Thermostat set points used the heating and cooling temperatures recommended in the 2008 California Title 24 Residential ACM (see

Table 2). The decision to switch between heating and cooling days was based on a 7-day running average of the outdoor air temperature. If the running average outdoor air temperature was less than or equal to 15.56°C (60°F) the HVAC system would operate in heating mode. If the running average was above 15.56°C then the HVAC system would switch to cooling mode. For climate zones 1A and 2A (the humid climates of Miami, FL and Houston, TX) the cooling set point was set to a constant 23.3°C (74°F) to represent more realistically how AC is used to maintain indoor temperature and reduce the humidity in that region (Rudd & Henderson 2007). Consequently these climates exhibit higher cooling energy use than if the standard set points from

Table 2 were used.

Table 2: Thermostat heating and cooling settings for simulations from the 2008 Title 24 ACM

Time			Heating		Cooling	
			[°C]	[°F]	[°C]	[°F]
0:00	→	1:00	18.3	65	25.6	78
1:00	→	2:00	18.3	65	25.6	78
2:00	→	3:00	18.3	65	25.6	78
3:00	→	4:00	18.3	65	25.6	78
4:00	→	5:00	18.3	65	25.6	78
5:00	→	6:00	18.3	65	25.6	78
6:00	→	7:00	18.3	65	25.6	78
7:00	→	8:00	20.0	68	28.3	83
8:00	→	9:00	20.0	68	28.3	83
9:00	→	10:00	20.0	68	28.3	83
10:00	→	11:00	20.0	68	28.3	83
11:00	→	12:00	20.0	68	28.3	83
12:00	→	13:00	20.0	68	28.3	83
13:00	→	14:00	20.0	68	27.8	82
14:00	→	15:00	20.0	68	27.2	81
15:00	→	16:00	20.0	68	26.7	80
16:00	→	17:00	20.0	68	26.1	79
17:00	→	18:00	20.0	68	25.6	78
18:00	→	19:00	20.0	68	25.6	78
19:00	→	20:00	20.0	68	25.6	78
20:00	→	21:00	20.0	68	25.6	78
21:00	→	22:00	20.0	68	25.6	78
22:00	→	23:00	20.0	68	25.6	78
23:00	→	0:00	18.3	65	25.6	78

ASHRAE 62.2 compliant bathroom exhaust fans of 25 L/s (50 cfm), kitchen range hoods of 50 L/s (100 cfm), and vented clothes dryers of 75 L/s (150 cfm) were simulated based on the following occupancy schedule: on weekdays one bathroom fan was operated for 30 minutes per occupant every morning (to simulate showering) and again for 10 minutes per occupant in the evening (to simulate bathroom usage). On weekends the fan run time per occupant was the same as for weekdays, only the operation times were constrained between 7 am and 11 pm. The kitchen range hood operated for one hour per day between 5.30 pm and 6.30 pm. On weekends there was an additional 30 minutes of operation in the morning between 9.30 am and 10.00 am. Two laundry days each week were simulated for clothes dryer operation. Each laundry event was three hours long.

Internal Loads

Both latent and sensible loads were included in the simulations. The daily latent heat gain from moisture generation followed the approach used previously by Walker and Sherman (2006) and Walker and Sherman (2007). The moisture generation rates were based on the design levels in ASHRAE Standard

160 (2009) (13.8 kg/day for four occupants) with corrections for kitchen and bathroom exhaust using the bathing, cooking and dishwashing estimates from Emmerich et al. (2005). Consistent with the operation of kitchen and bathroom exhaust fans, It was assumed that all the kitchen and bathroom generated moisture was vented directly to outside resulting in a reduction of 4 kg/day in internal moisture load. Therefore, the net moisture generation rate was 9.8 kg/day (21.5 lb/day). The daily sensible heat gain from lights, appliances, people and other sources used the Title 24 Alternative Calculation Method (CEC 2008) value of 5.9 kWh/day (20,000 Btu/day) for each dwelling unit, plus 0.0044 kWh/day (15 Btu/day) for each square foot of conditioned floor area. For the simulated house this resulted in a continuous sensible load of 630 W.

Economizer control for Ventilation Cooling

The economizers in this study used the following control strategy. The economizer operated when the outdoor temperature was 3.3 K (6°F) or more below the indoor set point and the house temperature was greater than 21°C (70°F). Because the system was unbalanced, a large hole in the ceiling with area 0.31 m^2 (3.34 ft^2) opened as a pressure relief. The hole was sized to result in approximately 2 Pa (0.008 in. water) of house pressurization based on the size of the economizer fan, which was dependent on the HVAC equipment sizing. The economizer would only be allowed to run while the HVAC system was in cooling mode. This strategy was to prevent inadvertent ventilation cooling by the economizer while the HVAC system was in heating mode. There is a limit for the indoor-outdoor temperature below which there is no advantage to economizer operation when the cooling effect of the incoming air offsets the heat from the air handler motor. If duct gains and losses and house infiltration effects are neglected, a 500 W air handler providing 500 L/s (1000 cfm) of air flow requires a temperature difference of 1K (1.6° F) to offset the additional air handler heat, so the above control strategy has a low temperature difference cutoff (3.3 K) that should avoid operation of the economizer when there is no benefit.

For each climate zone the economizers were sized depending on the airflow rate and power of the air handler unit (Table 3). While heating or cooling, the PSC air handlers in the study delivered approximately 1 L/s/W (2 cfm/W) while operating in both high-speed and low-speed modes. The PSC economizers operated in high-speed mode. The BPM air handlers delivered approximately 1 L/s/W (2 cfm/W) in high speed mode, and 5 L/s/W (10 cfm/W) in low speed mode (Walker 2008). The BPM economizers operated in low-speed mode to take advantage of the better power performance at lower airflow rates.

Table 3: Economizer power consumption and airflow rate by climate zone

PSC/BPM	Climate Zone 1A	2A	2B	3A	3B	3C	4A	4B	4C	5A	5B	6A	6B	7	8
PSC Power [W]	1167	675	1075	650	750	302	540	540	420	630	630	630	840	840	1050
PSC Airflow Rate [L/s]	1100	640	1010	610	710	290	510	510	400	590	590	590	790	790	990
BPM Power [W]	84	84	84	126	84	54	84	84	73	88	101	75	93	67	67
BPM Airflow Rate [L/s]	400	400	400	590	400	260	400	400	350	420	480	350	440	320	320

Simulation Tool

The energy consumption and IAQ of the modeled houses was evaluated using the REGCAP residential building simulation tool. The REGCAP model, developed and validated at the University of Alberta (Walker 1993) and Lawrence Berkeley National Laboratory (Walker & Sherman 2007), is a residential HVAC model that combines ventilation, heat transfer, and moisture models to determine annual residential energy use as a function of building characteristics and location. Commercially available software such as EnergyGuage (Fairey et al. 2000), and programs like HOT2000 (Halrecht et al. 1999) have been developed for use in home energy ratings (RESNET 2006). However, these software packages do not have very sophisticated ventilation models. Energy Plus, developed by the U.S. Department of Energy (DOE) is used for both commercial and residential building energy simulations, but has been shown to have deficiencies in modeling infiltration impacts in residential modeling (Spencer 2010).

REGCAP performs a heat and mass balance on the modeled house and HVAC system with a time resolution of one minute. REGCAP includes all HVAC system-related airflows (including duct leakage and registers), models of air conditioner performance that include the effects of coil airflows, indoor and outdoor air temperature, and humidity. REGCAP accounts for thermal losses and gains from the home due to conduction, radiation, and heat transfer to the outside from the building envelope and duct system, as well as solar gains. The REGCAP model calculates the home conditions for each minute and turns the thermal conditioning equipment on and off based on the calculated temperature of the home. The conditioning equipment is modeled as adding/removing energy from the space at the rate specified for the conditioning equipment. The temperature and energy output of the conditioning system from the previous time step, along with any heat gains or losses, are used to compute the temperature in the house for the next time step. The conditioning equipment is then turned on or off accordingly. The annual HVAC system energy use and building air exchange rate are determined.

REGCAP has been extensively verified and been shown to predict HVAC equipment energy consumption within 4% of measured systems. Ventilation rates are predicted within approximately 5% over a wide range of house leakage distributions and weather conditions (Wilson & Walker 1992a; Wilson & Walker 1992b; Siegel 1999; Walker et al. 1999; Siegel et al. 2000; Walker et al. 2002; Walker et al. 2005; Walker & Sherman 2007).

Results and Discussion

Indoor Air Quality (IAQ)

Figure 2 to Figure 5 show the relative dose and exposure for the house with and without a BPM economizer. The relative doses and exposures have been hourly averaged and the figures show the annual averages, minimums, and maximums during occupied time periods, for all IECC US climate zones. To comply with ASHRAE 62.2 which uses the principle of ventilation equivalence (Sherman et al. 2011b), an annual relative dose equal to or less than one is required. This is achieved in all cases. Decreasing relative dose and exposure indicate an increase in IAQ for occupants.

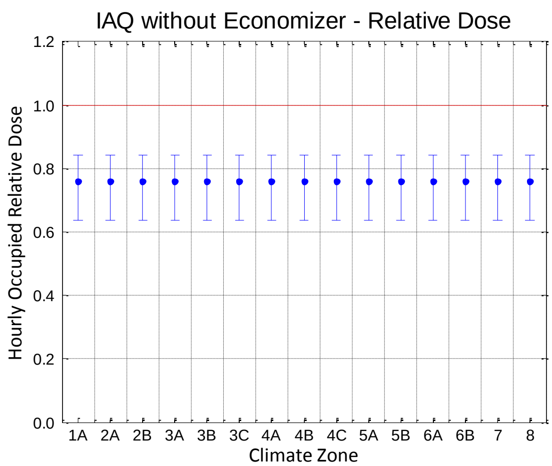


Figure 2: Hourly occupied relative dose without an economizer

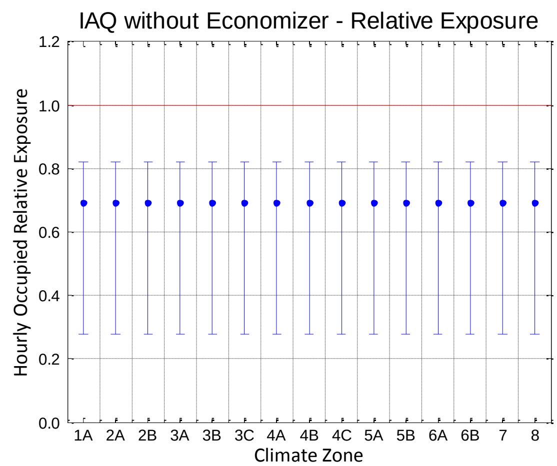


Figure 3: Hourly occupied relative exposure without an economizer

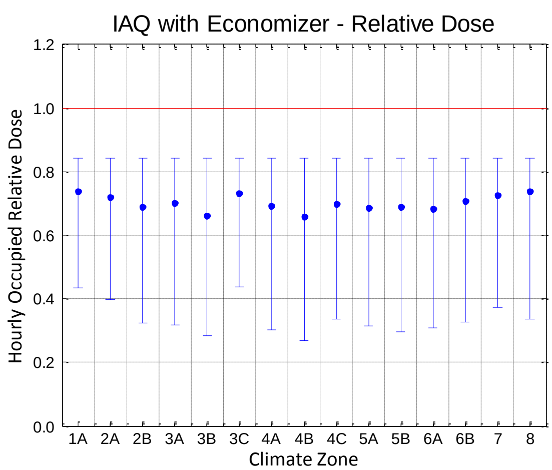


Figure 4: Hourly occupied relative dose with an economizer

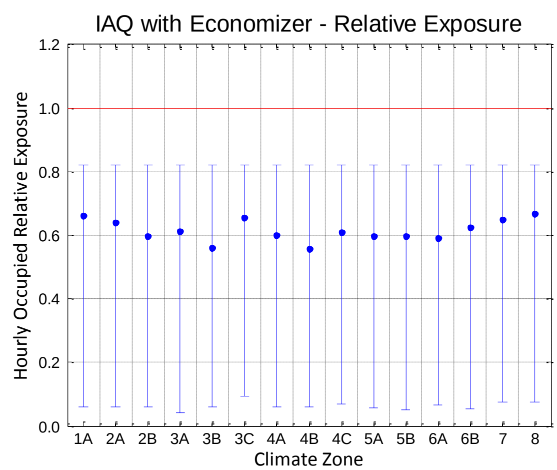


Figure 5: Hourly occupied relative exposure with an economizer

Without an economizer the mean, minimum and maximum relative dose and exposures are determined by the whole-house ventilation fan, plus the kitchen, bathroom and dryer exhaust and the fixed natural infiltration rate. Because these were the same in every climate, there is no climate-to-climate variation in the results. The relative dose and exposure are always less than one due to the contribution of the kitchen, bathroom and dryer exhausts, and infiltration towards the total ventilation rate, in addition to the whole-house ventilation fan used as the reference for the relative dose and exposure calculations.

In all climates, the economizer acts to further reduce the mean annual relative dose and exposure by approximately 40% resulting in improved IAQ. This is due to the additional airflow from the economizer fan. The relative dose and exposure minimums are low for the simulations with economizer operation because the economizer airflow rates are much larger than the ASHAE 62.2 compliant whole-house exhaust fan. Predicting specific health outcomes due to this reduction in relative dose and exposure are beyond the scope of this study, but may be investigated in the future when specific knowledge of pollutants of concern (such as their specific health impacts and emission rates) are better known.

The simulations with economizer operation showed that there is some climate-to-climate variation in the relative dose and exposure. This is due to the economizer operating for different amounts of time in each climate (as per the economizer algorithm) and because the economizer uses the air handler fan which has different flow rates in different climates. The climates with the most economizer operation see the greatest increase in IAQ. The maximum relative doses and exposures occur during the heating season when the economizer does not operate. As all the climate zones used the same kitchen, bathroom and dryer fan schedule the maximum relative doses and exposures are the same independent of climate zone. The minimums occur during economizer operation in the cooling season.

Energy

Figures 6 to 13 show the annual cooling energy, air handler operation time, air conditioner operation time and cooling peak load reduction on an annual basis. The cooling energy is defined as the total annual electrical energy used by the air-conditioning compressor, plus the electrical energy used by the air handler while operating in cooling mode and while operating in economizer mode. The operation time is simply the sum of all the time in the year which the air handler or air conditioner operates. The cooling peak load is defined as the total cooling energy used between the hours of 2 pm and 6 pm each day that air conditioning occurs. The values in boxes are the fractional changes due to economizer operation, where a positive number indicates that economizer operation increases energy use.

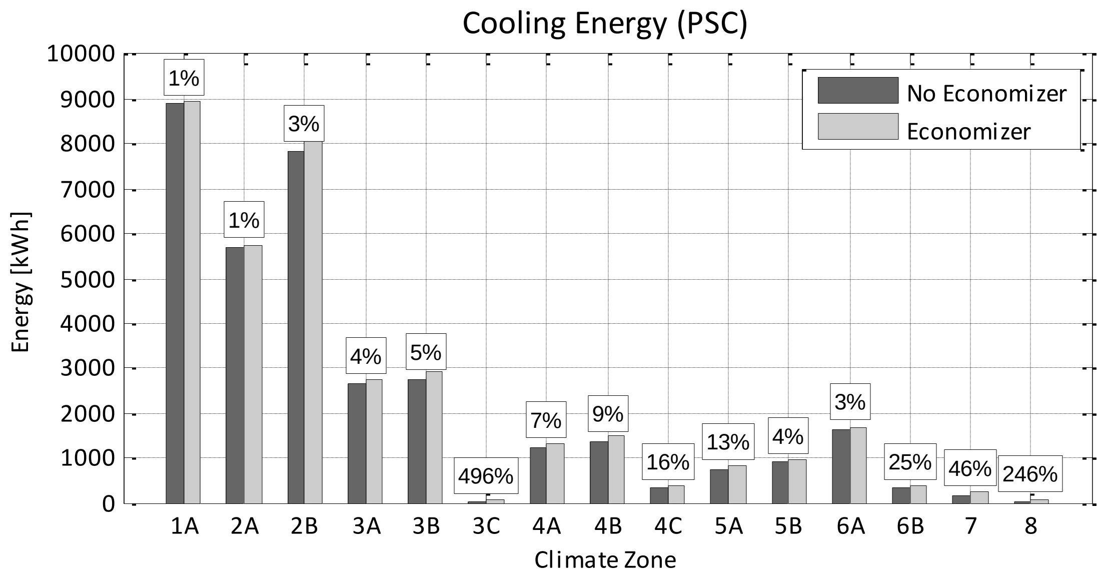


Figure 6: Annual cooling energy with and without an economizer in all climate zones (PSC air handler)

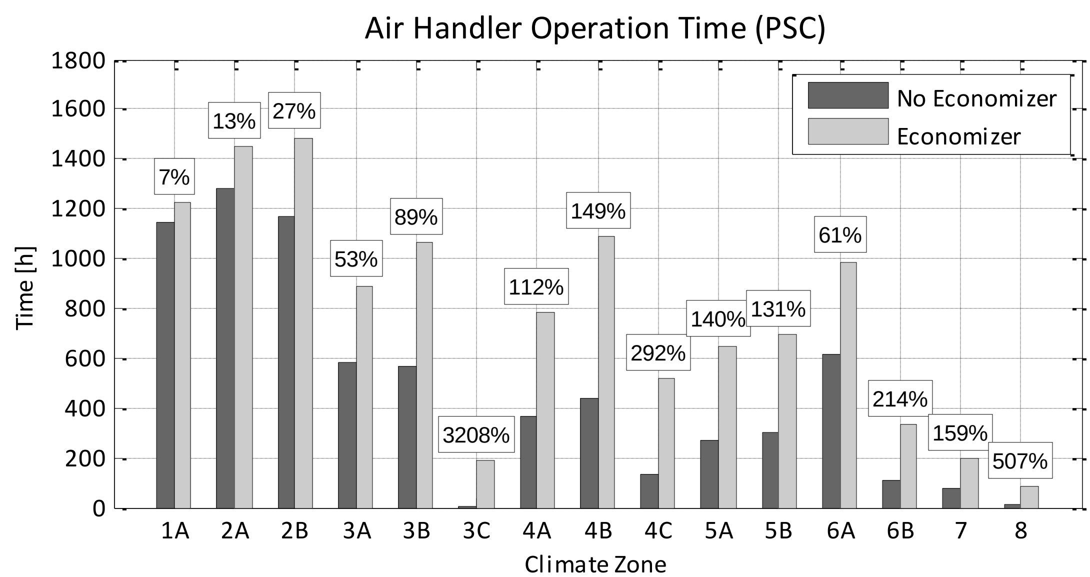


Figure 7: Annual air handler operation time with and without an economizer in all climate zones (PSC air handler)

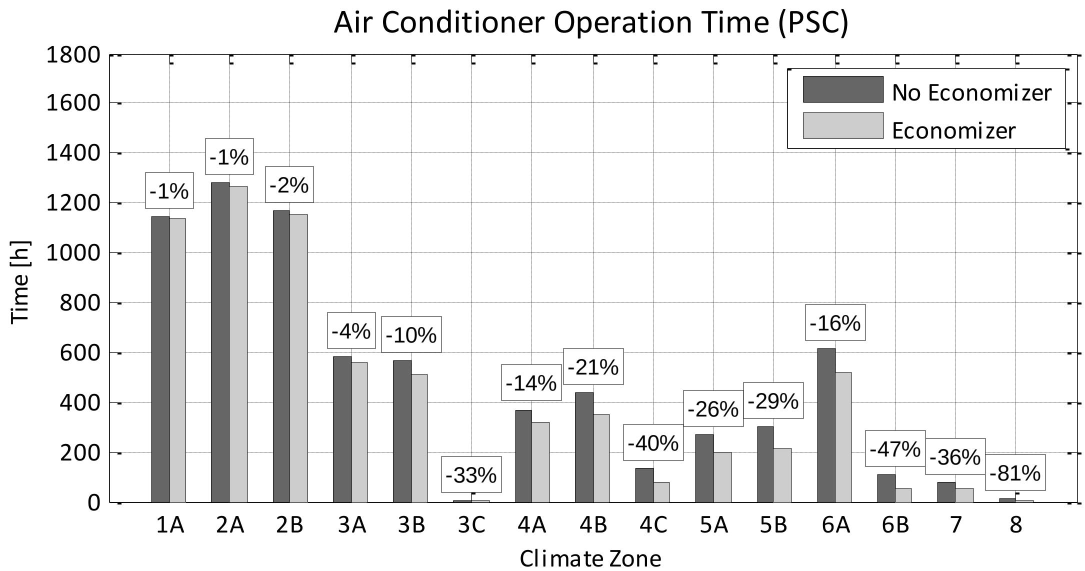


Figure 8: Annual air conditioner operation time with and without an economizer in all climate zones (PSC air handler)

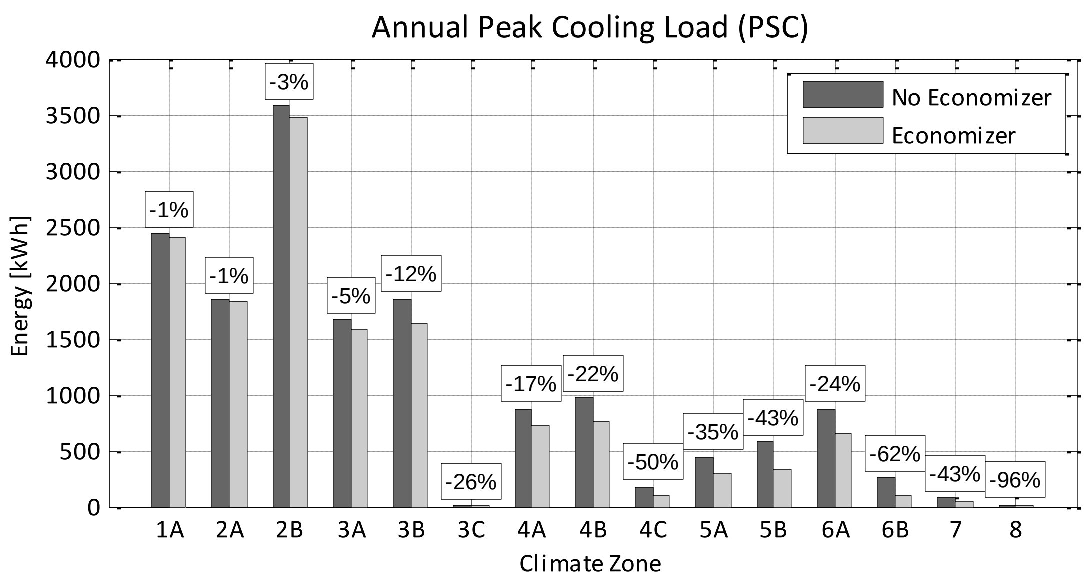


Figure 9: Annual peak cooling load with and without an economizer in all climate zones (PSC air handler)

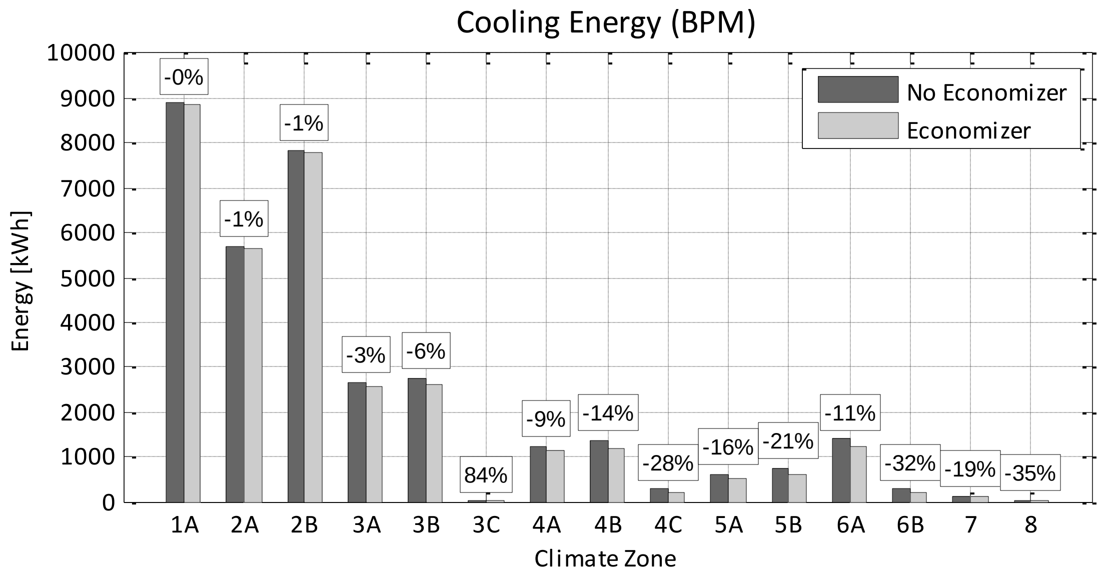


Figure 10: Annual cooling energy with and without an economizer in all climate zones (BPM air handler)

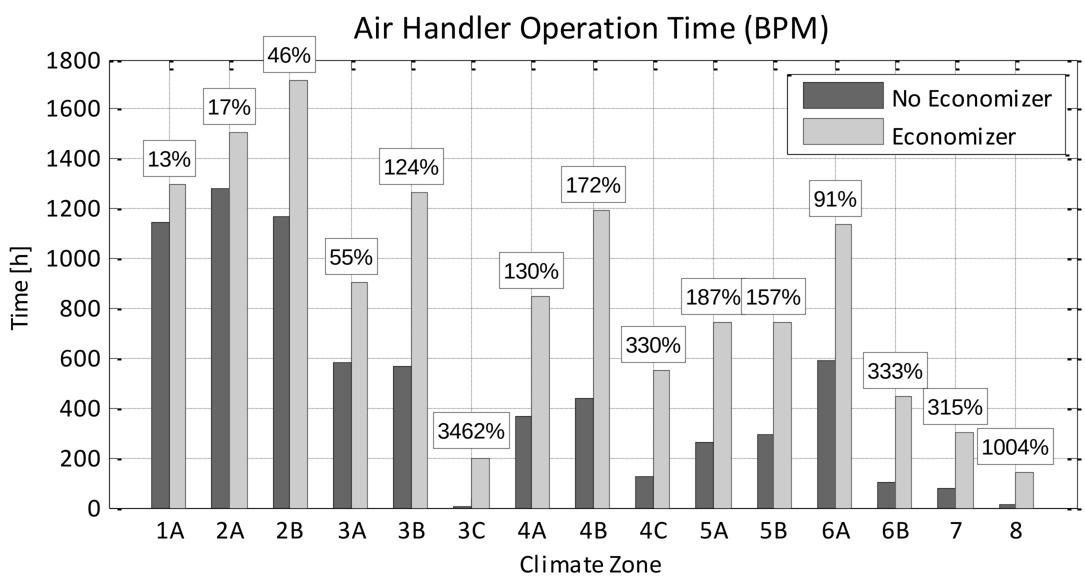


Figure 11: Annual air handler operation time with and without an economizer in all climate zones (BPM air handler)

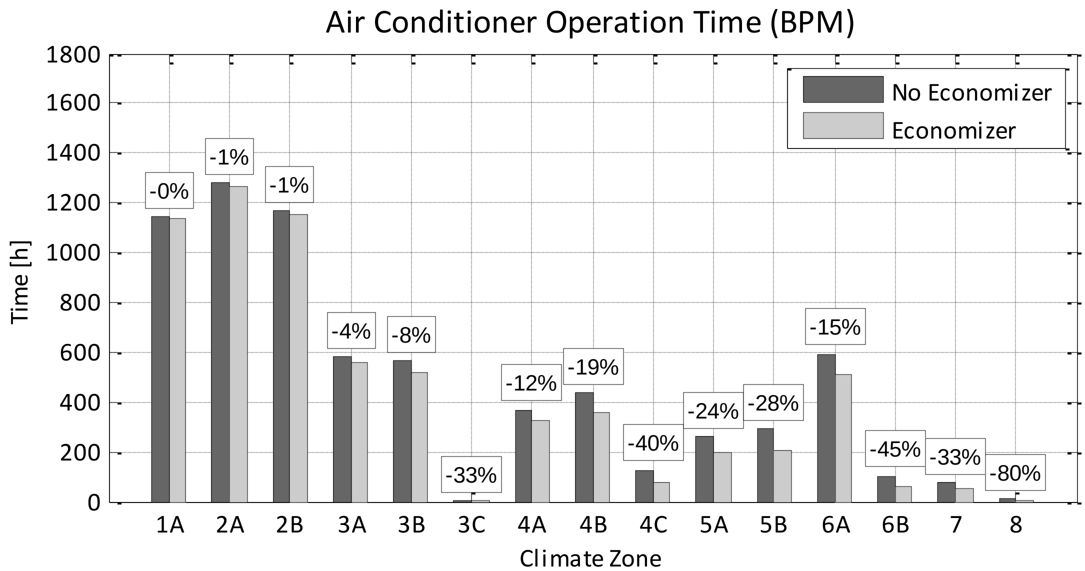


Figure 12: Annual air conditioning operation time with and without an economizer in all climate zones (PSC air handler)

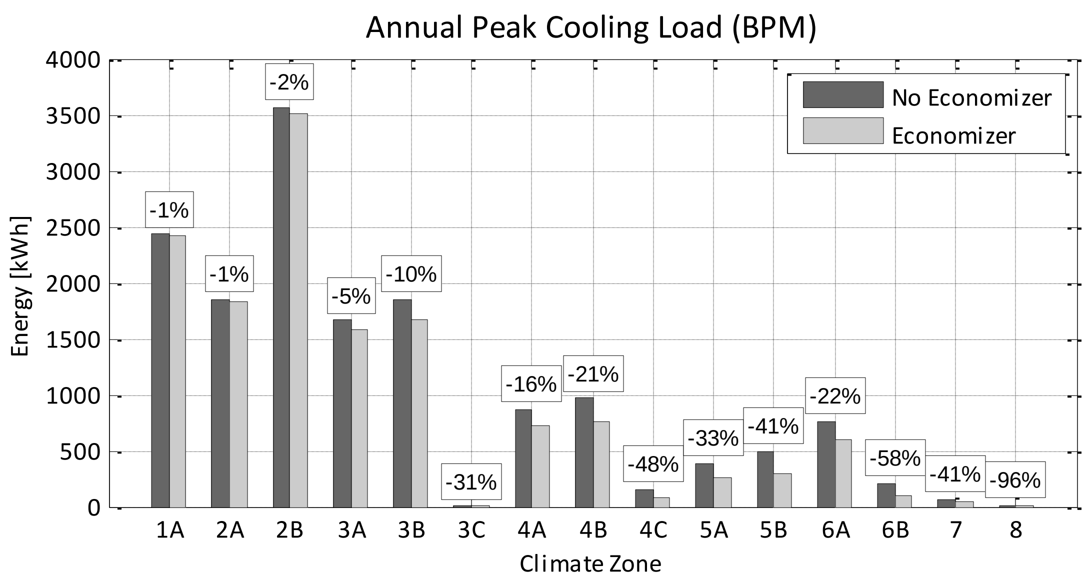


Figure 13: Annual peak cooling load with and without an economizer in all climate zones (PSC air handler)

Generally there are small increases in cooling energy use (on the order of 100 kWh/y) for PSC air handler economizers and similar magnitude decreases for BPM air handler economizers. This is mostly due to the lower fan power requirements for BPM economizer operation. The largest cooling energy savings are in climate zones 3, 4, 5, and 6. The reductions in AC operation are almost the same (within a few percent) for both motor types. These results indicate that the additional fan power required to operate the economizer offsets the cooling energy savings, and that the type of air handler in the system has a significant effect on energy savings (because the PSC air handler uses more power than the BPM air handler). This is further reinforced by the large increases in air handler operating time which more than double in some climates. The climates zones with the biggest increase in air handler operating time tend to be those with the greatest savings. Climate zones 1 and 2 that show small changes in cooling energy use also have relatively small increases in air handler operating time, because these climates have a relatively small amount of time where outdoor conditions are favorable to providing cooling. The small overall cooling energy savings also indicate that there may be too much operation of the economizer during times of small indoor-outdoor temperature differences when the cooling savings are marginal.

Peak load reductions occur when the house is cooled by economizer operation sufficiently below its set point early in the day that there is a reduction in AC operation later in the day at times of utility peak loads. The PSC air handler does show peak load reductions that are significant in some climates, for example 12% in climate zone 3B. For the BPM economizer the fractional peak load reductions are about 50% greater than the cooling energy savings and the results are within a few percent of the PSC peak load reduction results.

Duct leakage affects the energy gains and losses from ducts. Supply duct losses represent a loss of cooling energy directly to outside. Return leaks are more complex because the effects depend on the air temperature in the attic which is highly dynamic depending on solar loads, radiation to the sky and ventilation. To examine sensitivity to duct leakage the simulations were repeated with zero duct leakage to eliminate these effects. The results are summarized in Appendix A.

The indoor set point also influences the economizer effects by changing both the times of operation (because using the same indoor-outdoor control means that different outdoor temperatures are acceptable for economizer operation) and the amount of cooling the home requires. To examine this effect the simulations were repeated with fixed indoor economizer operation set points of 23.3°C (74 °F) and 25.6°C (78 °F), only above which would the economizer run. The results (summarized in Appendix A) show that for the 23.3°C (74 °F) set point the PSC results have a smaller annual cooling energy penalty and the BPM has reduced cooling energy savings. For the 25.6°C (78 °F) set point economizer operation was severely reduced, with air handler operating time changes on the order of 1% for both air handler types. This resulted in almost no impact of economizer use as it operated for so little of the time.

Indoor Humidity

Figure 14 and Figure 15 show the number of hours in a year that the indoor RH exceeds 70% for the PSC and BPM economizers respectively. As expected, the number of high humidity hours roughly scale with the outdoor humidity in each climate. Generally, the effect of the economizer is to raise the indoor humidity. Interesting exceptions occur in climates 1A and 3C where the economizer reduces the RH. For 1A the decrease in the number of hours with RH over 70% is because the economizer only operates during cooler shoulder seasons that are coincident with lower outdoor humidity, whereas on hotter more humid nights in the summer the economizer does not operate. In climate zone 3C, which has the most high-humidity hours, there is also a reduction in high-humidity events for the same reason - the economizer operates when it is less humid outdoors than indoors. In the climates where the economizer causes the greatest increases in high-humidity hours, there are still less than 2000 hours a year where the RH is above 70%.

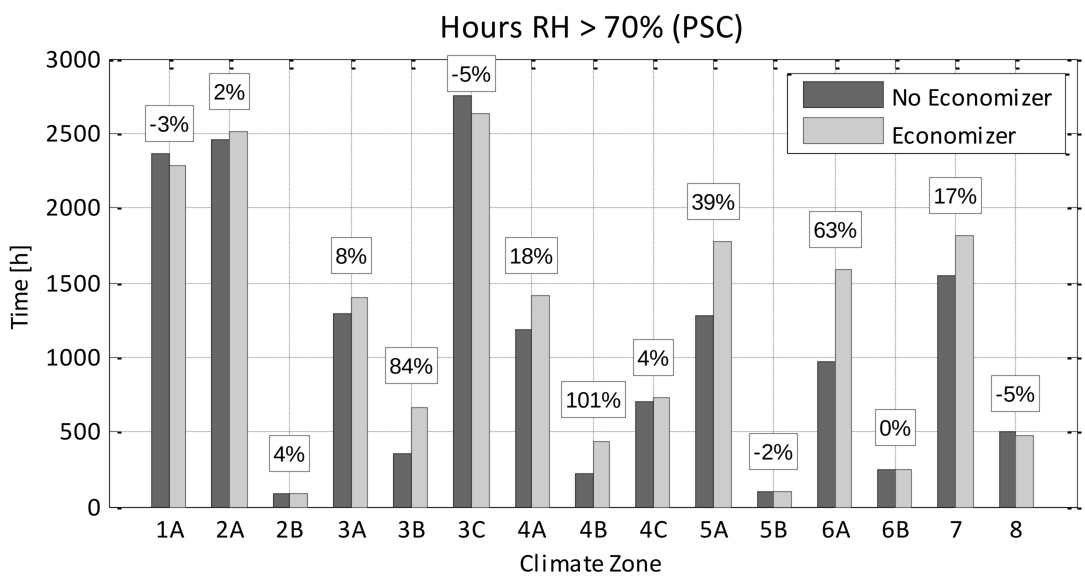


Figure 14: Number of hours in the year when the relative humidity exceeds 70%, with and without an economizer (PSC air handler)

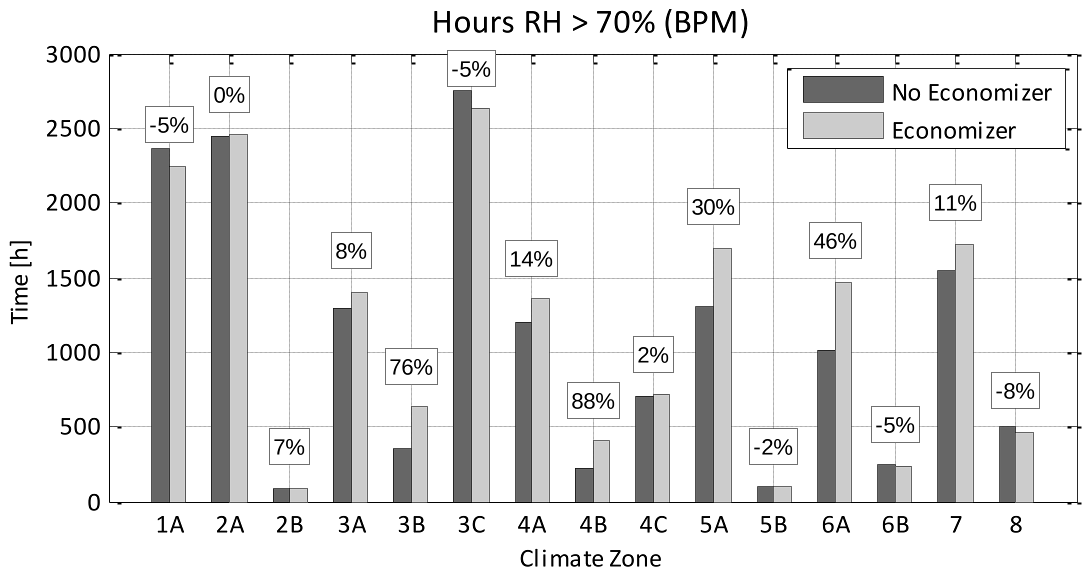


Figure 15: Number of hours in the year when the relative humidity exceeds 70%, with and without an economizer (BPM air handler)

To further illustrate the reasons for the relatively low impact of economizer operation , Figure 16 shows the *daily averaged* relative humidity (RH), while Figure 17 shows the *weekly averaged* RH. These timescales were used because the effects of humidity depend on duration as well as magnitude. Figure 16 shows that variations can be large over shorter timescales of 24-hours – but these short-term high humidity events do not lead to mold growth or structural issues and so are of little consequence other than occupant comfort. Comparing Figure 16 to Figure 17, Figure 17 shows more clearly when long-term high humidity events occur – particularly those above 70% in Houston (2A) and Memphis (3A) during shoulder seasons, and in the winter in Miami (1A) when the air conditioning is not operating to provide any dehumidification. Hot-dry climates with low annual average humidity in El Paso and Albuquerque (3B and 4B) do not show these shoulder seasons. Instead, their higher indoor RH is driven by increases in indoor humidity. Other climates such as Baltimore (4A) and Chicago (5A) show large humidity swings from winter to summer due to the large changes in ambient weather conditions for these locations. Zones 4A and 5A have average annual humidity between the 'hot-humid' and 'hot-dry' climates and have almost as many occurrences of prolonged high humidity events as the classically humid climates of Houston (2A) and Miami (1A). Climate zone 3C shows high RH but this is an artifact of low nighttime temperatures, and also a complete lack of dehumidification from air conditioning.

Figure 18 shows the economizer operating hours for each month. In the five hottest climates (1A-3B) it is too hot at night for economizer operation in the summer time and they only operate in the shoulder seasons. In more moderate climates the economizer only operates when indoor temperatures are high enough – i.e. in the summer.

Overall, these indoor humidity results show that humidity impacts are generally small - particularly in the hot humid climates that are generally of concern for economizer use. The use of an economizer can lead to one-week periods of high indoor humidity in less humid climates; however, it is not clear if this represents a clear humidity problem because the events are only one week in duration and they only occur a couple of times during the year. To examine this further ASHRAE Standard 160 was used which gives the following time/RH based criteria:

- To avoid mold growth requires a 30-day running average RH < 80%, a 7-day running average RH < 98% and a 24-h average RH < 100%.
- To avoid corrosion requires a 30-day running average surface RH < 80%.
- To avoid condensation on windows avoid > 24-h events.

Given that the high levels of indoor humidity occur when outdoor temperatures are generally higher than indoors, then the 70% RH upper limit that is being used should have surface temperatures lower than 70% RH and the results are far within the ASHRAE 160 criteria. The exceptions may be winter in Miami and the shoulder seasons in Houston, which is when the economizer operates. Note that Figure 14 shows that the economizer does not increase the number of hours with high humidity in Miami. This is because the high humidity events occur when the outdoor temperatures are cool and outdoor humidity is low (relative to the rest of the year), so the events are due to a lack of air-conditioner operation (and hence lack of dehumidification) and not due to an economizer bringing in large quantities of outdoor air. In Houston, the economizer operation is more directly linked to elevated indoor humidity because the economizer operates during the shoulder seasons when there is little air conditioner operation, but the outdoor air is still fairly humid. In some ways the economizer is a victim of its own success in this situation because it reduces air conditioner operation (thus saving energy) but this, in turn leads to less dehumidification. The overall conclusion is that the performance evaluation criteria in ASHRAE Standard 160 indicate that the high humidity events in our results are not problematic.

It should be noted that these results are for a light-weight timber frame building. Heavier construction buildings such as brick will have higher thermal mass and moisture storage potential. This would act to reduce the peak levels of high humidity throughout the year.

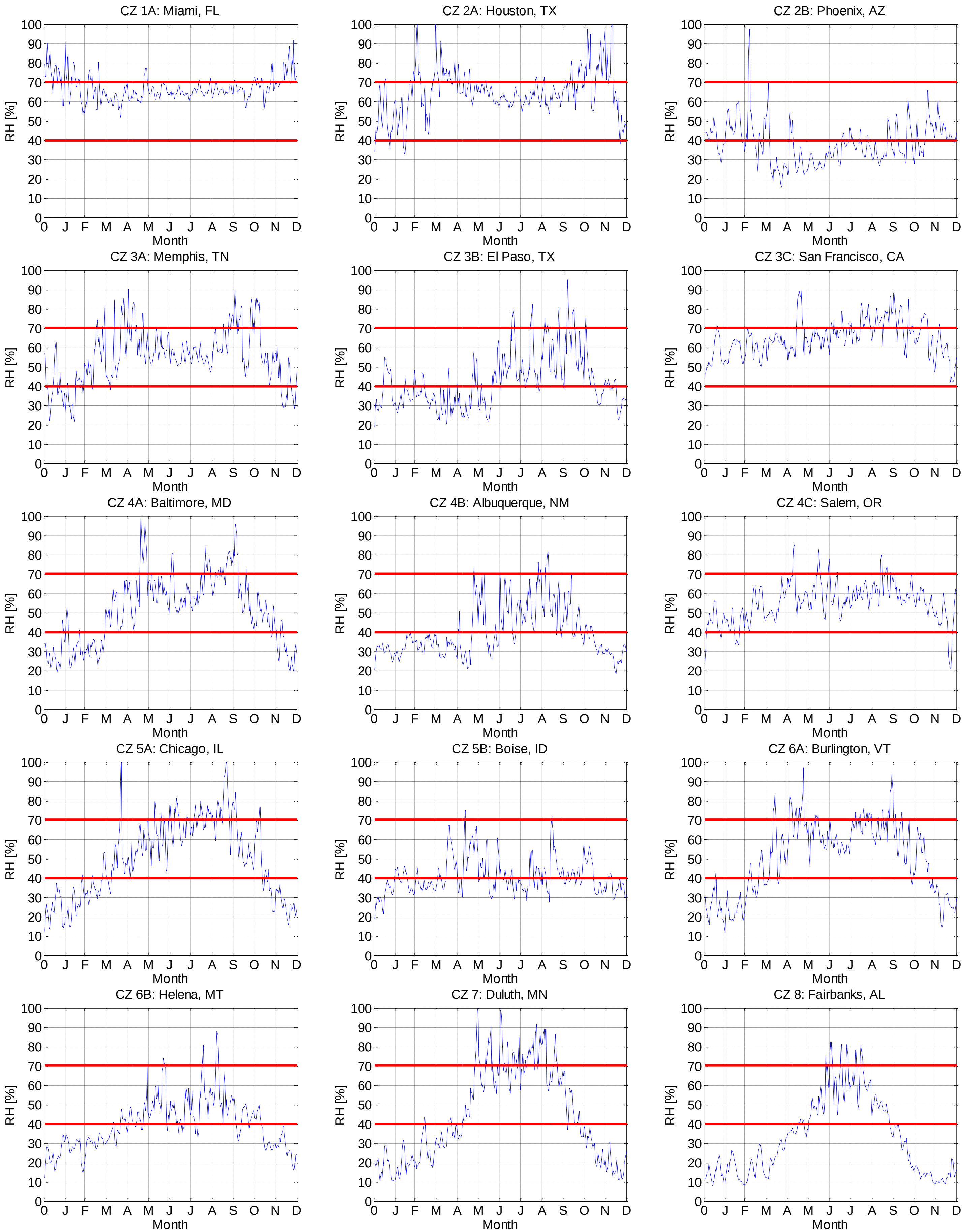
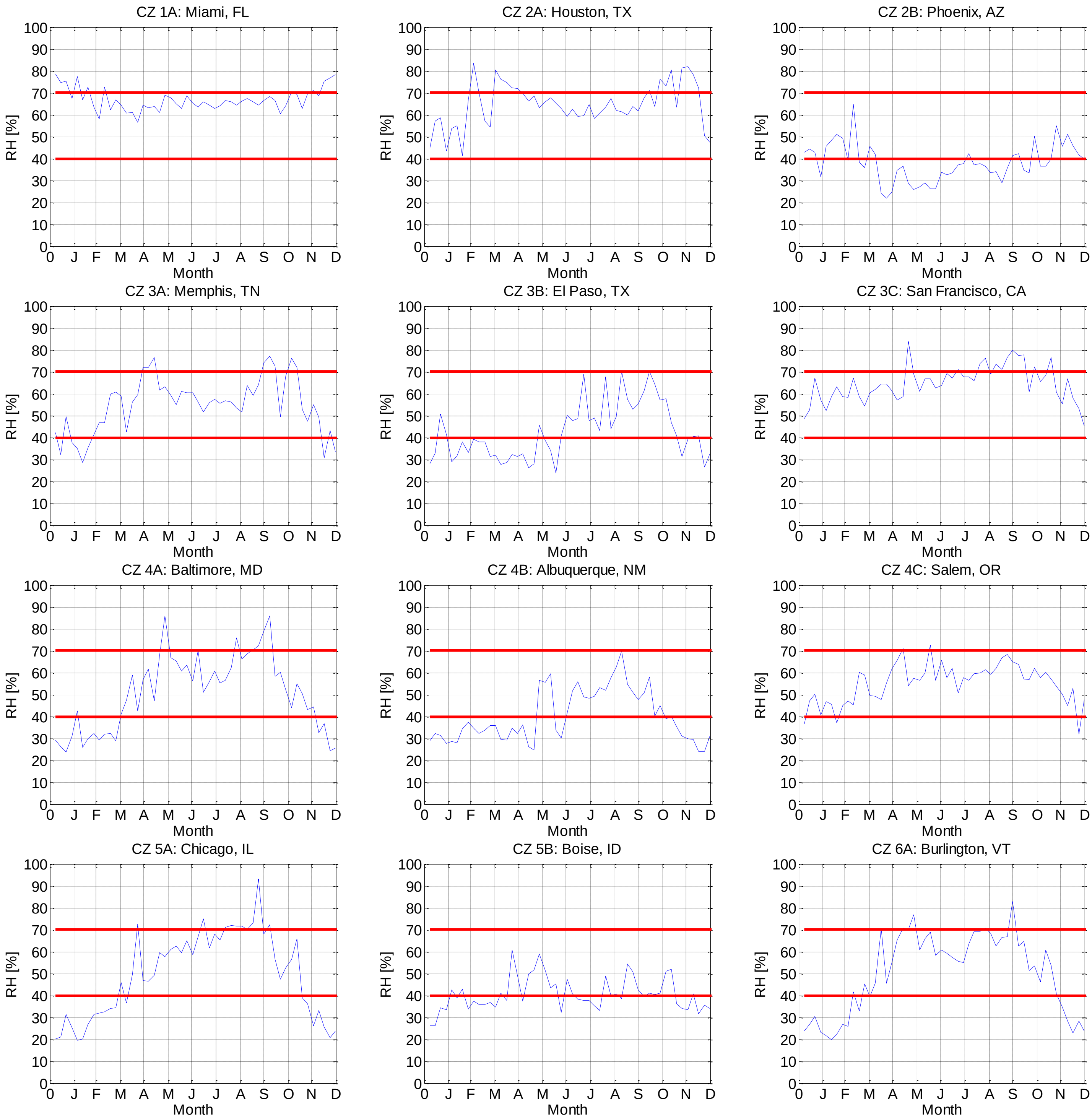

CZ 1A: Miami, FL
CZ 2A: Houston, TX
CZ 2B: Phoenix, AZ
CZ 3A: Memphis, TN
CZ 3B: El Paso, TX
CZ 3C: San Francisco, CA
CZ 4A: Baltimore, MD
CZ 4B: Albuquerque, NM
CZ 4C: Salem, OR
CZ 5A: Chicago, IL
CZ 5B: Boise, ID
CZ 6A: Burlington, VT
CZ 6B: Helena, MT
CZ 7: Duluth, MN
CZ 8: Fairbanks, AL
RH [%]
Month
0 J F M A M J J A S O N D

Figure 16: Daily average indoor relative humidity (RH). Red lines show 40% and 70%. Thick black dashed lines show the annual mean



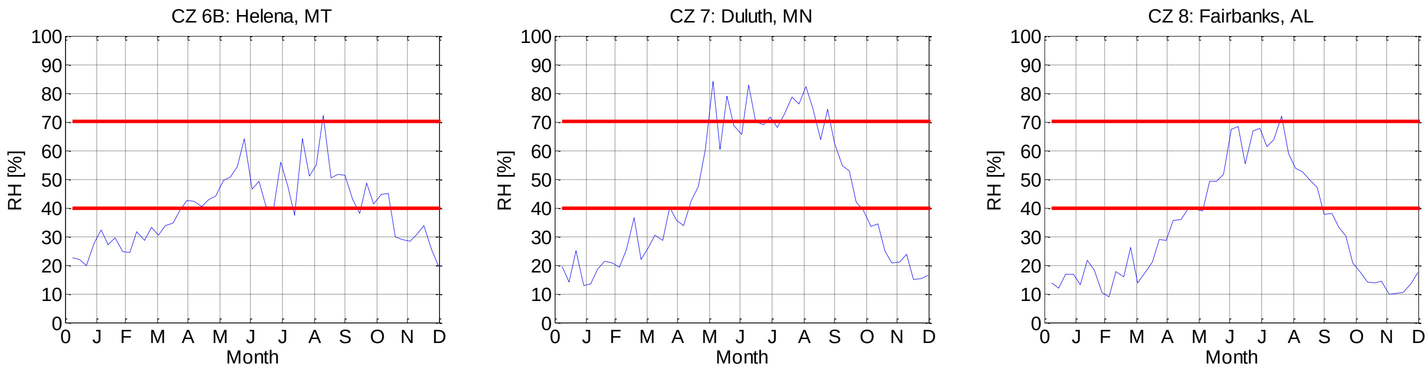
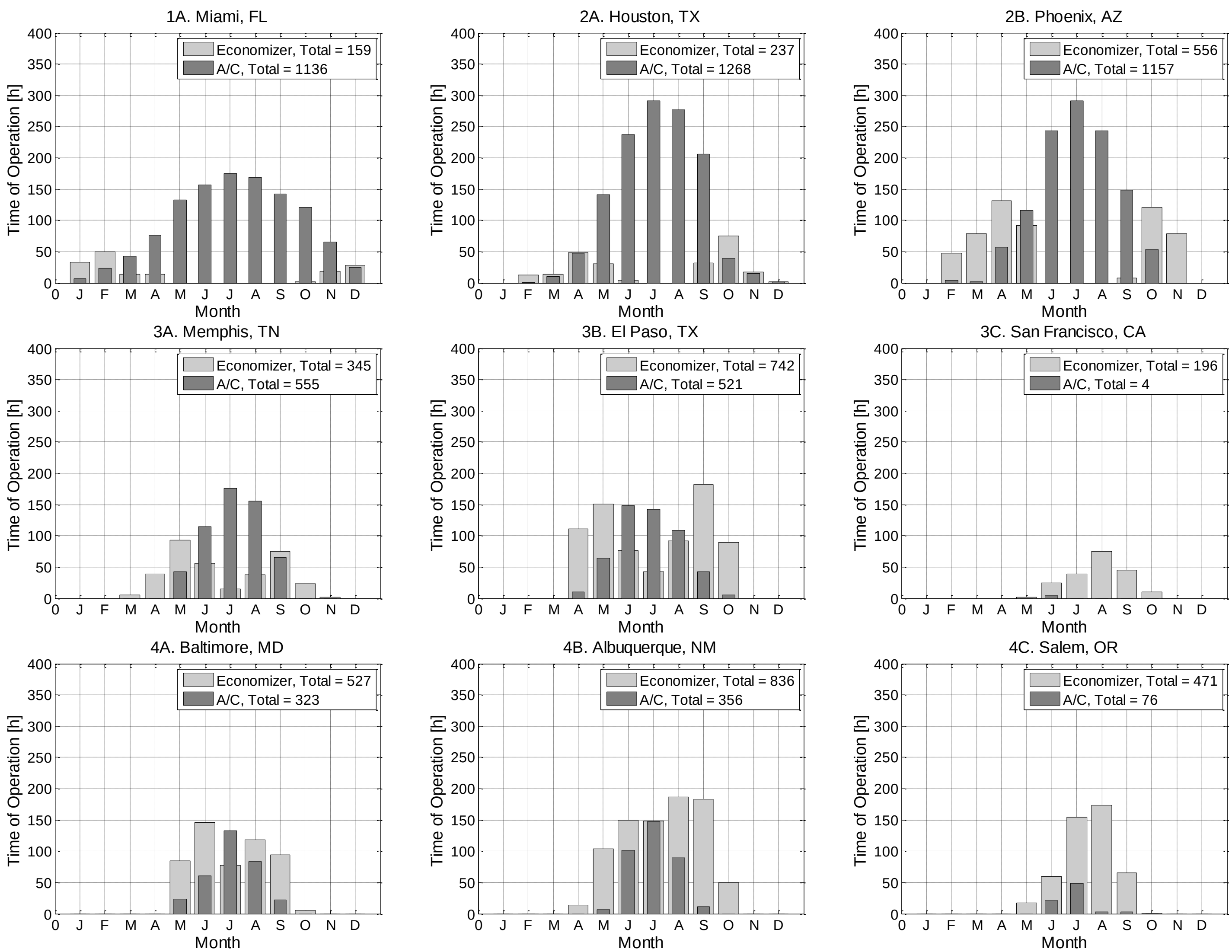


Figure 17: Weekly average indoor relative humidity (RH). Red lines show 40% and 70%. Thick black dashed lines show the annual mean.



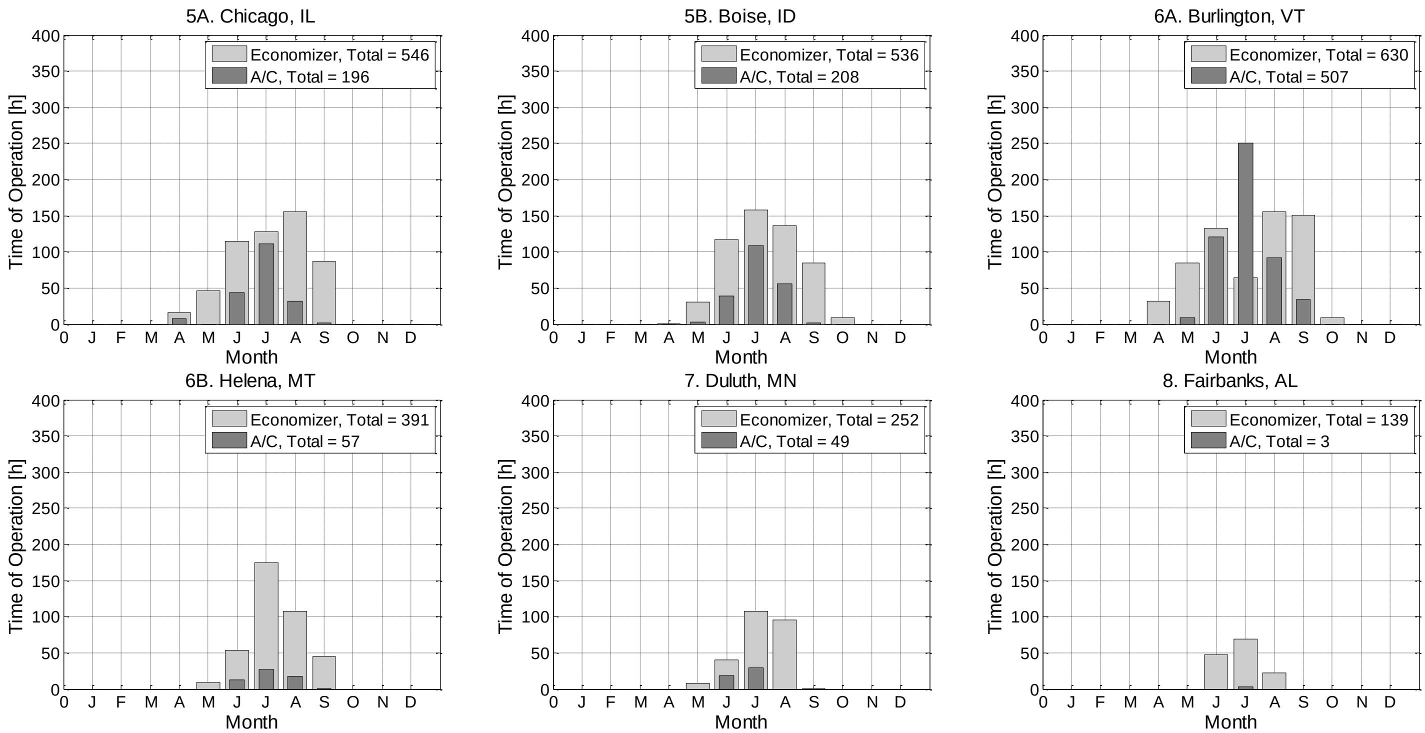


Figure 18: Economizer (BPM) and air conditioning operation hours over the year. Annual totals are contained in the key.

Conclusions & Recommendations

Residential economizers have annual energy savings up to 200 kWh/year of cooling energy in climates with large diurnal temperature swings (Climate zones 3, 4, 5 & 6, but not 1 or 2) for systems with BPM air handlers. For PSC air handlers, the extra operating time led to overall increases in energy use. These different results between motor types illustrate that caution must be employed when using ventilation cooling to ensure that the central forced air system blower does not consume too much power - either by motor design or from overly restrictive duct work. The moderate annual energy savings are due to fan energy increases associated with economizer operation which us why the lower power BPM air handlers show greater energy savings. It is possible that greater savings would be achieved if the central forced sir system were designed and installed to have very low air flow resistance with a resulting reduction in required blower power below the typical values used in this study. Both motor types had greater reductions in peak loads than energy savings, with about 10% peak load reductions in Climate zones 3A and 3B

In the hottest, most humid climate zone 1A economizer operation reduced the incidence of high humidity events. In Climate Zones 2A and 3A the humidity effects were almost negligible. These results are because time of operation of the economizer is not coincident with high outdoor humidity, e.g., economizers would only operate in shoulder seasons when outdoor humidity is lower than indoor humidity. Generally, the economizer does not operate when doing so would bring into the house air that is more humid than indoor air.

The large ventilation air flows induced y economizer operation result in reduction in relative exposures of about 40%. This represents a significant increase in relative indoor air quality.

One caveat with the results of this study is that the results are for lightweight wooden-framed homes. The results may change for more thermally massive brick/block structures.

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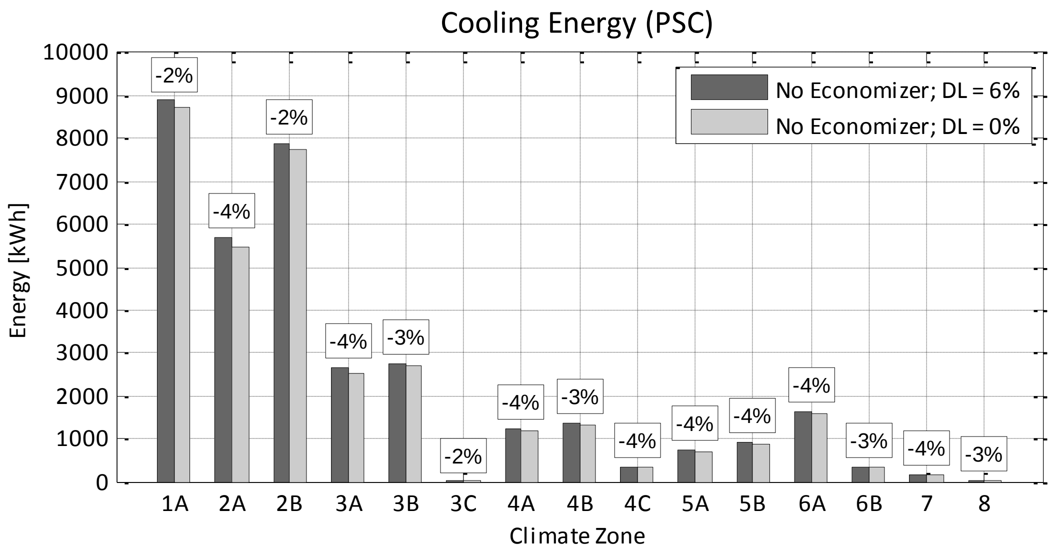
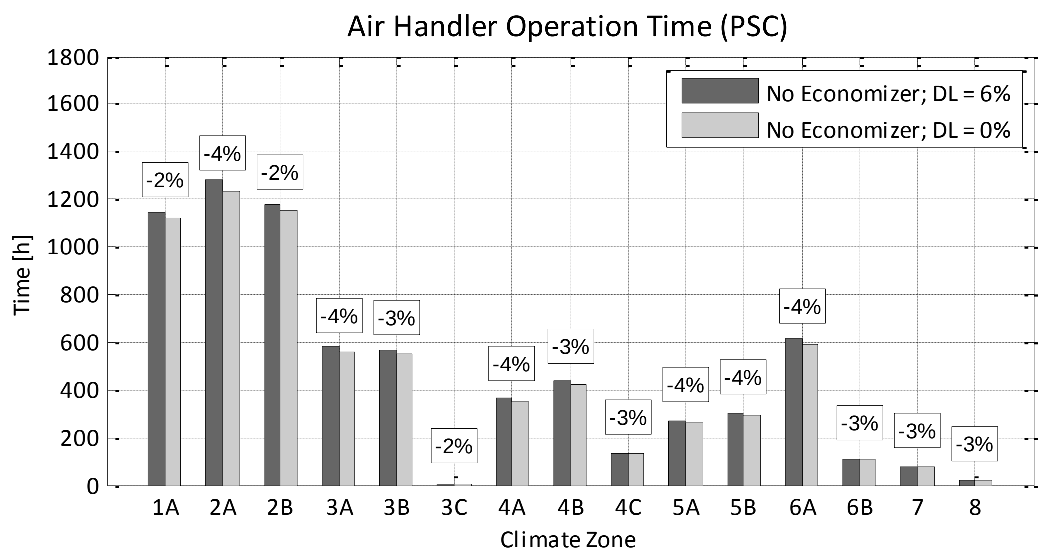
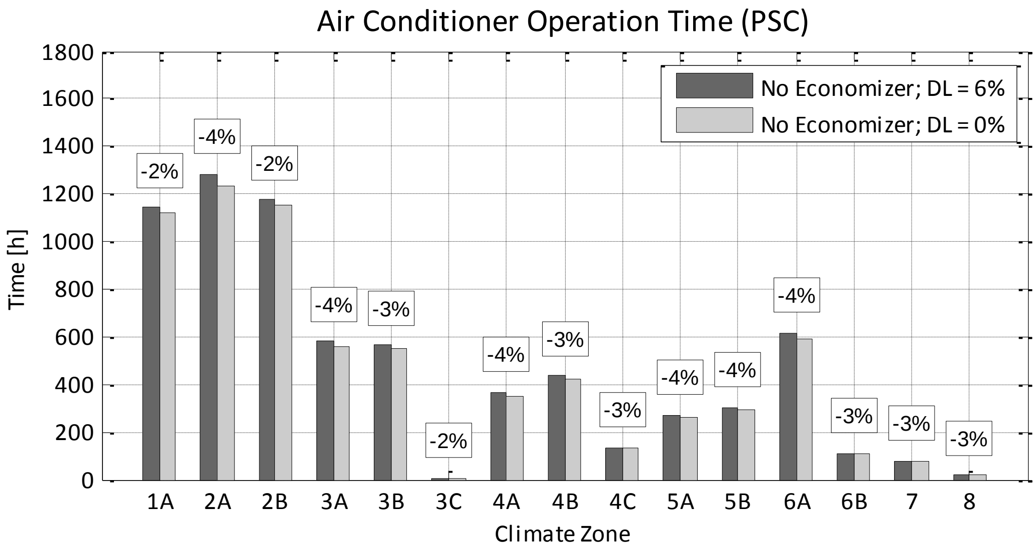
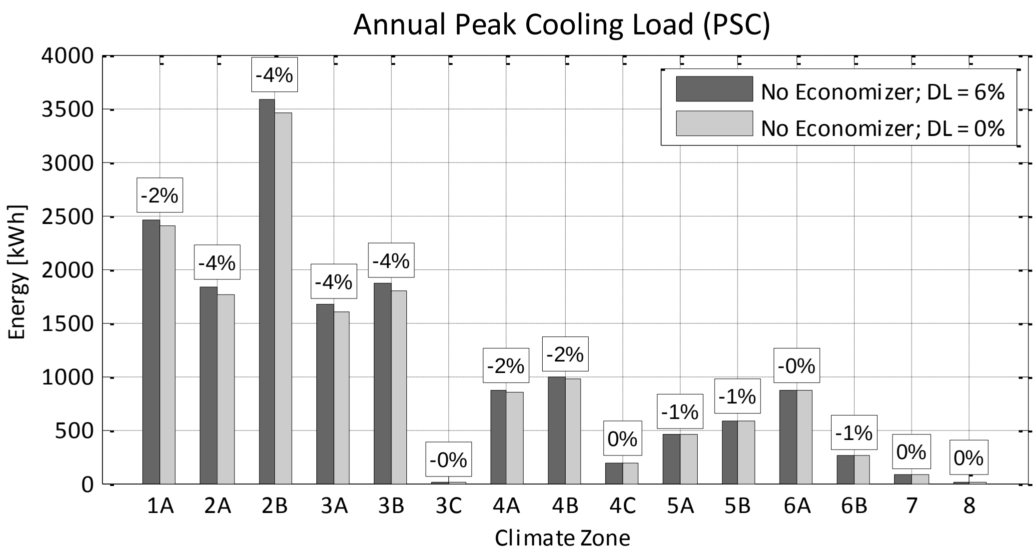
Appendix A: Effect of duct leakage and different indoor set points on simulation results

PSC Economizer – 6% and 0% Duct Leakage

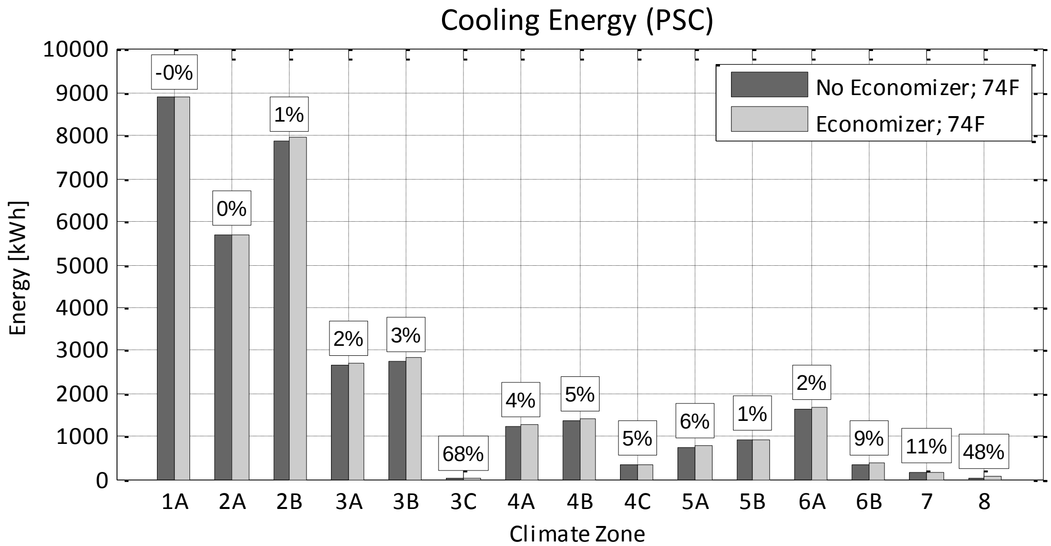


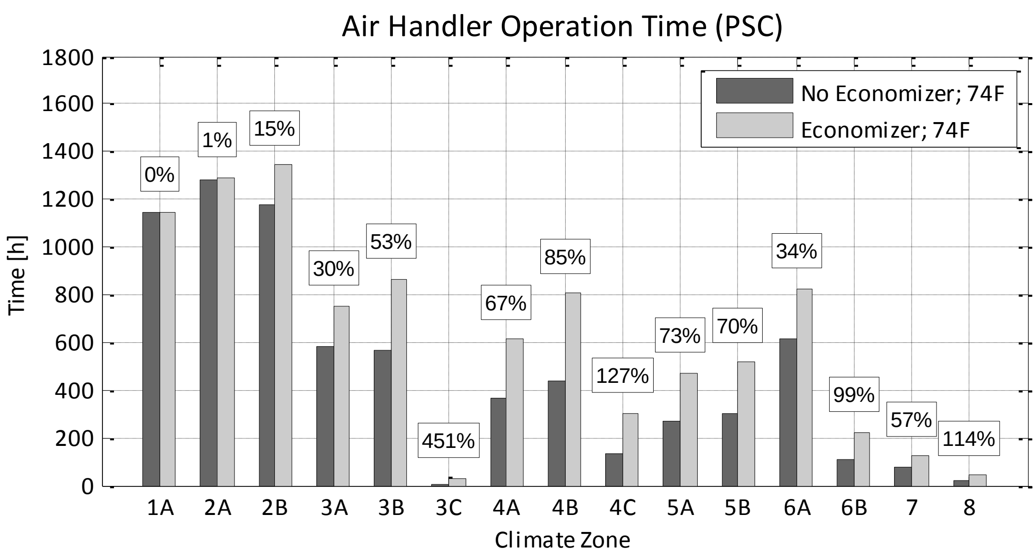


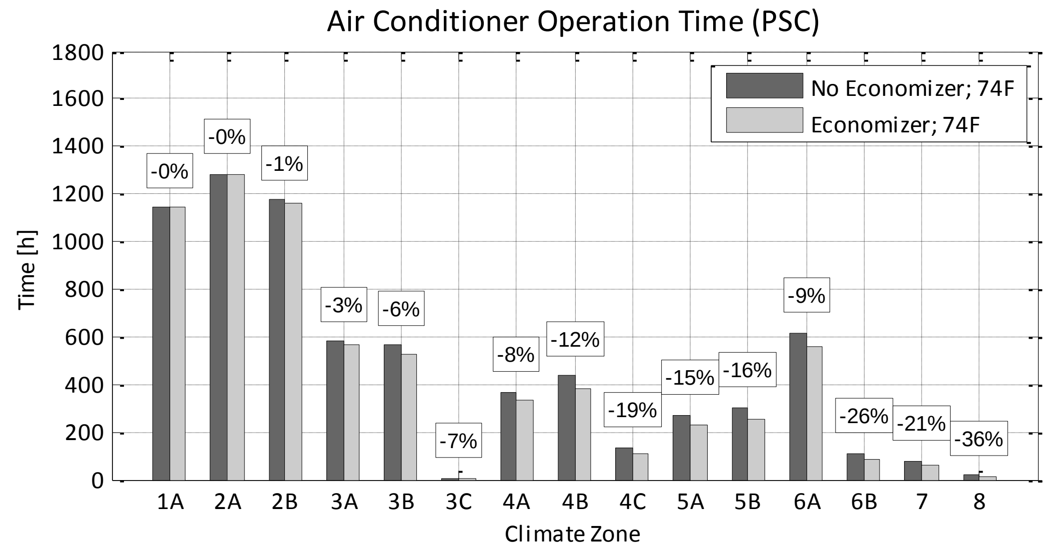

Air Conditioner Operation Time (PSC)
Time [h]
Climate Zone
No Economizer; DL = 6%
No Economizer; DL = 0%
1A 2A 2B 3A 3B 3C 4A 4B 4C 5A 5B 6A 6B 7 8
-2% -4% -2% -4% -3% -2% -4% -3% -3% -4% -4% -4% -3% -3% -3%
0 200 400 600 800 1000 1200 1400 1600 1800


Annual Peak Cooling Load (PSC)
Energy [kWh]
Climate Zone
No Economizer; DL = 6%
No Economizer; DL = 0%
1A 2A 2B 3A 3B 3C 4A 4B 4C 5A 5B 6A 6B 7 8
-2% -4% -4% -4% -4% -0% -2% -2% 0% -1% -1% -0% -1% 0% 0%
0 500 1000 1500 2000 2500 3000 3500 4000

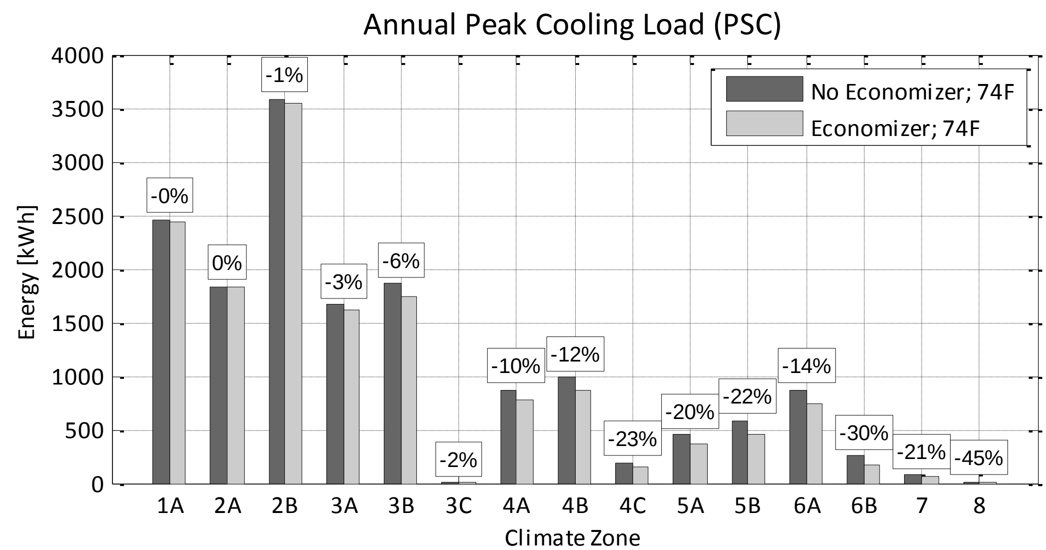
PSC Economizer – 74°F (23.3°C) Economizer Set Points



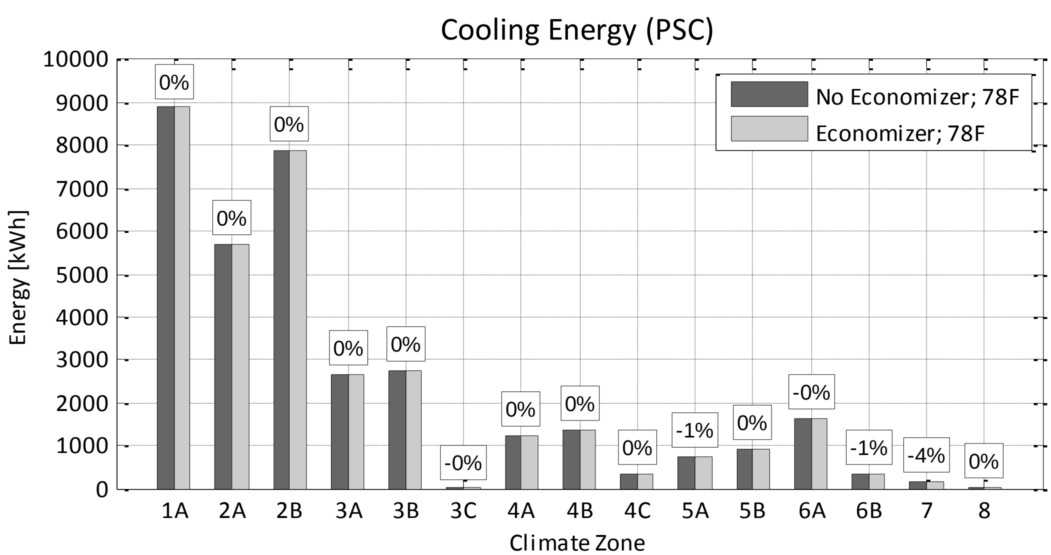




Air Conditioner Operation Time (PSC)
Time [h]
Climate Zone
No Economizer; 74F
Economizer; 74F
1A 2A 2B 3A 3B 3C 4A 4B 4C 5A 5B 6A 6B 7 8
-0% -0% -1% -3% -6% -7% -8% -12% -19% -15% -16% -9% -26% -21% -36%

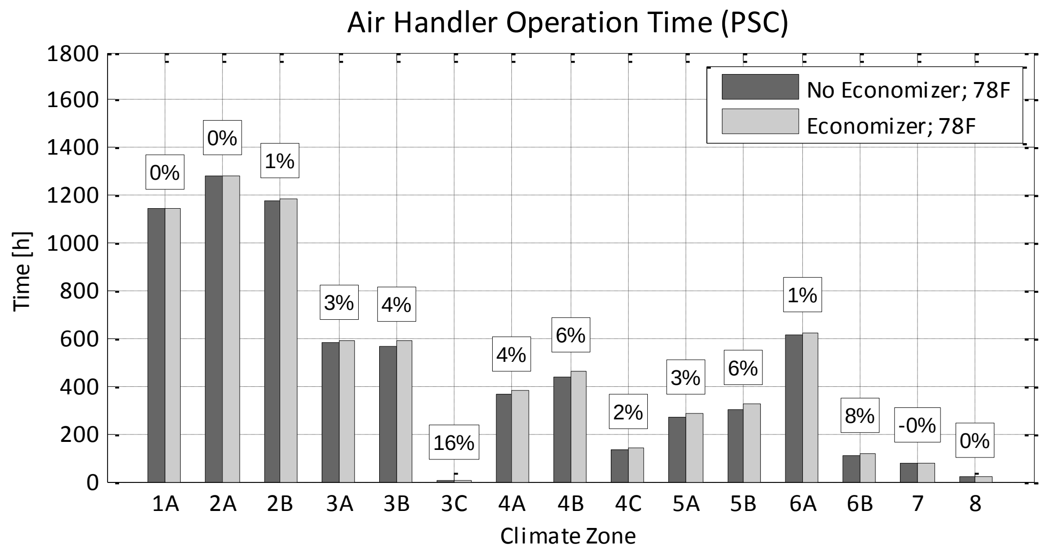


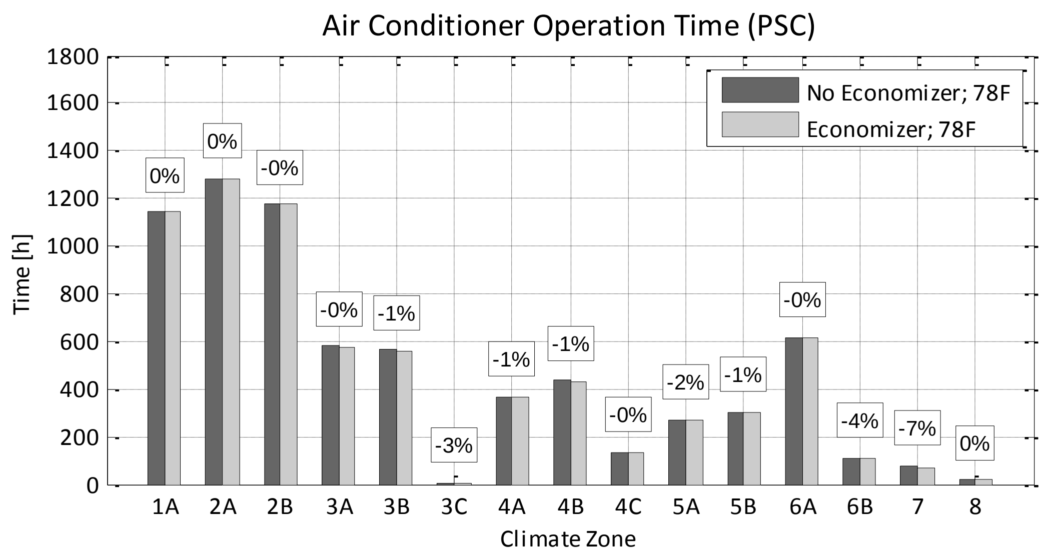
Annual Peak Cooling Load (PSC)
Energy [kWh]
Climate Zone
No Economizer; 74F
Economizer; 74F
1A 2A 2B 3A 3B 3C 4A 4B 4C 5A 5B 6A 6B 7 8
-0% 0% -1% -3% -6% -2% -10% -12% -23% -20% -22% -14% -30% -21% -45%

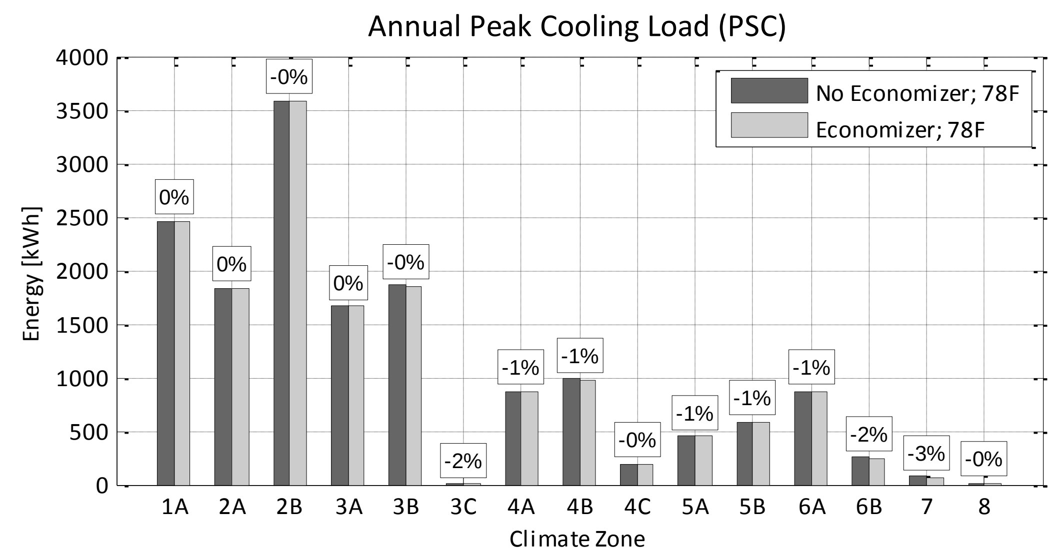


Cooling Energy (PSC)
Energy [kWh]
Climate Zone
No Economizer; 78F
Economizer; 78F
1A 2A 2B 3A 3B 3C 4A 4B 4C 5A 5B 6A 6B 7 8
0% 0% 0% 0% 0% -0% 0% 0% 0% -1% 0% -0% -1% -4% 0%

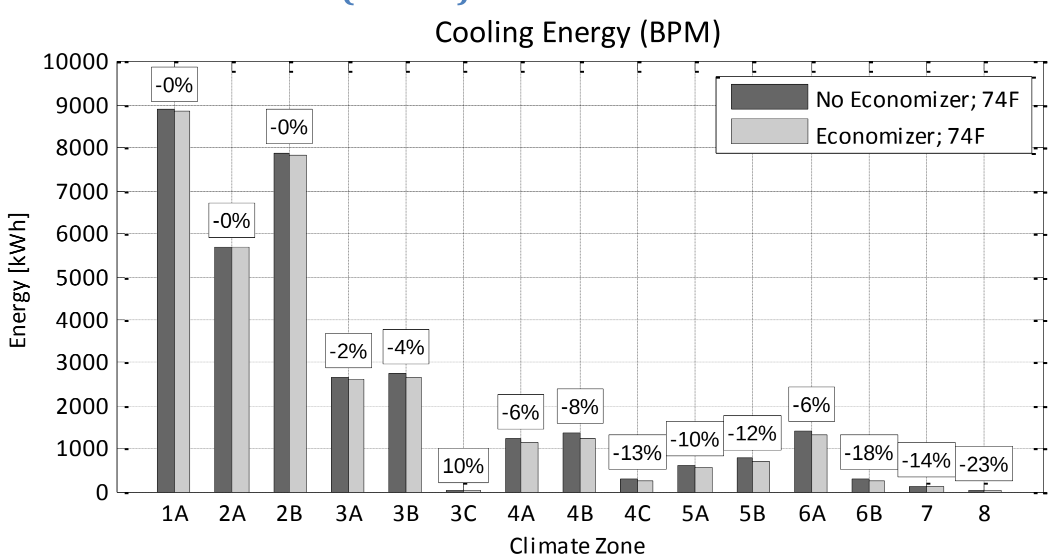
PSC Economizer – 78°F (25.6°C) Economizer Set Points

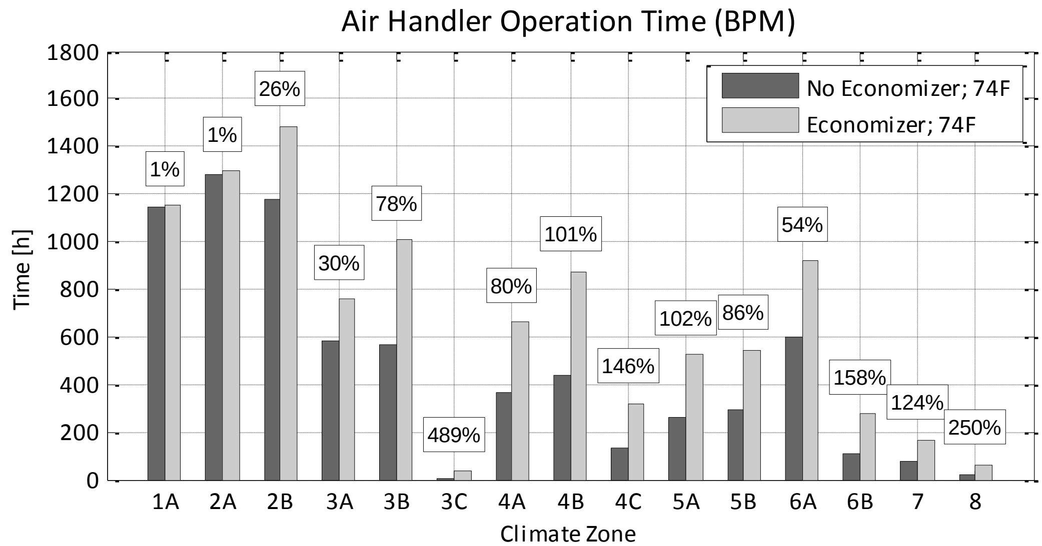


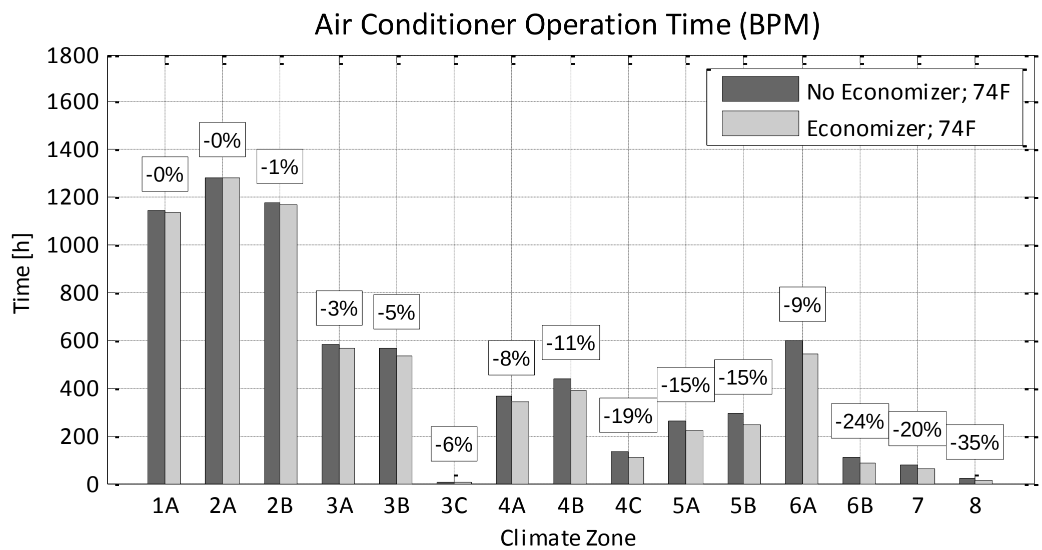


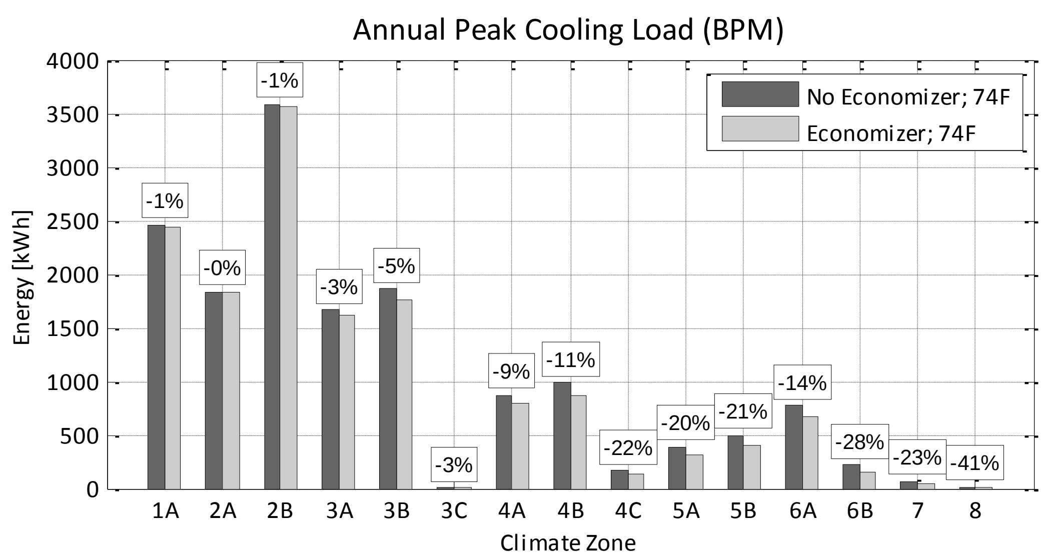


BPM Economizer - 74°F (23.3°C) Economizer Set Points

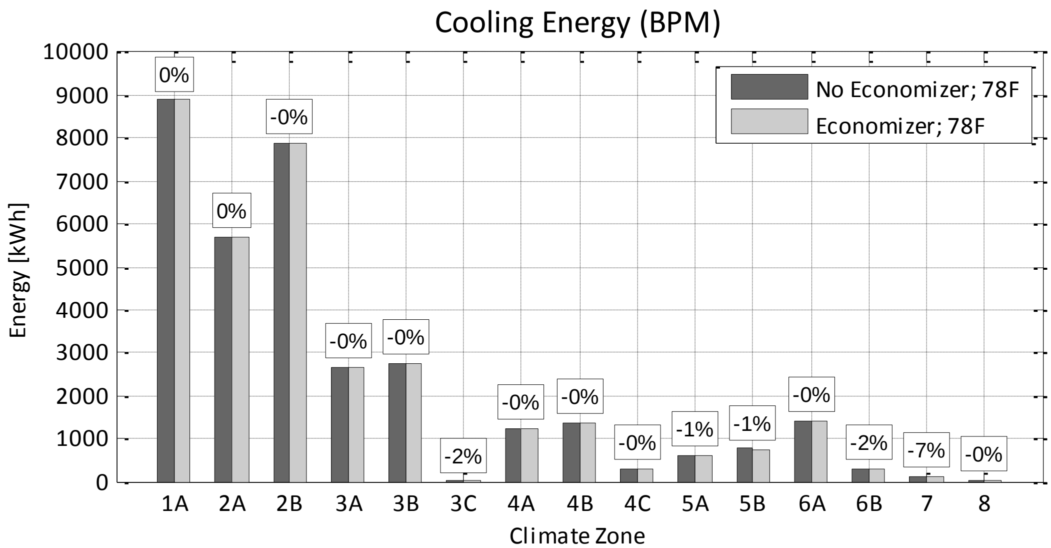


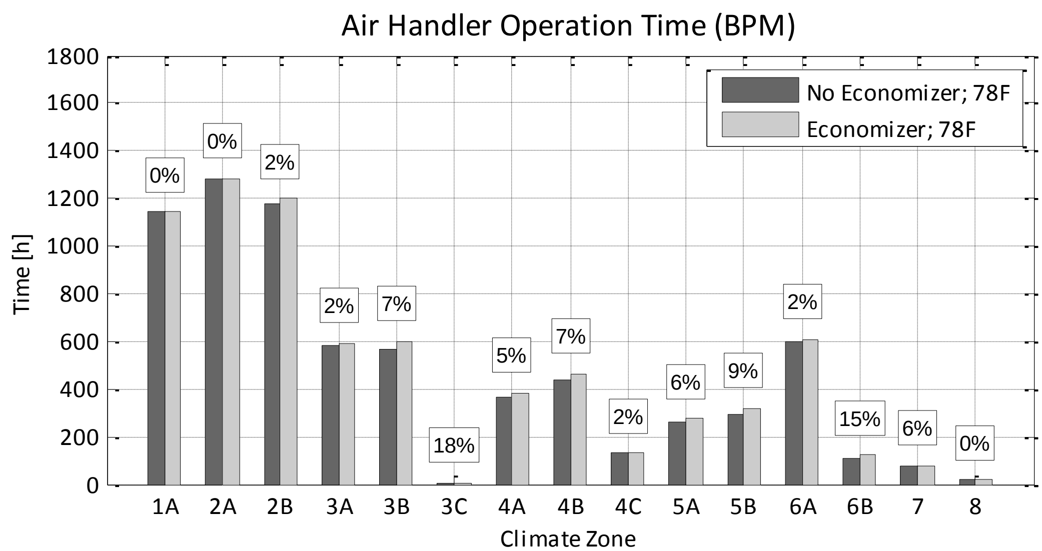

Air Handler Operation Time (BPM)
Time [h]
Climate Zone
No Economizer; 74F
Economizer; 74F
1A 2A 2B 3A 3B 3C 4A 4B 4C 5A 5B 6A 6B 7 8
1% 1% 26% 30% 78% 489% 80% 101% 146% 102% 86% 54% 158% 124% 250%

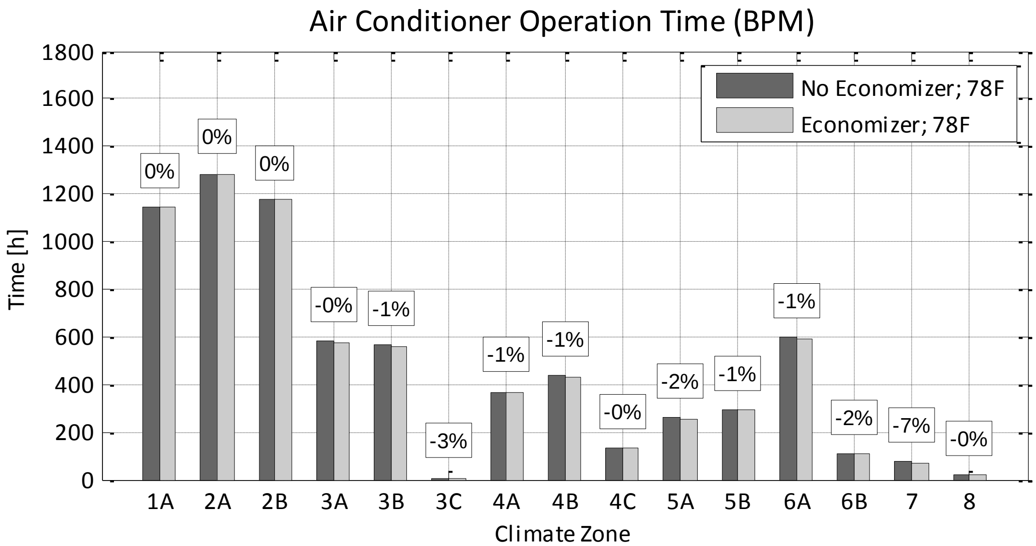

Air Conditioner Operation Time (BPM)
Time [h]
Climate Zone
No Economizer; 74F
Economizer; 74F
1A 2A 2B 3A 3B 3C 4A 4B 4C 5A 5B 6A 6B 7 8
-0% -0% -1% -3% -5% -6% -8% -11% -19% -15% -15% -9% -24% -20% -35%

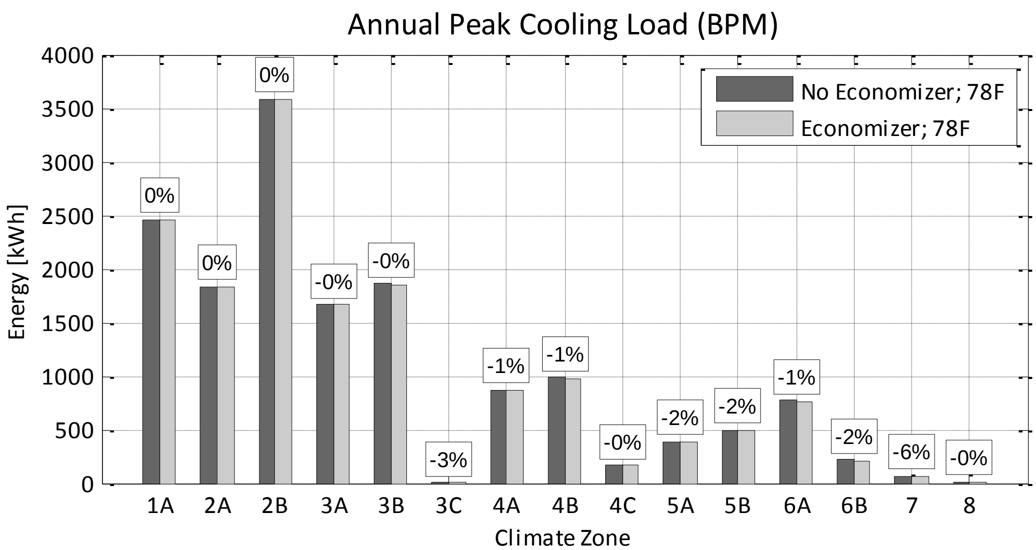

Annual Peak Cooling Load (BPM)
Energy [kWh]
Climate Zone
No Economizer; 74F
Economizer; 74F
1A 2A 2B 3A 3B 3C 4A 4B 4C 5A 5B 6A 6B 7 8
-1% -0% -1% -3% -5% -3% -9% -11% -22% -20% -21% -14% -28% -23% -41%

BPM Economizer - 78°F (25.6°C) Economizer Set Points






Air Conditioner Operation Time (BPM)
Time [h]
Climate Zone
No Economizer; 78F
Economizer; 78F
1A 2A 2B 3A 3B 3C 4A 4B 4C 5A 5B 6A 6B 7 8
0% 0% 0% -0% -1% -3% -1% -1% -0% -2% -1% -1% -2% -7% -0%


Annual Peak Cooling Load (BPM)
Energy [kWh]
Climate Zone
No Economizer; 78F
Economizer; 78F
1A 2A 2B 3A 3B 3C 4A 4B 4C 5A 5B 6A 6B 7 8
0% 0% 0% -0% -0% -3% -1% -1% -0% -2% -2% -1% -2% -6% -0%

Appendix B: Air Conditioning and Economizer Operation times for the PSC Economizer

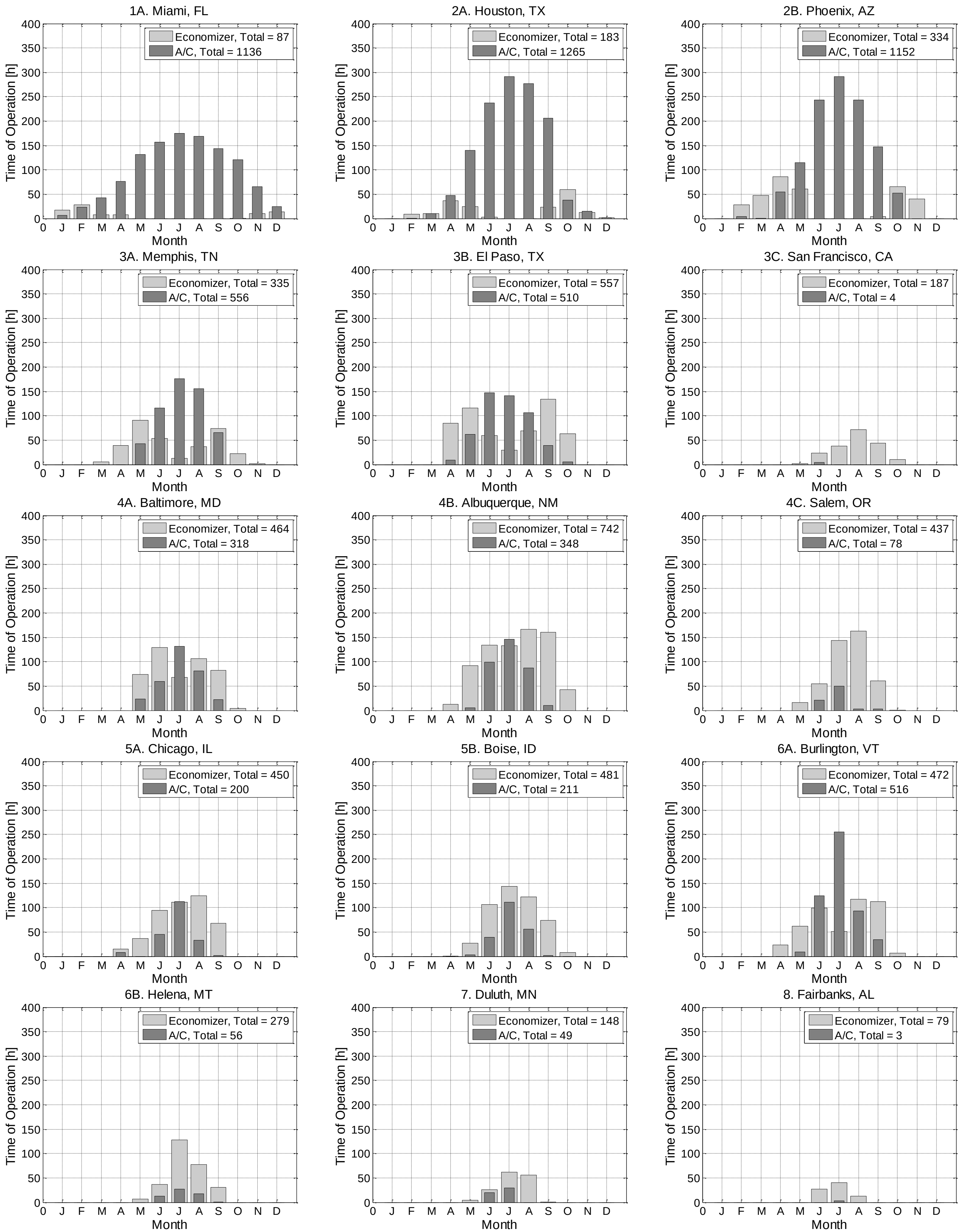

1A. Miami, FL
Economizer, Total = 87
A/C, Total = 1136
2A. Houston, TX
Economizer, Total = 183
A/C, Total = 1265
2B. Phoenix, AZ
Economizer, Total = 334
A/C, Total = 1152
3A. Memphis, TN
Economizer, Total = 335
A/C, Total = 556
3B. El Paso, TX
Economizer, Total = 557
A/C, Total = 510
3C. San Francisco, CA
Economizer, Total = 187
A/C, Total = 4
4A. Baltimore, MD
Economizer, Total = 464
A/C, Total = 318
4B. Albuquerque, NM
Economizer, Total = 742
A/C, Total = 348
4C. Salem, OR
Economizer, Total = 437
A/C, Total = 78
5A. Chicago, IL
Economizer, Total = 450
A/C, Total = 200
5B. Boise, ID
Economizer, Total = 481
A/C, Total = 211
6A. Burlington, VT
Economizer, Total = 472
A/C, Total = 516
6B. Helena, MT
Economizer, Total = 279
A/C, Total = 56
7. Duluth, MN
Economizer, Total = 148
A/C, Total = 49
8. Fairbanks, AL
Economizer, Total = 79
A/C, Total = 3
Time of Operation [h]
Month
0 J F M A M J J A S O N D

Figure 19: Economizer (PSC) and air conditioning operation hours over the year. Annual totals are contained in the legend