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Authors
Turner, William J.N
Walker, Iain S.
Roux, Jordan

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William J.N. Turner
Iain S. Walker
Jordan Roux

Environmental Energy Technologies Division

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ABSTRACT

This study used an advanced airflow, energy and humidity modeling tool to evaluate residential air-side economizers and mechanical pre-cooling strategies using the air conditioner, in all US DOE Climate Zones for a typical new home with ASHRAE Standard 62.2 compliant ventilation. A residential air-side economizer is a large supply fan used for night ventilation. Mechanical pre-cooling used the building air conditioner operating at lower than usual set before the peak demand period. 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 HVAC systems. The short time steps also allow for more precise evaluation of HVAC system cycling operation. The results showed that a residential economizer can save large quantities (more than 2000 kWh/year) of cooling energy – in some cases the energy savings offset all of the mechanical ventilation related energy use. Using a high performance Brushless Permanent Magnet (BPM) air handler in the HVAC system saved an additional 12% of cooling energy compared to more typical Permanent Split Capacitor (PSC) air handlers. However, economizers may cause problems with excess humidity in climate zones 2A and 3A (Houston and Memphis) due to increases in indoor humidity. The economizer was most effective in the hot and dry Climate Zones 2B, 3B (Phoenix, El Paso) and the marine Climate Zone 3C (San Francisco). There were less significant, but still desirable, energy savings in mixed temperature Climate Zones 4A, B & C (Baltimore, Albuquerque and Salem).

The effectiveness of pre-cooling is highly dependent on climate zone and the selected pre-cooling strategy. In this study the pre-cooling strategies included two levels of temperature depression: 22.2° and 23.3°C and pre-cooling times of 8, 5 and 3 hours. The results showed the expected tradeoff between peak savings and increased energy use. In looking to maximize peak energy savings and minimize the extra cooling energy used, our results showed that the high cooling climates (zones 1A-3B) gave the best results for the shortest time and least temperature depression. Climates with less cooling showed better results for pre-cooling with the least temperature depression. Climate Zones 3C and 8 showed no advantage with pre-cooling strategies due to near-zero air conditioning loads. One caveat with the pre-cooling recommendations is that they were for a lightweight wood frame construction and may change for heavier brick/block structures not included in this study.

KEYWORDS
Pre-Cooling, Night Ventilation, Air-Side Economizer, Mechanical Cooling, Ventilation Cooling

Introduction

There is currently a drive toward reducing the maximum instantaneous load on power grids. 'Peak energy demand' refers to the time of day when the loads on the gas and electricity distribution infrastructures reach a maximum. During the winter months this is typically between 4am and 8am while external temperatures are at their coldest and the heating demand is greatest. During the summer months the demand tends to reach a maximum between 4pm and 8pm when the high cooling demand coincides with people returning home from work. Consequently the residential air conditioning load is the highest.

During these peak periods the extra demand on the grid is met by increasing capacity via the operation of power plants with a higher marginal cost and CO_2 emissions. This increases the generation cost for each kilowatt-hour for the utility company. The cost is then passed down to the consumer in increased utility rates. Failure to increase the capacity of the grid can lead to wide scale blackouts when the energy demand outstrips the supply.

Utility companies in the US are beginning to offer tariff-based incentives to consumers to help reduce peak energy demand and hence cost, such as 'Time of Use' (TOU) schemes where a schedule is set by the utility company offering cheaper energy prices during off peak times and more expensive energy during peak times. This encourages consumers to shift their main energy use to periods in which energy generation is less expensive and the overall demand may be met more easily.

Pre-cooling is a strategy that attempts to remove some of the increased peak demand on the electricity grid by shifting the cooling load to non-peak times. The cooling thermostat set points are reduced in the period preceding the peak period in order to force the air-conditioner to switch on. This cools the thermal mass of the house while electricity prices and generation costs are lower. The set points are then raised during the peak period. As the building takes time to warm up, the operation of the cooling equipment is delayed during the hot peak period. Additionally, the efficiency of air-conditioners (Energy Efficiency Ratio or EER) increases with lower outdoor temperatures, so their energy consumption is less while operating during off-peak periods. Beutler (2003) demonstrated via simulation that this approach could reduce peak period residential air-conditioner operation by 75 to 84%.

Pre-cooling may also be achieved by ventilation cooling using an *economizer*, whereby nighttime ventilation air is used to cool the thermal mass of the building. This is common in commercial buildings where energy standards such as ASHRAE Standard 90.1 (ASHRAE, 2010b) require the installation of economizers to provide ventilation cooling. Using computer simulations of a range of pre-cooling strategies, Braun (1990) showed that peak loads could be reduced by 10 to 35% in commercial buildings. Becker and Paciuk (2002) showed that nighttime ventilation and regular cooling could reduce peak demand by 43 to 56%. Springer (2007) combined pre-cooling and ventilation cooling strategies while monitoring test residential buildings in Sacramento, and found that peak-period compressor energy consumption could be reduced by up to 88%.

However, delivering large quantities of outside air to the indoor environment has implications regarding levels of humidity. Traditionally economizer use has not been recommended in humid climates as supplying moist air can increase the indoor humidity resulting in comfort and possible health issues for the occupants.

This study looks at the potential for mechanical air-conditioner pre-cooling combined with nighttime ventilation cooling to reduce the peak electricity load and energy consumption. We also looked at the humidity implications of using an economizer. A computer modeling approach was used to study the load reduction of several cooling strategies in 15 different US climates. The results of the simulations were used to assess the balance between peak energy reductions and off-peak energy consumption, while still providing good thermal comfort and indoor air quality (IAQ).

Economizers

Economizers (also known as 'air-side economizers') are large supply fans that 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 commonplace in large, commercial buildings where cooling loads can be high. For commercial buildings, economizers are required in order to meet ASHRAE Standard 90.1 (2010b) - the North American energy efficiency standard for buildings (except low-rise residential). Running 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).

Residential applications of economizers are less widespread. Unlike their commercial counterparts, residential building loads are dominated by heat exchange through the building envelope (air infiltration, solar gains etc.) and need fewer hours of cooling at low ambient temperatures (Ueno and Straube, 2011). However, there is still potential for energy savings via the reduced use of mechanical cooling from nighttime ventilation.

In typical residential applications the forced air air handler is used as the economizer fan. An automatic damper opens allowing the economizer to distribute outside air to the occupied zone via the supply ducts. To avoid over-pressurizing the house during operation of the economizer, some pressure relief mechanism is needed. This could take the form of a motorized skylight in the ceiling or pressure relief dampers in the return ducts, for example. The main purpose of economizers is to provide cooling. However, because they supply outside air they also ventilate incidentally. In the future, with the use of the equivalent ventilation principle (already accepted as part of ASHRAE Standard 62.2), there will be ways to take credit for this in ventilation standards.

There are drawbacks to economizer use. During humid weather, the increased ventilation rate resulting from economizer use can increase the indoor humidity, potentially leading to comfort and moisture problems. Higher humidity levels can also increase the latent heat load on air conditioning units. Traditionally it has been thought that the use of an economizer is not recommendable in humid climates such as Miami, Florida for this very reason. If improperly installed, the economizer components can lead to increased envelope leakage, especially through the dampers (McWilliams and Walker, 2005). This can cause higher levels of infiltration leading to larger space-conditioning loads. Similarly, sealed and well-insulated ducts are necessary for effective use of an economizer to avoid warmer or contaminated attic, crawlspace or garage air being passed into the occupied zone.

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). Other literature reviews on indoor humidity and health effects include Baughman and Arens (1996) and Arundel et al. (1986). When relative humidity exceeds 50% for prolonged periods dust mite populations increase more rapidly (Arlian, 1992), with a consequential increased risk of asthma (Institute of Medicine, 2000) and exposure to other dust mite allergens. Fungal growth is expedited by dampness in houses (Gallup et al., 1987, Waegemaekers et al., 1989, Douwes et al., 1999) which leads to the production of harmful fungal spores and allergens. Fungi can also lead to the structural damage of buildings. In Europe, Japan and Australia the most common wood-rotting fungus, among others, is the dry-rot fungus *Serpula lacrymans* (Singh, 1999), which can spread quickly throughout timber in a building and compromise the structural integrity.

When bringing in the large quantities of air associated with economizers any humidity in the outdoor air is also brought into the home. This is a particular issue in hot humid climates. In this study we will examine this issue in more depth using sophisticated humidity model. The model account for moisture removal by air conditioners, sources of moisture in the home due to occupants, moisture coupling between air in the home and other home components that act as moisture storage. 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, internal gains from cooking, showering etc. Therefore, indoor humidity will be higher than outdoor humidity due to presence of occupants and ventilation will decrease indoor humidity. However, the use of air conditioning (or dehumidifiers) removes moisture from inside air. When this rate of removal is faster than the rate of addition from occupants, the indoor humidity will be lower than outdoors. In this case, ventilation, particularly at the high rates of economizers, can increase indoor humidity. This implies that economizer use may only be appropriate in humid climates during shoulder seasons when the outdoor humidity and air temperature are lower and air conditioning is rarely used.

Low sensible load, energy efficient homes are particularly sensitive to humidity issues because their small air conditioning systems lack the capacity or operating time to control humidity. This is further exacerbated by climates where the nighttime temperatures do not fall low enough for economizer operation – typically these are hot, humid climates. The economizer then only operates during shoulder seasons when there is little air conditioning operation. The simulations in this study will examine these humidity issues in detail and the results will be used to make recommendations for the possible restriction of economizer operation.

PSC and BPM Air Handler Motors

Residential economizers generally use the heating/cooling system air handler to deliver the ventilation cooling air to the house. Studies by CMHC (1993) and Walker (2008) have shown that residential air handlers are almost an order of magnitude less efficient than their larger commercial counterparts. Typical residential air handler performance is approximately 1L/s/W and can be reduced further by poor cabinet and duct design. Brushless Permanent Magnet (BPM) air handlers have the capability at least to quadruple this performance to 4L/s/W at lower airflows or for low airflow resistance duct systems.

The standard fan in a residential forced-air heating and cooling system has a permanent split capacitor (PSC) type motor. PSC motors can be optimized for efficiency and power factor at a rated load. They are widely considered to be the most reliable of single-phase motors. 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). Due to the way the speed is controlled in a PSC motor, a fan operating at a fractional speed consumes approximately the same power as one operating at full speed, with an accompanying decrease in efficiency (Walker et al., 2003).

An alternative to PSC motors are Brushless Permanent Magnet (BPM) motors. The speed of BPM motors are electronically controlled and can be set specifically to match the airflow requirements for each application. They are designed to speed up or slow down in an attempt to preserve airflow regardless of the static pressure across the fan, e.g. when filters become dirty and increase the restriction of airflow. This self-moderation helps maintain an airflow range through the heat exchanger, close to the optimal flow rate for which the fan and heat exchanger were designed. The drawback is cost - BPM air handlers are more expensive than PSC air handlers and so are rarely found in residential HVAC systems, though Raymer (2010) notes that this trend is changing.

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 HVAC applications involving night ventilation with an economizer. In this study it was assumed that the economizer operated at cooling airflow rate, however, it might be possible to operate the economizer at lower airflow rates in a low sensible cooling load home where the reduced cooling effect of the lower flow still provides a significant fraction of the cooling requirements.

Simulations

In this study, we investigated two mechanical pre-cooling strategies:

1. Night ventilation using an economizer

2. Mechanical pre-cooling using the house air-conditioner, in conjunction with night ventilation using an economizer

For each strategy there was a reference case, used to determine the effect of pre-cooling. For the night ventilation with economizer simulations, the reference case was a house with no whole-house ventilation. The reference for the mechanical pre-cooling simulations using the air conditioner was a house with continuous whole-house mechanical exhaust ventilation and an economizer, but no mechanical pre-cooling using the air conditioner. For both reference cases, the air conditioner ran to the standard operating set points (see Table 3) and infiltration effects were included in the ventilation rate of the house. Additional simulation details are to be found in Appendix A.

Climate Zones

We performed simulations for all DOE climate zones (1 – 8) using TMY3 weather data (Wilcox and Marion, 2008) for their representative cities (see Figure 1 and Table 1) (Briggs et al., 2003).

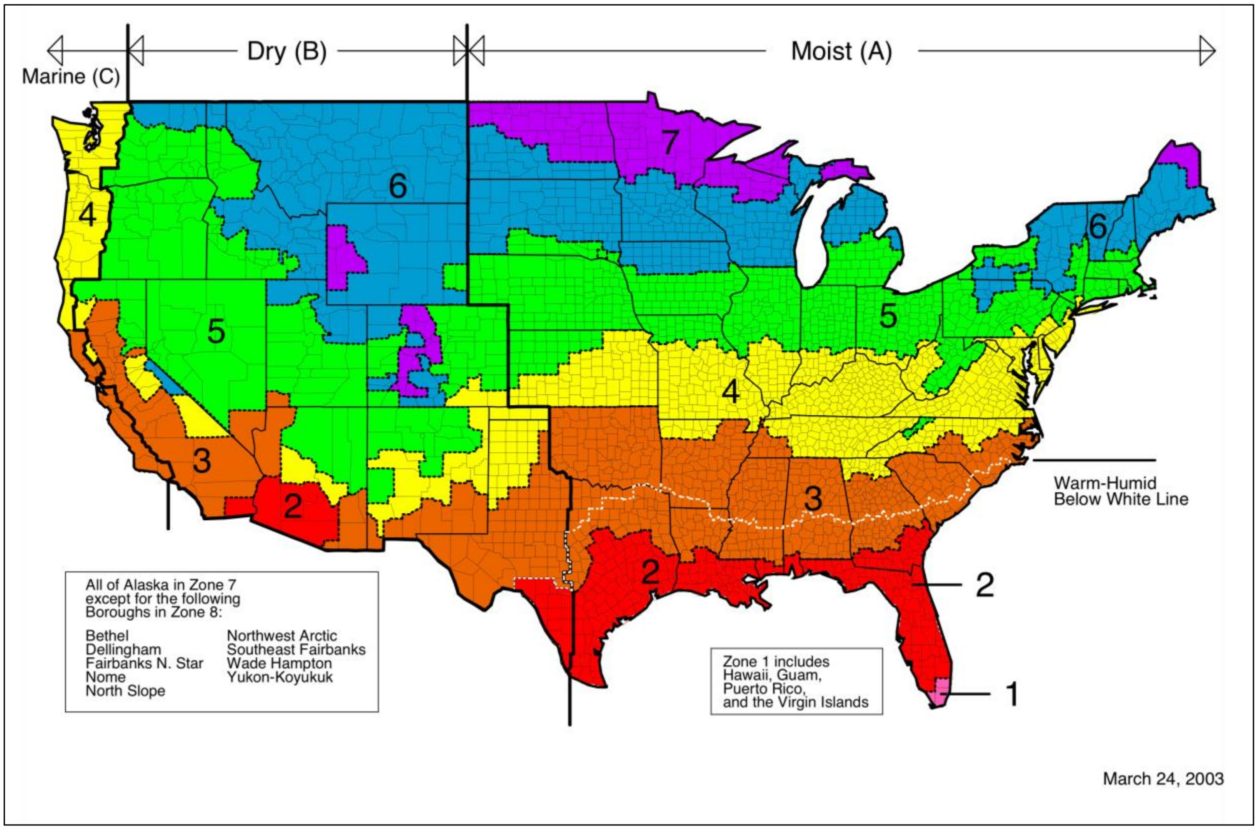


Figure 1: IECC Climate Zones for the United States (Briggs et al., 2003)

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

House geometry was based on the California State Energy Code Title 24 Prototype C (Nittler and Wilcox, 2008), which is a reasonably performing new home (Figure 2). It is better than most existing homes, but not a high performance home like those found in the Building America program. It has an occupied living area of 195 m^2 (2,100 ft^2) with uniform 2.5 m (8.2 ft^2) ceilings, and a volume of 488 m^3 (17,220 ft^3). The house was simulated to contain four occupants with three bedrooms, three bathrooms and one kitchen. Envelope leakage was 4.8 ACH_{50}, typical of new construction, based on recent studies by Offerman (2009) and Wilcox (2011).

Figure 2: The Title 24 housing Prototype C, with 195 m2 occupied floor area

Indoor Air Quality (IAQ)

Indoor Air Quality was determined by calculating the indoor relative dose and exposure to indoor contaminants (see Sherman (2008) and (Sherman and Wilson, 1986)), based on a constant indoor emission rate. A relative dose of unity is what an occupant would receive if they lived in a house ventilated to meet the ASHRAE Standard 62.2 minimum whole-house mechanical ventilation rate. Above unity indicates a lower ventilation rate, and below unity indicates a higher ventilation rate. Relative exposure is calculated every minute. Relative dose is a 24-hour time running average of the relative exposure. The annual relative dose and exposure were calculated for all the simulations based on the occupancy schedule (see below).

Building Occupancy and Fan Scheduling

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

Operation of additional ventilation systems was based around the above occupancy schedule. On weekdays one bathroom fan (with an airflow of 25 L/s to be compliant with ASHRAE Standard 62.2) 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 times were constrained between 7 a.m. and 11 p.m. An algorithm was used to add some degree of daily variability into the bathroom fan schedules. This algorithm did not violate the criteria of a maximum of 40 minutes bathroom fan operation per occupant per day and the general occupancy time periods. The algorithm was used to generate a full yearlong schedule. For each home the same yearlong pre-calculated schedule was used in each simulation. Thus there was variability from day to day as the simulations progressed through the year, but the same variability was used for each simulation. In other words, for any given day of the year the schedule was the same.

The kitchen range hood operated for one hour per day between 5.30 p.m. and 6.30 p.m. On weekends there was an additional 30 minutes of operation in the morning between 9.30 a.m. and 10.00 a.m. The range hood had an airflow rate of 50 L/s to meet the minimum ASHRAE Standard 62.2 requirement.

Clothes dryers operated irrespective of occupancy, as if they were set to run on a timer. Two laundry days each week were simulated. Dryer operation was for three consecutive hours between 11 a.m. and 2 p.m. The clothes dryer exhaust had an airflow rate of 75 L/s.

Internal Loads

The internal loads that make indoor humidity higher than outdoors depend on occupant activities. The daily latent heat gain from moisture generation will follow the approach used previously by Walker and Sherman (2006) and Walker and Sherman (2007). The moisture generation rates are based on ASHRAE Standard 160 (ASHRAE, 2009) with corrections for kitchen and bathroom exhaust using the bathing, cooking and dishwashing estimates from Emmerich et al. (2005) (see

Table 2). We assumed that all the kitchen and bathroom generated moisture was vented directly to outside using exhaust fans.

For the daily sensible heat gain from lights, appliances, people and other sources, the Title 24 ACM (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 was used. For the simulated house this meant a sensible load of 630 W and a moisture net generation rate of 9.8 kg/day (21.5lb/day). Loads were delivered to the occupied zone at a constant rate throughout the day and were not altered for seasonal adjustments.

Table 2: Internal occupancy based moisture generation rates from ASHRAE Draft Standard 160P and Emmerich et al.

Number of Occupants	Moisture Generation Rate [kg/day]	Bathing, Cooking and Dishwashing [kg/day]	Net Generation Rate [kg/day]
2	7.8	3.2	4.6
3	12.1	3.6	8.5
4	13.8	4.0	9.8
5	14.7	4.4	10.3

Ventilation Cooling using an Economizer

The economizers in this study operated in cooling mode when the outdoor temperature was 3.3°C (6°F) or more below the indoor set point (see Table 3) 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.

For each climate zone the economizers were sized to match the largest airflow rate and power consumption of the air handler unit. The airflow rate for both PSC and BPM air handlers was 400 cfm/ton (57 L/S/kW) of cooling capacity. The power draw was 0.5 W/cfm for the PSC air handler and 0.167 W/cfm for the BPM air handler.

Mechanical Pre-Cooling

Pre-cooling was simulated by reducing the thermostat set points during the pre-peak time periods. A thermostat with set-point temperatures depending on time-of-day was used. An initial set of reference simulations were run with no pre-cooling.

- There were two pre-cooling thermostat temperatures of 22.2°C and 23.3°C (72°F and 74°F)
- Pre-cooling windows were:
    - 08:00 to 16:00 (long)
    - 10:00 to 16:00 (medium)
    - 12:00 to 16:00 (short)
- The peak period was defined as 16:00 to 20:00 (four hours in length)

Baseline Thermostat Set Points

Table 3 shows the thermostat set points used in the simulations for the heating and cooling equipment. These were then changed according to the pre-cooling regimes above.

Table 3: Thermostat Set Points

Time		Heating		Cooling	
Start	End	°C	°F	°C	°F
0:00	1:00	20.0	68	25.0	77
1:00	2:00	20.0	68	25.0	77
2:00	3:00	20.0	68	25.0	77
3:00	4:00	20.0	68	25.0	77
4:00	5:00	20.0	68	25.0	77
5:00	6:00	20.0	68	25.0	77
6:00	7:00	20.0	68	25.0	77
7:00	8:00	21.1	70	26.7	80
8:00	9:00	21.1	70	26.7	80
9:00	10:00	21.1	70	26.7	80
10:00	11:00	21.1	70	26.7	80
11:00	12:00	21.1	70	26.7	80
12:00	13:00	21.1	70	26.7	80
13:00	14:00	21.1	70	26.7	80
14:00	15:00	21.1	70	26.7	80
15:00	16:00	21.1	70	26.7	80
16:00	17:00	21.1	70	25.0	77
17:00	18:00	21.1	70	25.0	77
18:00	19:00	21.1	70	25.0	77
19:00	20:00	21.1	70	25.0	77
20:00	21:00	21.1	70	25.0	77
21:00	22:00	21.1	70	25.0	77
22:00	23:00	21.1	70	25.0	77
23:00	0:00	20.0	68	25.0	77

Note 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 air conditioners are used to maintain indoor temperature and reduce the humidity. Consequently these climates exhibit higher energy use than if the set points in Table 3 were used.

Results and Discussion

Below will be discussed the implications on IAQ, the annual energy use of the home, and the indoor relative humidity (RH), from adding an economizer to provide night ventilation. Then the potential for peak demand reductions using mechanical pre-cooling combined with night ventilation will be discussed.

The home under study was of lightweight wood-frame construction. While this is typical of the majority of new home construction, homes built from brick or block with higher thermal mass may have different optimum results due to the longer time constants associated with heating and cooling the structures. The issue of higher mass homes may be addressed in future work.

Night Ventilation with an Economizer

Adding an economizer to provide night ventilation cooling has implications for IAQ, and for the annual energy consumption of the house, and the impact of humidity on the indoor environment.

Indoor Air Quality (IAQ)

Figure 3 to Figure 6 show the relative dose and exposure for the house with and without an economizer. The relative doses and exposures are hourly averages, minimums, and maximums during occupied time periods, for all US climate zones. The default infiltration credit of 10 L/s per 100 m2 of floor area, as per the 2010 edition of ASHRAE Standard 62.2, is included in the dose and exposure calculations. To comply with ASHRAE 62.2 an annual relative dose equal to or less than 1 is required.

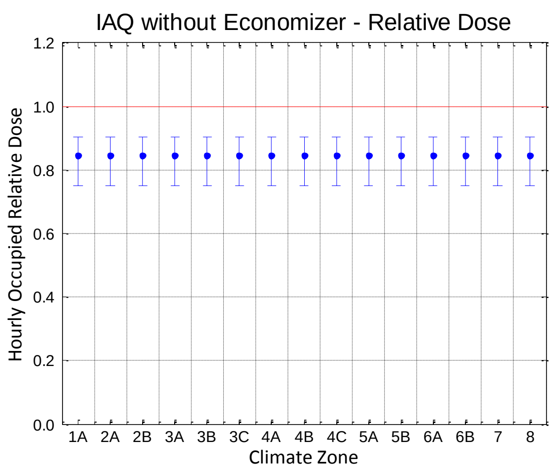


Figure 3: Hourly occupied relative dose without an economizer, by climate zone

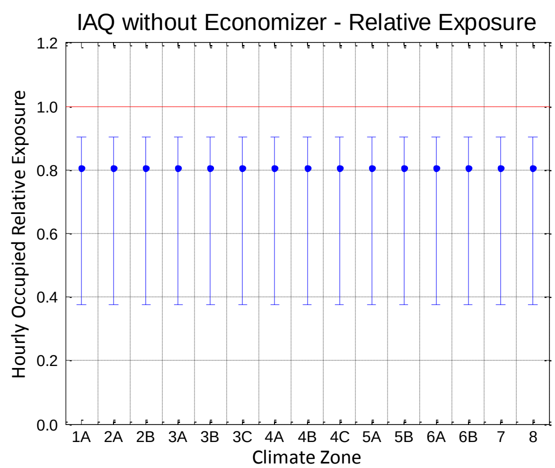


Figure 4: Hourly occupied relative exposure without an economizer, by climate zone

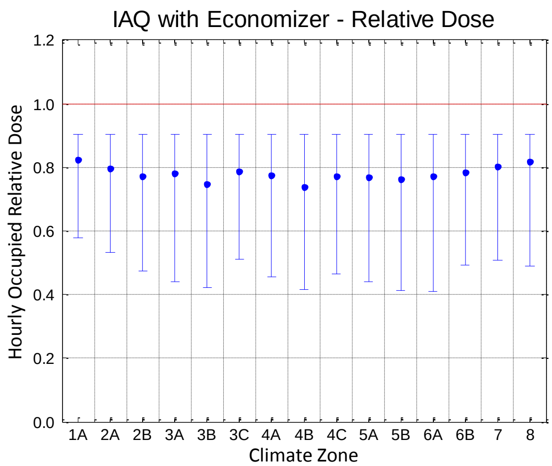


Figure 5: Hourly occupied relative dose with an economizer, by climate zone

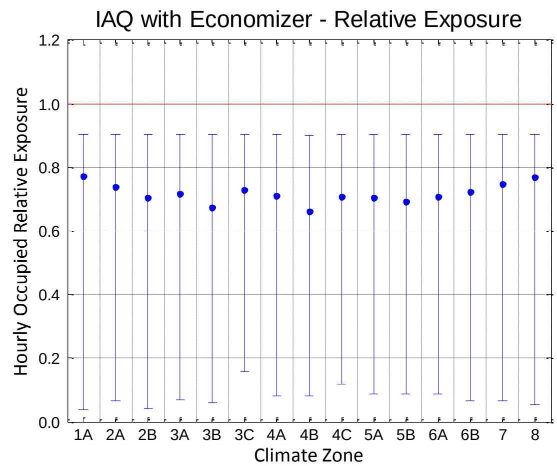


Figure 6: Hourly occupied relative exposure with an economizer, by climate zone

Without an economizer the mean, minimum and maximum relative dose and exposures are determined by the whole-house ventilation fan, plus the other exogenous fans. As these were sized to meet ASHRAE 62.2 based on floor area and number of bedrooms (as a proxy for number of occupants), there is no climate-to-climate variation in the results.

With an economizer we see some climate-to-climate variation in the relative dose and exposure. This is due to the economizer fan running to an algorithm based on indoor and outdoor temperatures. The economizer system also uses the air handler fan which is sized based on climate. The maximum doses and exposures are determined by the schedule for the exogenous fans running during the heating season because the economizer does not operate. As all the climate zones use the same fan schedule the maximums are the same independent of climate zone. The minimums are determined by the combination of the exogenous fans running while economizer is operating in the cooling season.

In all climates, the economizer acts to reduce the mean annual relative dose and exposure resulting in improved IAQ. This is due to the additional airflow from the economizer fan. The climates with the most economizer operation see the greatest increase in IAQ.

Energy

The results of the simulations were used to calculate the energy use with and without PSC and BPM motor-driven economizer operation. To address the humidity issue, the number of hours that the indoor RH exceeded 70% was calculated.

The ventilation energy is all of the energy associated with adding a whole-house ventilation system to a house. This includes both the fan energy used to drive the ventilation airflow, and the extra space-conditioning energy associated with the increased air exchange rate. The reference simulations had no whole-house ventilation or economizer operation, but did have infiltration from cracks and leaks in the building envelope and task ventilation from bathroom/kitchen fans etc. A whole-house exhaust fan running both with and without an economizer was then added. The difference in total annual energy use between the reference case and another case with whole-house ventilation is the ventilation energy. Figure 7 shows the ventilation energy incurred from adding whole-house ventilation with and without an economizer. The biggest energy savings were in climates where it cools down at night: 2A, 2B, 3B and 3C. In climates 1A, 3A, and 4B it does not cool down enough at night for economizer operation. The best economizer performance was in zone 2A where the energy associated with ventilation is actually negative due to the reduction in air conditioner energy use being larger than energy associated with ventilation. In 2A the total reduction in energy was 1,716 kWh. The colder climates see much lower ventilation energy savings from adding an economizer.

The BPM-driven economizer (see Figure 8) has an additional 10% ventilation energy saving compared to a PSC –driven economizer, due to the lower energy used to operate the relatively large air handler (compared to usual ventilation air handlers). This is enough to offset completely the ventilation energy requirement for the whole year in climate zones 2A, 2B, 3B and 3C. Figure 9 shows a direct comparison between the cooling energy when using a PSC economizer and when using a BPM economizer. The cooling energy is the sum of the energy used by the air conditioner and the air handler while the air conditioner is running. Climate zones 1A, 2A and 2B show the largest absolute cooling energy savings from using a BPM motor rather than a PSC motor. The average cooling energy saving across all climate zones is 12%.

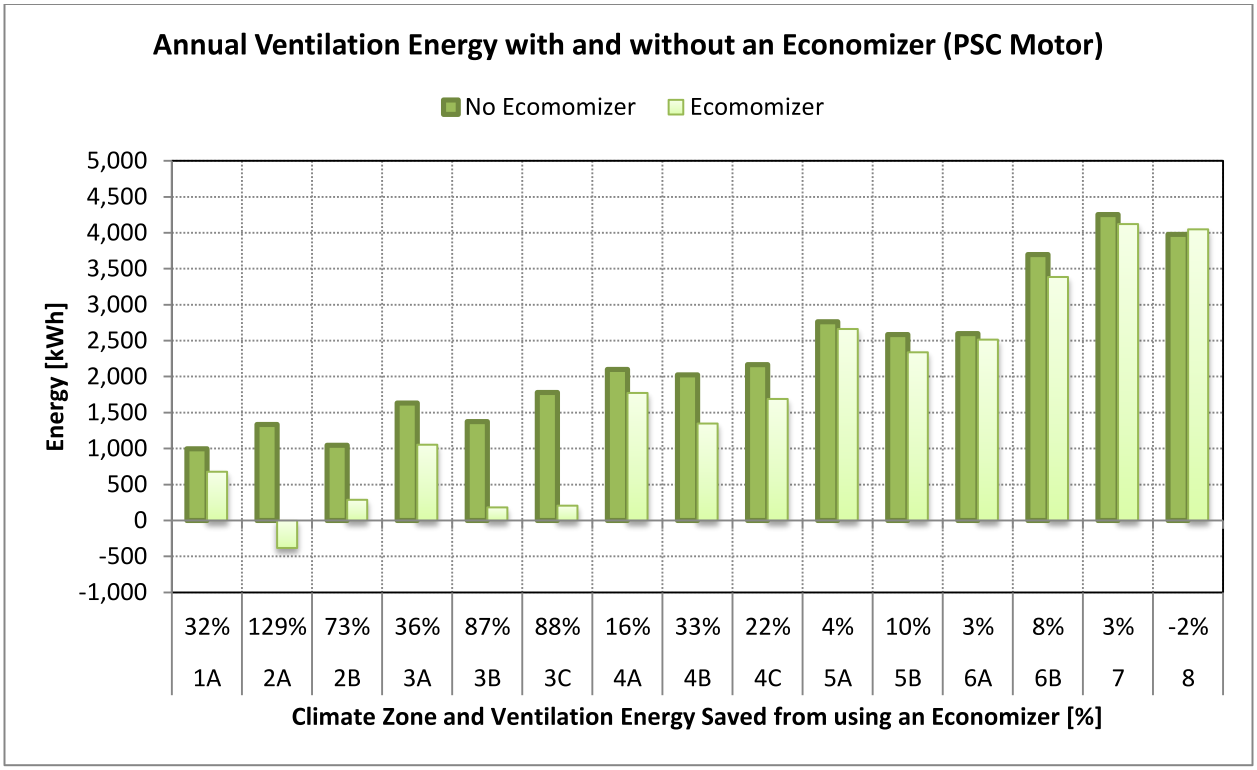


Figure 7: Ventilation energy incurred from adding whole-house ventilation with and without an economizer (PSC motor)

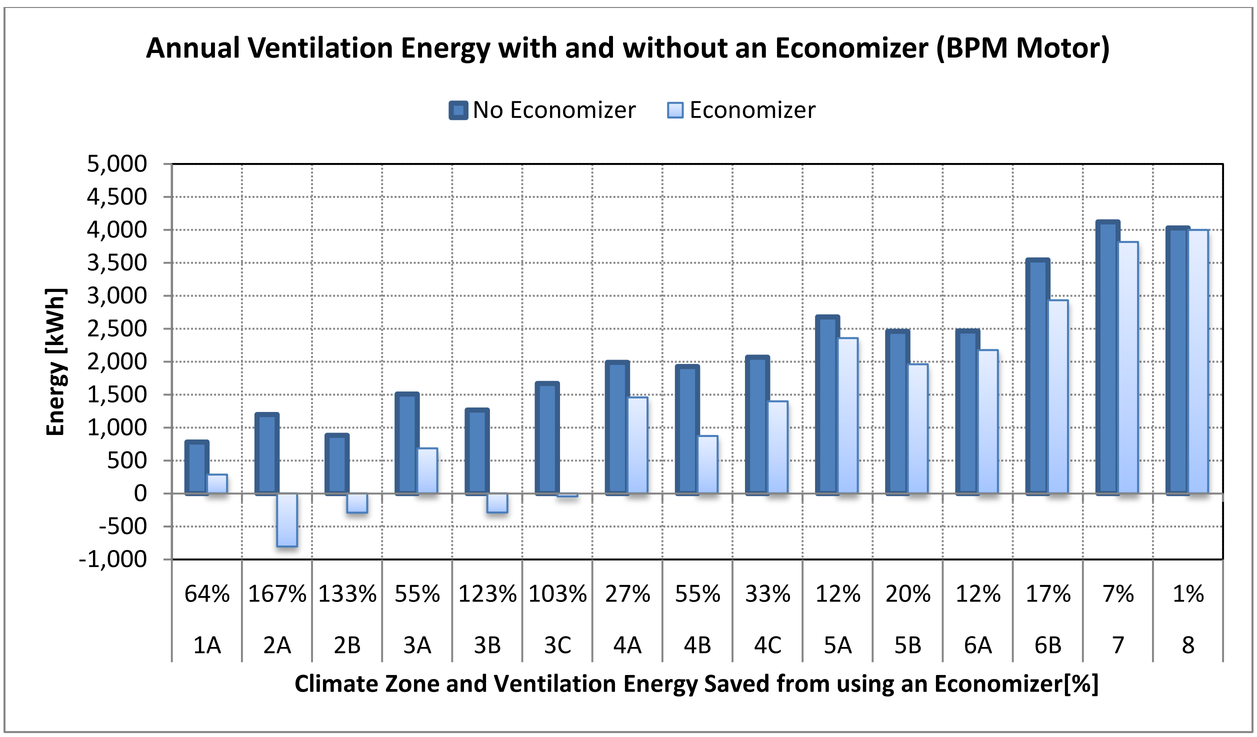


Figure 8: Ventilation energy incurred from adding whole-house ventilation with and without an economizer (BPM motor)

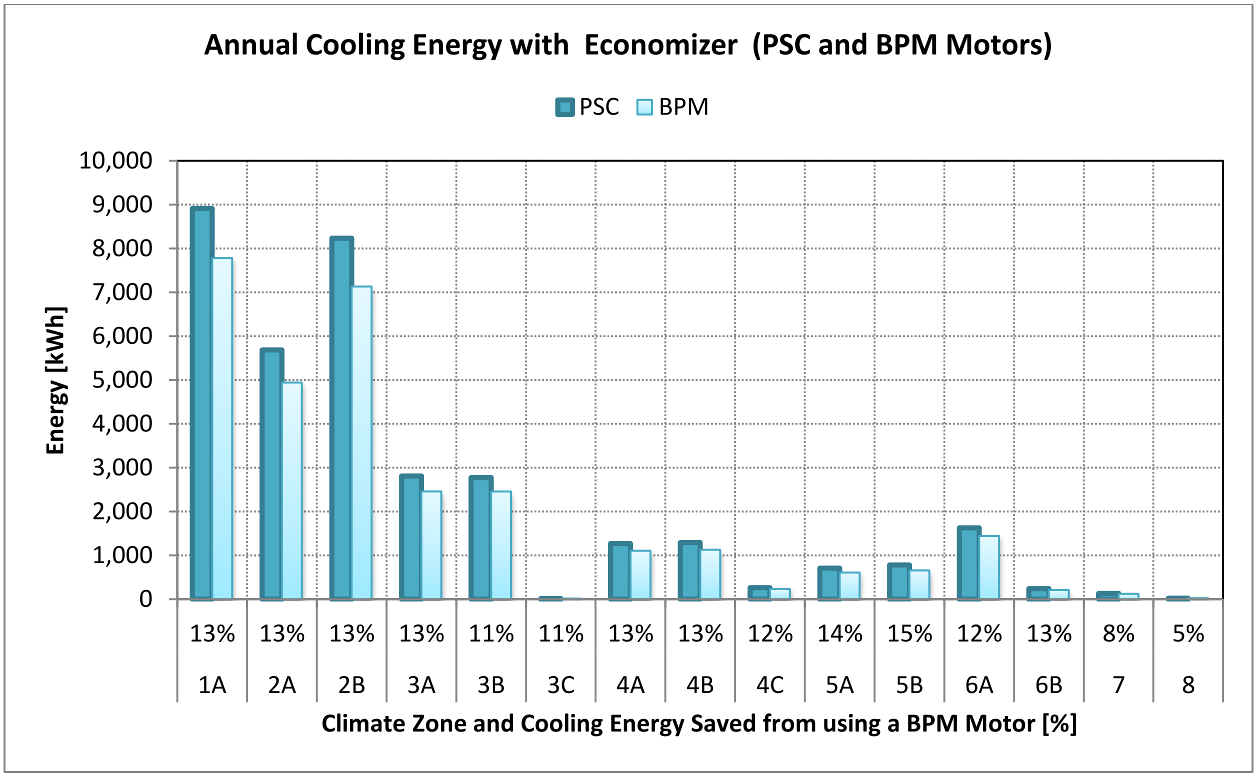


Figure 9: Direct comparison between PSC and BPM-driven economizers

Indoor Humidity

Figure 10 shows the number of hours in a year that the indoor RH exceeded 70%. The high humidity hours roughly scale with the outdoor humidity in each climate, as expected. The effect of the economizer is generally to raise indoor humidity. An interesting exception is in 1A where the economizer had little effect. This is because the economizer only operates on cooler nights that are coincident with lower outdoor humidity, whereas on hotter humid nights it does not operate.

To understand these results better, Figure 11 and Figure 12 show the *daily* relative humidity (RH) and humidity ratio, respectively, while Figure 13 and Figure 14 show the *weekly* RH and humidity ratio, respectively. We looked at different timescales because the effects of humidity depend on duration as well as magnitude. ASHRAE Standard 160P (2009) uses 70% as design criteria for relative humidity based on Viitanen and Salonvaara (2001). High humidity can become a problem when it is above 70% for timescales on the order of days. Condensation will form on windows and the indoor environment can be unpleasant for occupants. When it 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 and Carmody, 1994) and mold growth with potentially serious health implications, and porous parts of the structure can start to rot. Figure 11 shows that variations can be large over shorter timescales of 24-

hours – but it is only important if the peaks persist for longer. Comparing Figure 11 to Figure 13 – Figure 13 shows more clearly when long term high humidity events occur – particularly those above 70% in Houston (CZ 2A) and Memphis (CZ 3A) during shoulder seasons, and in the winter in Miami (CZ 1A) when the air conditioning is not operating to provide any dehumidification. Dry climates in El Paso and Albuquerque (Climate zones 3B and 4B) do not show these shoulder seasons. Instead, their higher indoor RH is driven by increases in indoor humidity (as shown in the humidity ratio values in Figure 14).

Figure 15 shows at what times of the year the economizer operates. 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.

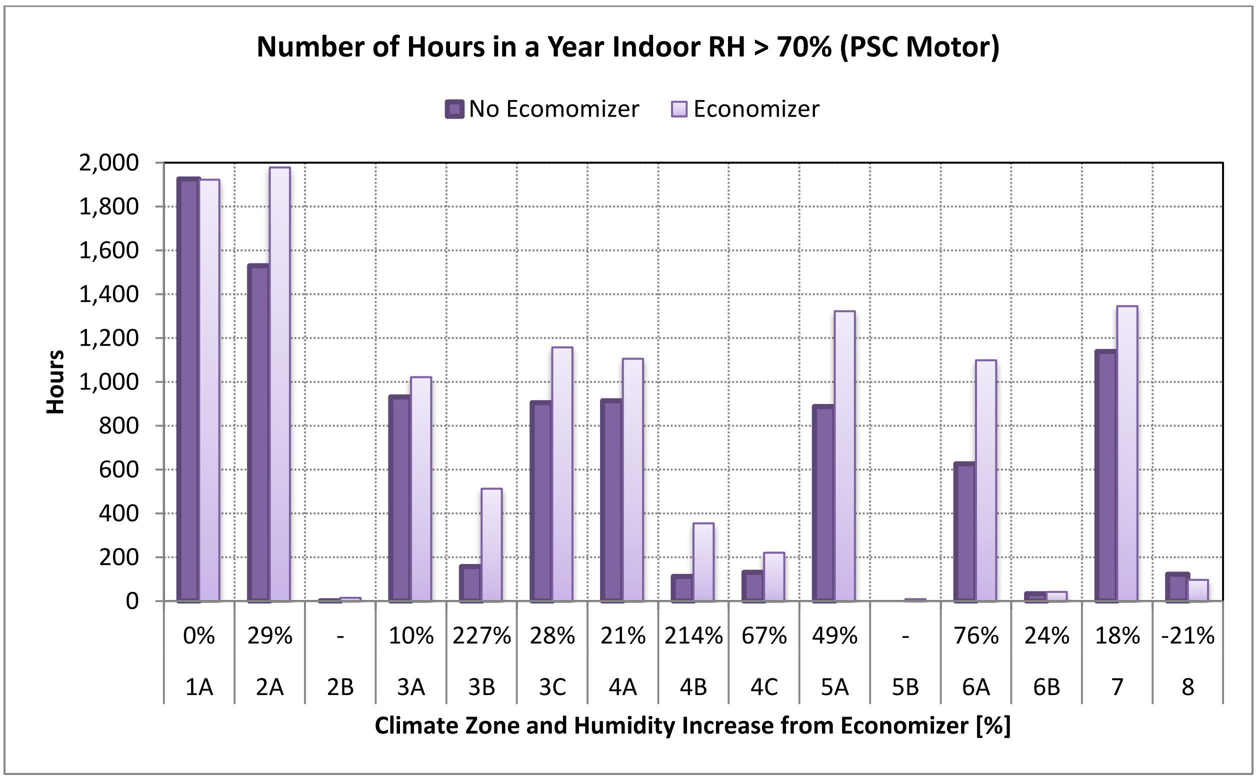


Figure 10: Indoor relative humidity with and without an economizer (PSC motor)

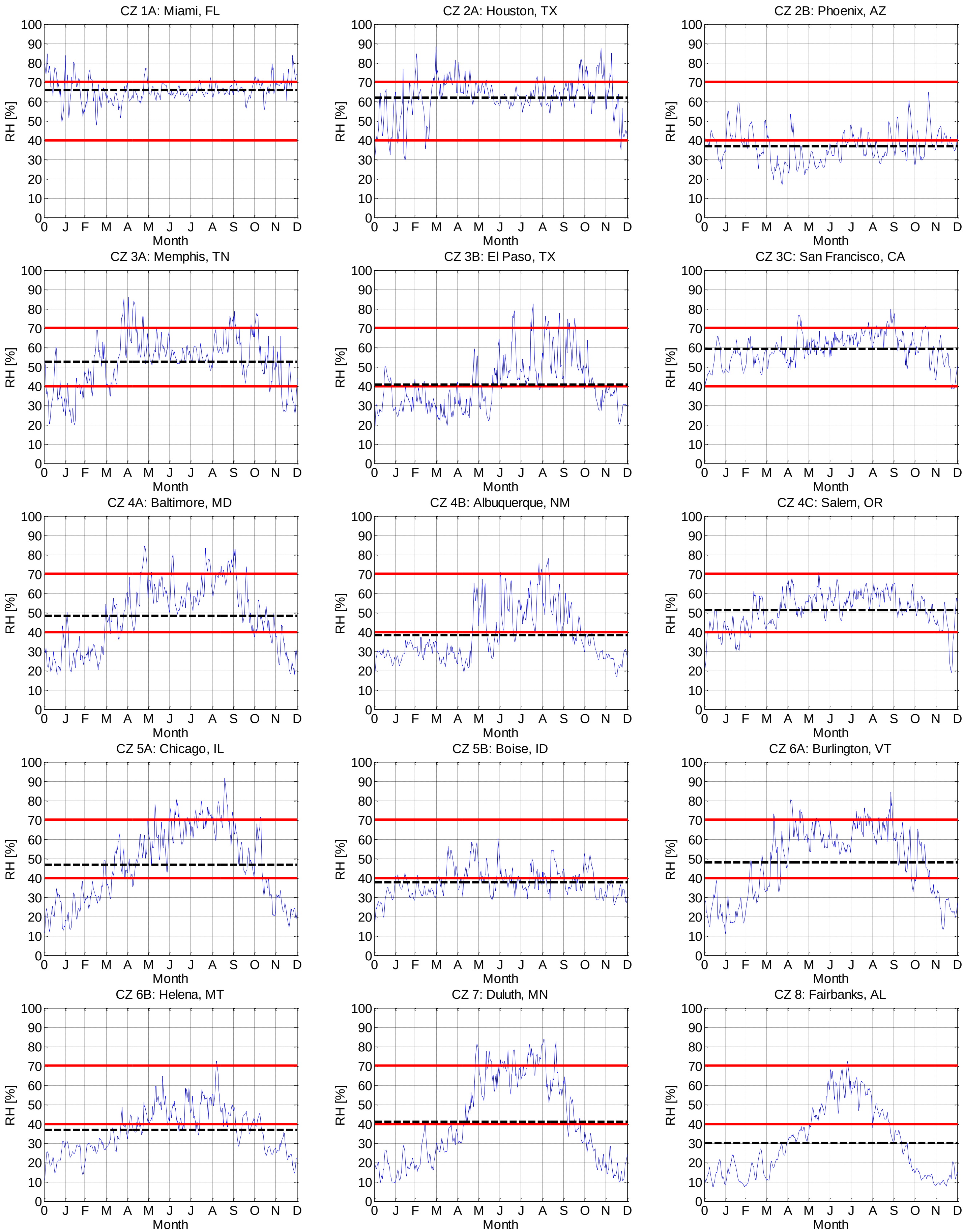
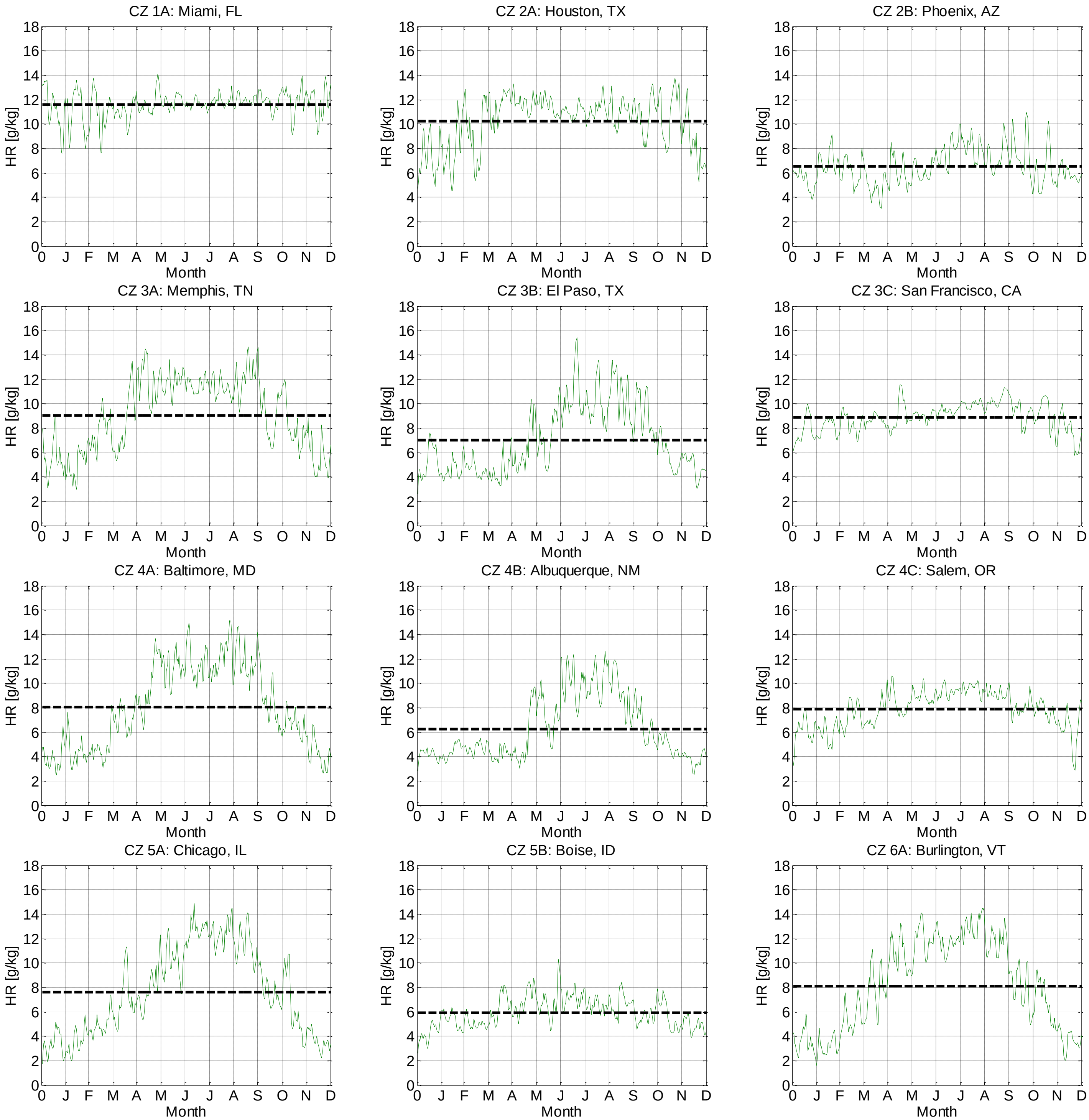

CZ 1A: Miami, FL
CZ 2A: Houston, TX
CZ 2B: Phoenix, AZ
CZ 3A: Memphis, TN
CZ 3B: El Paso, TX
CZ 3C: San Francisco, CA
CZ 4A: Baltimore, MD
CZ 4B: Albuquerque, NM
CZ 4C: Salem, OR
CZ 5A: Chicago, IL
CZ 5B: Boise, ID
CZ 6A: Burlington, VT
CZ 6B: Helena, MT
CZ 7: Duluth, MN
CZ 8: Fairbanks, AL
RH [%]
0 10 20 30 40 50 60 70 80 90 100
Month
0 J F M A M J J A S O N D

Figure 11: Daily average indoor relative humidity (RH). Red lines show 40 and 70%. Thick black dashed lines show the annual mean



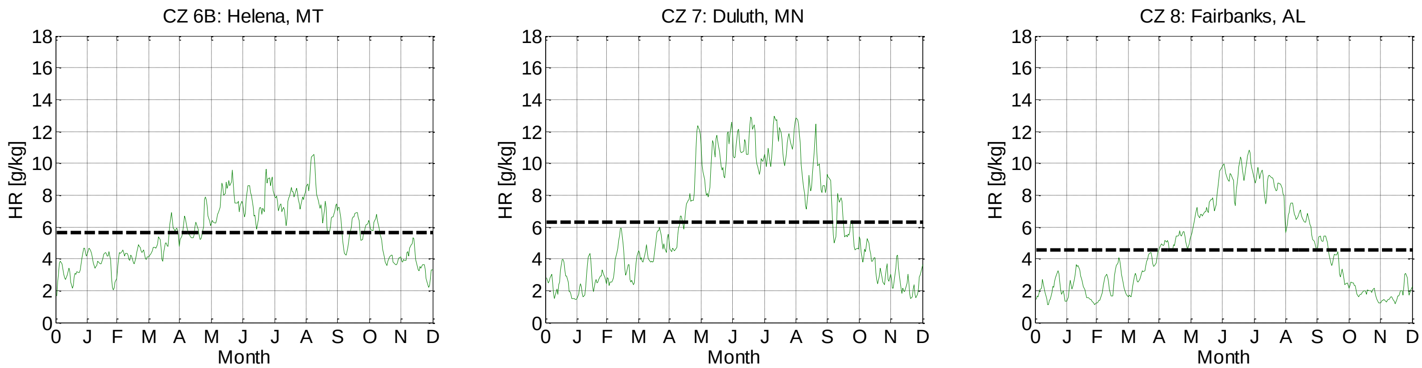
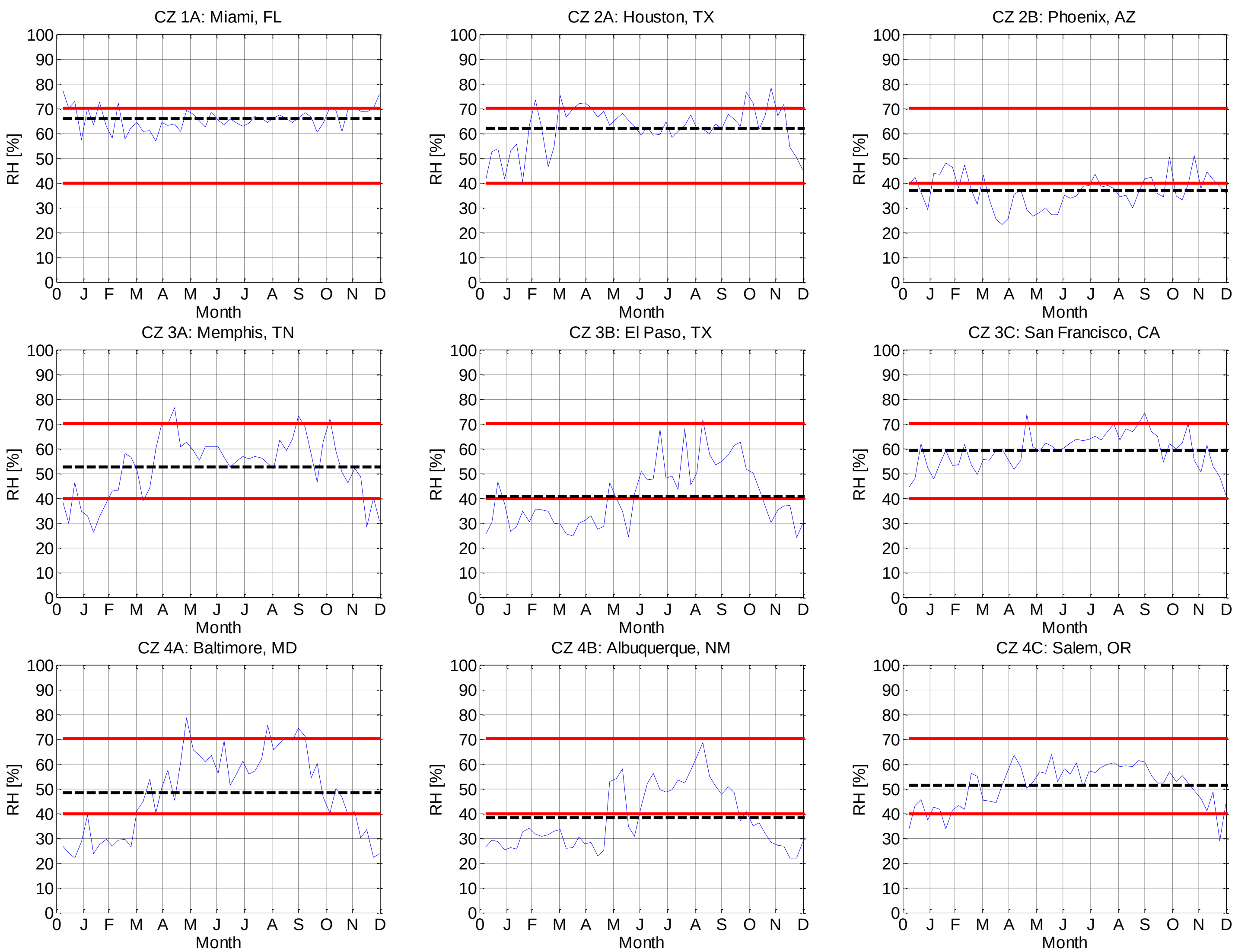


Figure 12: Daily average indoor humidity ratio with the annual mean (thick black dashed line)



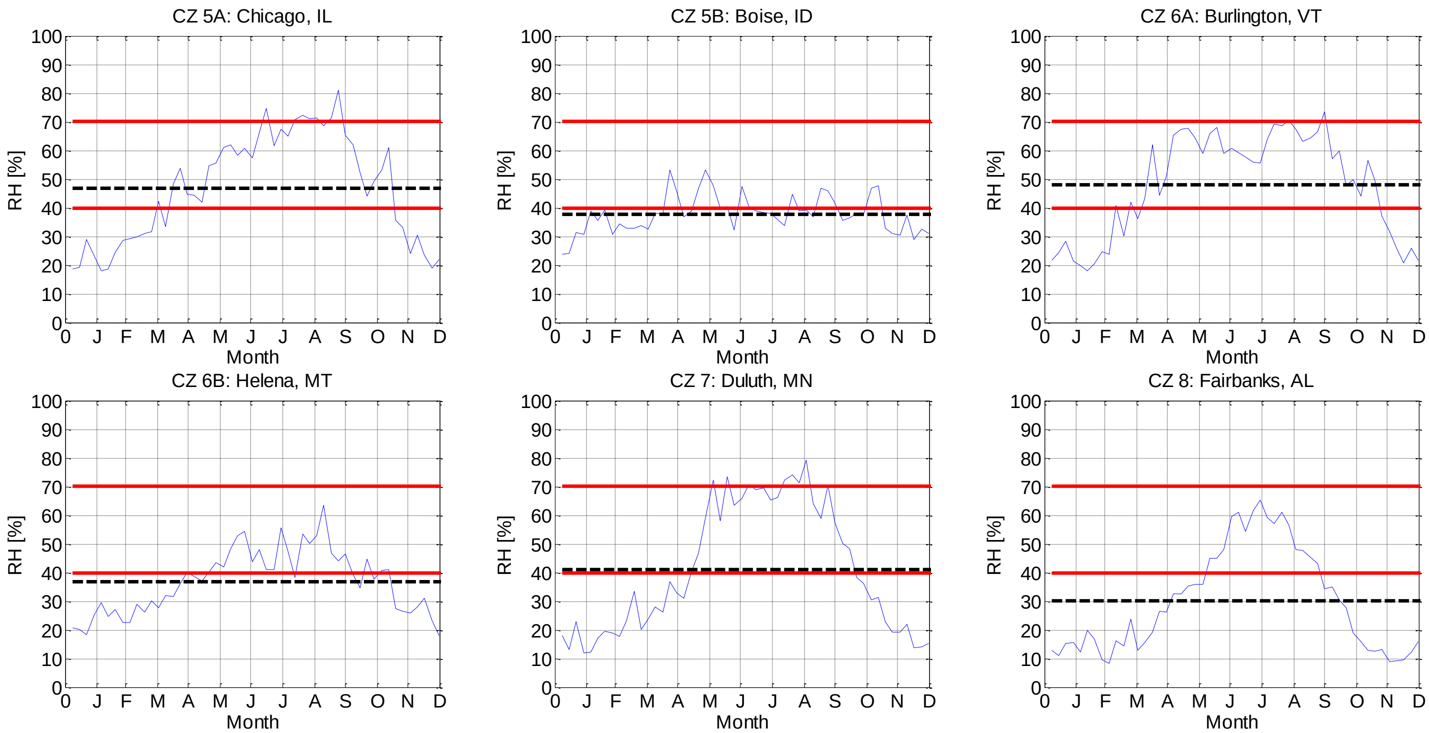
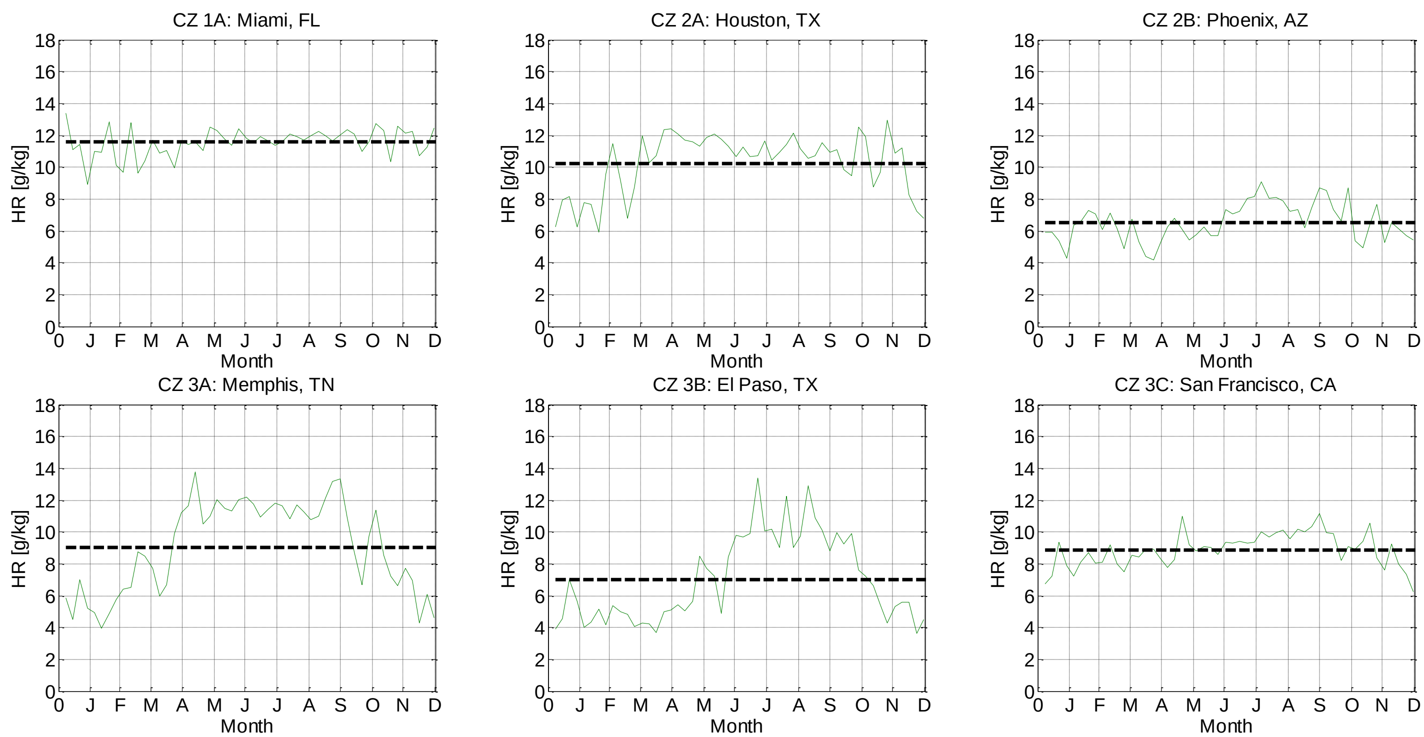


Figure 13: Weekly average indoor relative humidity (RH). Red lines show 40 and 70%. Thick black dashed lines show the annual mean.



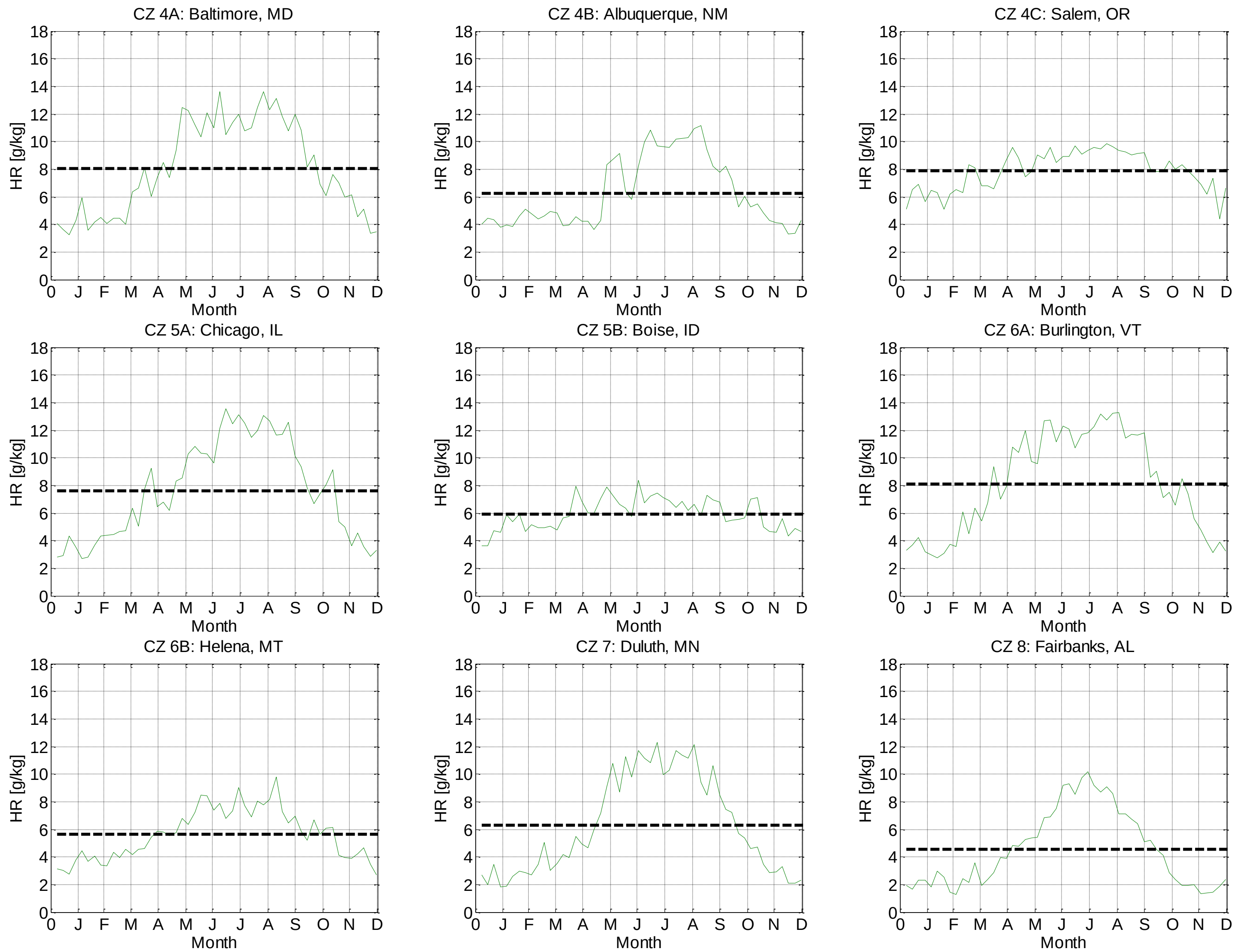
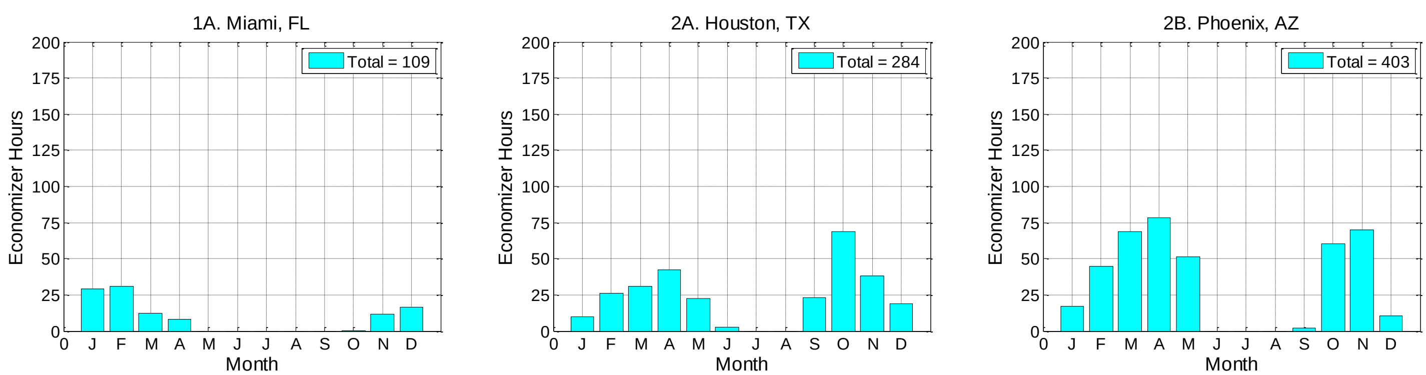


Figure 14: Weekly average indoor humidity ratio with the annual mean (thick black dashed line)



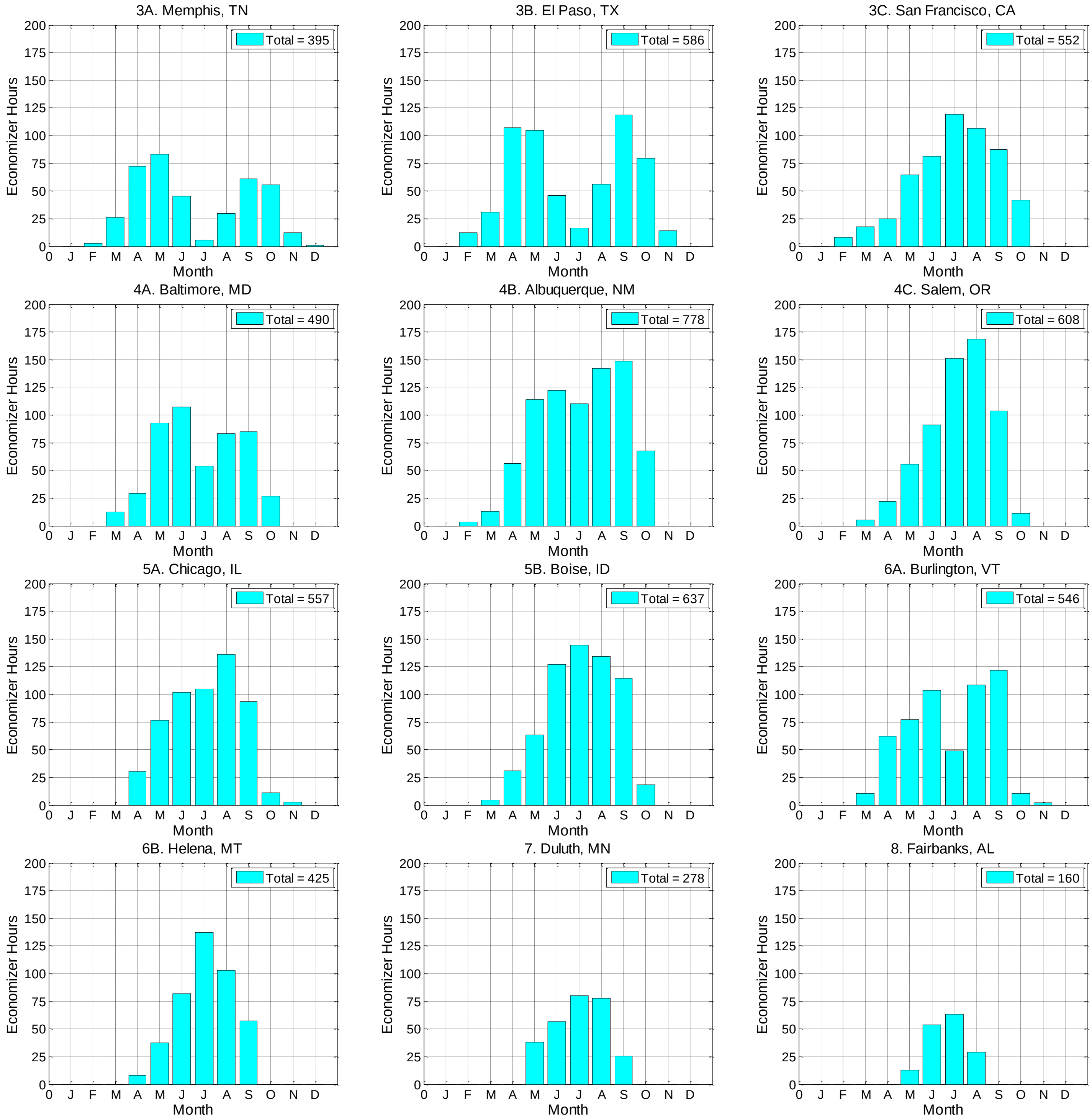


Figure 15: Economizer operation hours over the year. Annual totals are contained in the inset boxes

Mechanical Pre-Cooling

The results for the mechanical pre-cooling using the air conditioner (by adjusting the air-conditioner set points) in conjunction with economizer operation will be discussed below

Indoor Air Temperature

Figure 16 shows simulated pre-cooling results for one 24-hour period in climate zone 3B (El Paso, TX). The indoor air temperature is plotted for the three different pre-cooling regimes. The reference case (blue line) has no pre-cooling and so the indoor air temperature rises to the indoor set point just after midday when the air-conditioner turns on. The long pre-cooling period (red line) shows the air-conditioner turning on at 8am and keeping the house temperature at around 22.2°C up until the peak period begins at 4pm. Then the thermostat set point increase and the air-conditioner switches off. In this case the compressor remains switched off until around 6pm so 50% of the cooling peak load was removed. The green and purple lines show the medium (11am to 4pm) and short (1pm to 4pm) pre-cooling periods respectively. The indoor temperature for the medium pre-cooling case is the same at the start of the peak period (4pm) as for the long pre-cooling case. This indicates that the longer pre-cooling period was unnecessary because the air conditioner was sized so that it could reduce the internal air temperature to the same point in both long and medium cases. The short pre-cooling period only brings the indoor temperature down to 24.3°C in the time available, and so the air conditioner switches back on around one hour into the peak period.

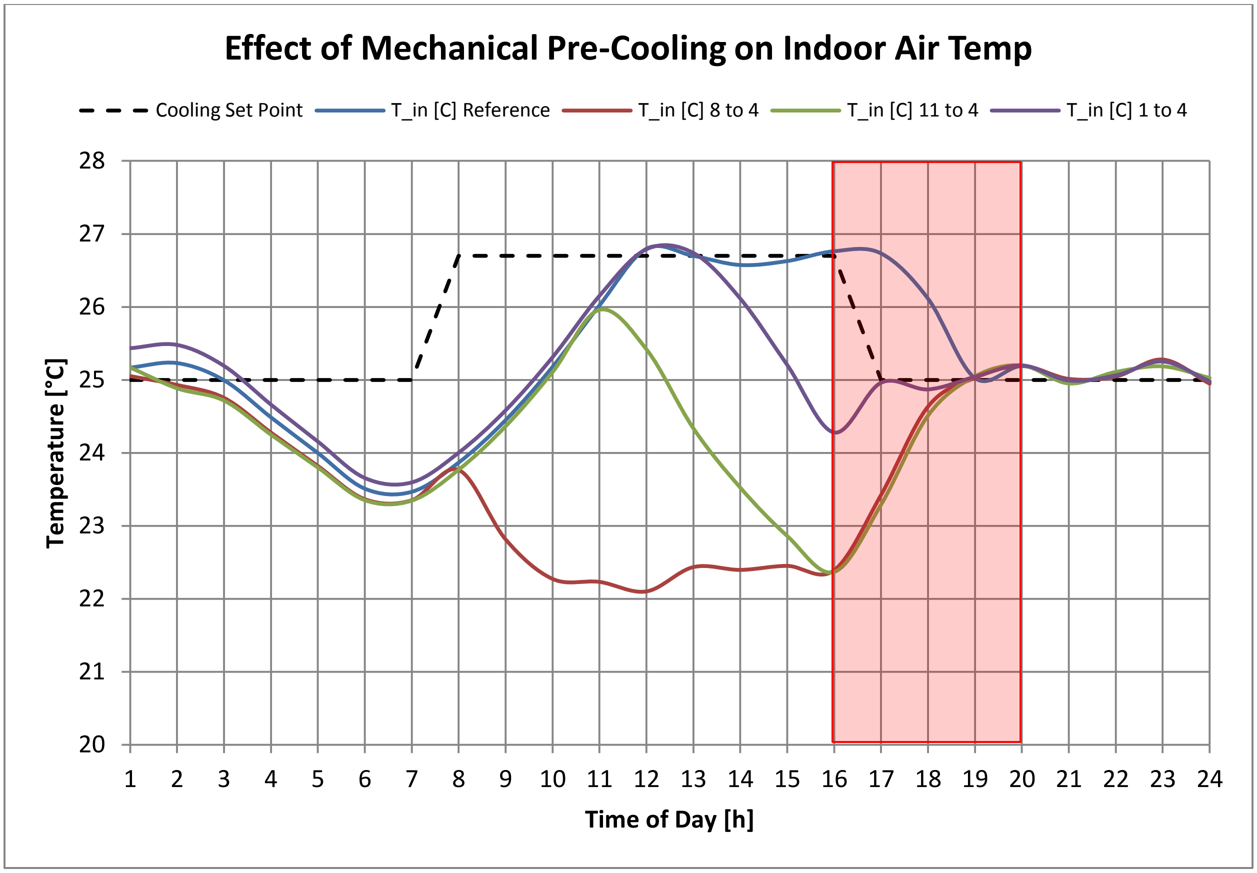


Figure 16: Effect on indoor air temperature of changing the cooling set points at different times to produce pre-cooling. 22.2°C or 72°F set point used, climate zone 3B – El Paso, TX. The dashed line shows the standard (no pre-cooling) cooling set point temperatures throughout the day. The red box indicates the peak period.

Absolute Peak Period Energy Reductions

The absolute peak period energy reductions (ΔE_P) are shown in Table 4. ΔE_P was calculated by summing all the energy consumed by the house during all of the cooling peak periods with no pre-cooling, and then subtracting the energy consumed during the coincident periods from the simulations with pre-cooling. We can see that the most absolute energy savings are to be had in the warmer climate zones of 1A, 2A, 2B, 3A and 3B where mechanical cooling is most prolific. The most peak energy saved is 2,080 kWh in climate zone 1A (Miami, FL). Peak savings are still possible in the climate zones with warm summers such as 4A (Baltimore, MD) and 4B (Albuquerque, NM). Obviously, the climate zones with very little air-conditioning use (e.g. 3C – San Francisco, CA and 8 – Fairbanks, AK) see the lowest peak energy reductions.

Table 4: Absolute peak period energy reductions, ΔE_P [kWh], compared to the baseline case with no pre-cooling (Peak Period = 4 pm to 8 pm). Long, medium and short are the pre-cooling lengths (8 hours, 6 hours and 4 hours respectively)

	ΔE_P [kWh]: 22.2°C			ΔE_P [kWh]: 23.3°C		
CZ & Reference City	Long	Medium	Short	Long	Medium	Short
1A. Miami, FL	2,080	2,080	2,080	1,860	1,860	1,850
2A. Houston, TX	1,960	1,910	1,670	1,780	1,780	1,620
2B. Phoenix, AZ	1,620	1,610	1,520	1,280	1,280	1,260
3A. Memphis, TN	1,010	960	790	860	850	750
3B. El Paso, TX	1,140	1,130	1,070	920	920	910
3C. San Fran, CA	10	10	10	10	10	10
4A. Baltimore, MD	640	590	470	580	550	460
4B. Albuquerque, NM	630	570	450	540	510	420
4C. Salem, OR	160	160	140	120	120	110
5A. Chicago, IL	370	330	280	370	330	280
5B. Boise, ID	400	390	350	300	300	270
6A. Burlington, VT	630	550	430	600	530	430
6B. Helena, MT	180	180	180	120	130	130
7. Duluth, MN	80	70	60	70	60	50
8. Fairbanks, AK	20	20	20	10	10	10

$1500 \le \Delta E_P$
$1000 \le \Delta E_P < 1500$
$500 \le \Delta E_P < 1000$
$\Delta E_P < 500$

Fractional Peak Period Energy Reductions

The fractional peak period energy reductions (R) are shown in Table 5. They were calculated by summing all of the energy consumed during the annual cooling peak periods, and then comparing the energy consumed during the coincident periods for the non-pre-cooling reference simulations and expressing as a percentage of the non-pre-cooling energy consumption for the same time period. Pre-cooling to 22.2°C can remove 97% of the annual peak cooling energy in climate zone 1A (Miami, FL). Generally, the longer the pre-cooling period and the lower the pre-cooling set point temperature the larger the peak savings. However, these are offset somewhat by the increased energy consumption during the pre-cooling periods when the air-conditioner would not normally be running. The averages across all climate zones for the 22.2°C set points are 86% for the long pre-cooling period, 82% for the medium and 74% for the short. For the higher 23.3°C set point the averages are 72% for the long, 70% for the medium and 64% for the short.

Table 5: Fractional peak period energy reductions, R [%], compared to the baseline case with no pre-cooling (Peak Period = 4 pm to 8 pm)

	R [%]: 22.2°C			R [%]: 23.3°C		
CZ & Reference City	**Long**	**Medium**	**Short**	**Long**	**Medium**	**Short**
1A. Miami, FL	97	97	97	86	87	86
2A. Houston, TX	91	89	78	84	83	75
2B. Phoenix, AZ	59	59	56	47	47	46
3A. Memphis, TN	90	85	71	77	76	67
3B. El Paso, TX	85	84	80	69	69	68
3C. San Francisco, CA	84	84	84	57	57	57
4A. Baltimore, MD	93	85	69	83	80	66
4B. Albuquerque, NM	87	78	62	73	69	58
4C. Salem, OR	83	82	70	61	61	56
5A. Chicago, IL	90	81	69	89	79	67
5B. Boise, ID	77	75	66	57	56	51
6A. Burlington, VT	78	68	54	75	66	53
6B. Helena, MT	91	91	90	65	65	65
7. Duluth, MN	86	77	63	79	70	57
8. Fairbanks, AK	96	96	96	81	81	81

90 ≤ R
80 ≤ R < 90
70 ≤ R < 80
R < 70

Peak-to-Penalty Energy Ratio

In order to quantify the pre-cooling trade-off between peak period energy saved and extra off-peak energy used, we have introduced the 'peak-to-penalty' energy ratio Γ:

$$\boldsymbol{\Gamma} = \frac{|\boldsymbol{\Delta E_P}|}{\Delta E_P + \Delta E_C} \qquad \text{Equation 1}$$

Where: ΔE_C = extra off-peak cooling energy (i.e. energy penalty) [kWh]

ΔE_P, = peak energy reduction [kWh]

When Γ is equal to one, each *additiona*l kWh of energy used from pre-cooling removes one kWh of energy used during the peak period. Figure 17 shows the house cooling energy use for two simulations – one with no pre-cooling (purple) and one with pre-cooling (gray). The peak period is confined within the red box. With no pre-cooling there is a sharp rise in energy use during the peak period. With pre-cooling there is high energy use before the peak period, but then this drops during the peak period.

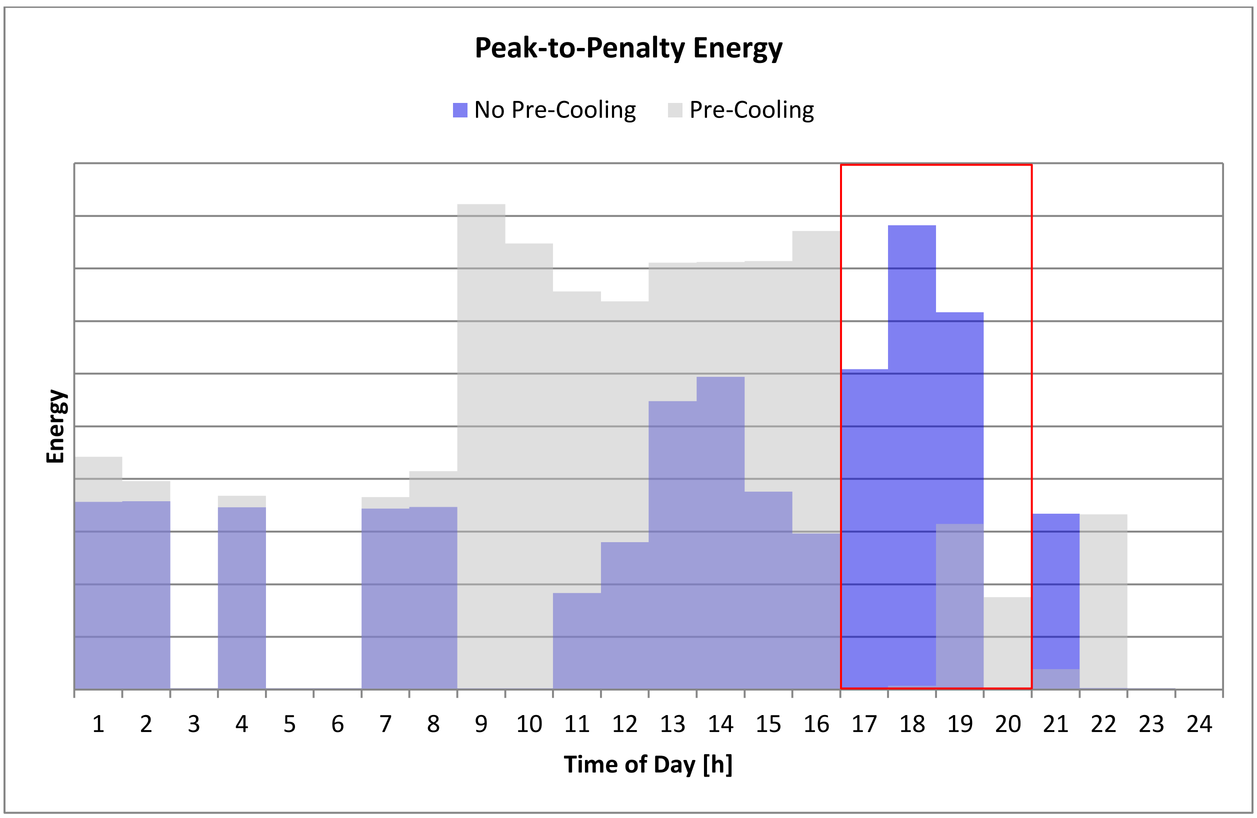


Figure 17: The purple area represents the average hourly building energy use with no pre-cooling. The grey area represents the energy use with pre-cooling. $|\Delta E_P|$ is the magnitude of the difference between the purple area and the grey area during the peak period (red box). ΔE_C is the difference between the purple area and the gray area outside the peak period.

Table 6 shows the peak-to-penalty energy ratio for the different pre-cooling strategies. The total energy use of the house during any pre-cooling regime is always greater than the total energy use of the house with no pre-cooling regime – irrelevant of the length of the pre-cooling period or the pre-cooling temperature set point, i.e., Γ is always less than one. It is generally highest for the shorter pre-cooling periods with the higher thermostat set point of 23.3°C. Γ decreases as the pre-cooling time periods get longer and the pre-cooling set point gets lower. The rate of change of Γ is non-linear with the pre-cooling time period, suggesting that shorter, warmer pre-cooling periods are more efficient at removing peak load, although they remove less peak load in total. In terms of guiding selection of an optimum pre-cooling strategy, we want to select the higher peak savings that have the least energy penalty. It is clear from these results that many cases combine both peak and total energy savings. A reasonable limit, then is only to use options that have $\Gamma > 0.5$. Because the range of absolute peak energy savings for a given climate is not very large (Table 4), in most cases the less extreme temperature settings and shorter cooling periods are optimum. When trying to maximize peak energy savings and minimize total cooling energy, our results showed that the high cooling climates (zones 1A-3B) gave the best results for the shortest pre-cooling time period and least temperature depression. Climates with less cooling showed better results for pre-cooling with the least temperature depression. Only one Climate Zone – 3C - showed no advantage with pre-cooling strategies, due to the low air conditioning load.

Table 6: Peak-to-Penalty energy ratio for all 15 climate zones

	Γ : 22.2°C			Γ : 23.3°C		
Reference City	Long	Medium	Short	Long	Medium	Short
Miami, FL	0.37	0.44	0.56	0.56	0.62	0.73
Houston, TX	0.61	0.67	0.72	0.70	0.75	0.81
Phoenix, AZ	0.48	0.58	0.70	0.66	0.73	0.82
Memphis, TN	0.55	0.60	0.65	0.63	0.69	0.75
El Paso, TX	0.58	0.62	0.70	0.68	0.72	0.79
San Fran, CA	0.10	0.10	0.10	0.27	0.27	0.27
Baltimore, MD	0.54	0.56	0.59	0.65	0.69	0.73
Albuquerque, NM	0.56	0.57	0.58	0.67	0.69	0.72
Salem, OR	0.49	0.49	0.46	0.74	0.74	0.74
Chicago, IL	0.49	0.49	0.46	0.74	0.74	0.74
Boise, ID	0.68	0.69	0.69	0.78	0.78	0.79
Burlington, VT	0.57	0.59	0.62	0.69	0.71	0.75
Helena, MT	0.58	0.57	0.59	0.77	0.77	0.77
Duluth, MN	0.38	0.37	0.35	0.67	0.68	0.67
Fairbanks, AK	0.31	0.31	0.31	0.76	0.76	0.76

$0.75 \leq \Gamma < 1.00$
$0.50 \leq \Gamma < 0.75$
$0.25 \leq \Gamma < 0.50$
$0 \leq \Gamma < 0.25$

Potential Carbon Savings

The net effect of mechanical pre-cooling using the air conditioner is always to increase the total annual energy use of the house. However, by removing peak loads the need for running higher carbon producing power plants during peak periods (in some US States but not all) can be reduced. This means that the net carbon output can be reduced, even though more energy is used.

Under some crude assumptions we can make some hypothetical calculations for the carbon reduction from pre-cooling. The carbon intensity of a state's energy supply is reflective of its energy-fuel mix. The states with high carbon intensities tend to be the states with high per capita emissions (DOE, 2012). The US national average carbon intensity in 2009 was 54.2 kilograms of carbon dioxide per giga-joule (kg CO_2/GJ). Vermont is a state that has a large proportion of nuclear energy. Its carbon intensity in 2009 was 32.2 kg CO_2/GJ. A gas-fired power station, suitable for use during the peak period, typically has a carbon intensity of 68.0 kg CO_2/GJ (Bilek et al., 2006). We can then demonstrate that pre-cooling can lead to a reduced carbon output despite using more total energy.

For the pre-cooling results from Vermont, multiply the total off-peak cooling energy used (converted into giga-joules) by the carbon intensity of Vermont (typical of low carbon off-peak state). Then add to this the peak energy used multiplied by the carbon intensity of a typical gas-fired power plant. The total is an approximation for the annual carbon output for the home due to air conditioner use (Table 7). The results show that pre-cooling can reduce the total carbon output from air conditioner use despite a net increase in energy consumption.

Table 7: Hypothetical carbon savings from pre-cooling in Vermont

Cooling Set Point [°F]	Pre-Cooling Times	Annual Difference in CO_2 [kg]
72	08:00 - 16:00	-26.0
72	11:00 - 16:00	-26.2
72	13:00 - 16:00	-25.6
74	08:00 - 16:00	-45.7
74	11:00 - 16:00	-43.3
74	13:00 - 16:00	-38.5

Conclusions & Recommendations

Two pre-cooling strategies have been analyzed:

1. Night ventilation using an economizer
2. Mechanical pre-cooling using the house air-conditioner, in conjunction with night ventilation using an economizer.

The results are summarized as follows:

- The results showed that a residential economizer can save more than 2000 kWh/year of cooling energy in climates with large diurnal temperature swings
- In some cases the economizer energy savings offset all of the mechanical ventilation related energy use. Namely - climate zones 2A, 2B, 3B and 3C for a BPM motor and 2A for a PSC motor
- Using a high-performance BPM air handler in the HVAC system saved an additional 12% of annual ventilation energy compared to more typical PSC air handlers
- Economizers may cause problems with excess humidity in climate zones 2A (Houston, TX) and 3A (Memphis, TN) due to increases in indoor humidity, so their use in these climates should be restricted to homes with independent humidity control
- Mechanical pre-cooling using the air conditioner can remove up to 97% of the peak cooling load at the settings tested. This is heavily dependent on climate zone, the length of the pre-cooling period and the pre-cooling set point
- In order to maximize peak energy savings while using the least amount of cooling energy, the high cooling climates (zones 1A-3B) should use a short pre-cooling time period (3 hours) with the smallest temperature depression (23.3°C (74°F) set point)
- Climates with less cooling demand should use a longer pre-cooling time period (8 hours) with the smallest temperature depression (23.3°C (74°F) set point)
- Pre-cooling is not recommended in Climate Zone 3C (San Francisco, California) or Climate Zone 8 (Fairbanks, Alaska). There was no advantage from pre-cooling due to near-zero air conditioning use in these climates

- One caveat with the pre-cooling recommendations is that they are for lightweight wooden-framed homes. The recommendations may change for heavier brick/block structures not included in this study.

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Appendix A: Additional Simulation Details

The following are additional simulation details that have been moved to improve readability of the main document.

Envelope Leakage

The leakiness of the house in this report was based on recent studies by Offerman (2009) and Wilcox (2011), that show air leakage of about 4.8 ACH_{50} is typical for new construction.

Leakage distribution was one-quarter in the floor, one-quarter in the ceiling, and half in the walls. There were no open flues, fireplaces or windows (although passive stacks were included in certain simulations). The garage was omitted from the simulations and treated as outside, unconditioned space.

Insulation and Fenestration

R-Values of walls and ceilings, U-Factors and Solar Heat Gain Coefficients (SHGC) for windows were based on the IECC 2009 values. House insulation used to determine the non-ventilation building load varied by climate (see Table 8).

Table 8: House Insulation Levels from IECC 2009 Table 402.1.1

Climate Zone	Representative City	Glazing		Ceiling	Walls	Ducts Outside Conditioned Space
		U	SHGC			
1A	Miami, FL	0.65	0.3	R30	R13	R8
2A	Houston, TX	0.65	0.3	R30	R13	R8
2B	Phoenix, AZ	0.65	0.3	R30	R13	R8
3A	Memphis, TN	0.50	0.3	R30	R13	R8
3B	El Paso, TX	0.50	0.3	R30	R13	R8
3C	San Francisco, CA	0.50	0.3	R30	R13	R8
4A	Baltimore, MD	0.35	0.3	R38	R13	R8
4B	Albuquerque, NM	0.35	0.3	R38	R13	R8
4C	Salem, OR	0.35	0.3	R38	R20	R8
5A	Chicago, IL	0.35	0.3	R38	R20	R8
5B	Boise, ID	0.35	0.3	R38	R20	R8
6A	Burlington, VT	0.35	0.3	R49	R20	R8
6B	Helena, MT	0.35	0.3	R49	R20	R8
7	Duluth, MN	0.35	0.3	R49	R21	R8
8	Fairbanks, AK	0.35	0.3	R49	R21	R8

Exterior surface area for wall insulation scales with floor area and number of stories. The total building surface area is typically three times the floor area (based on the BSC/Building America data set). A simple rule of thumb developed from measured data from several thousand new homes and from the simplified Title 24 prototype C in the Alternative Compliance Manual is that the wall area is typically 1.22 times the floor area for a one-story home. Window area was 20% of floor area with windows equally distributed between North, South, East and West. Clear glazing was simulated with exterior shading of 50%. There was a 40 ft^2 door for the dwelling, assumed to face north and with a U-Factor of 0.50.

HVAC Equipment

Below are the details of the heating, cooling and mechanical ventilation systems used in the simulations.

Whole-House and Auxiliary Ventilation

All mechanical ventilation systems used are listed in Table 9, and were chosen from the Home Ventilation Institute Directory (HVI, 2011). The whole-house fan used (dependent on ventilation strategy) was sized to meet the ASHRAE 62.2 minimum airflow rate. The RIVEC fan used in the hybrid ventilation strategy was sized to be 1.25 times the ASHRAE 62.2 minimum because it would operate intermittently.

We included intermittent operation of bathroom, kitchen and clothes dryer fans, also sized to meet ASHRAE 62.2. Bathroom fans operated at 25 L/s (50 cfm). Kitchen range hood fans had airflow rates of 50 L/s (100 cfm). Clothes dryer fans were simulated as exhaust fans with an airflow rate of 75 L/s (150 cfm).

Table 9: Ventilation equipment for the different simulation houses (HVI, 2011)

System	Equipment	Airflow Rate		Power [W]
		[L/s]	[cfm]	
Whole-House Fan	*Panasonic, FV-08VKS2*	28	60	11.8
RIVEC Fan	*Panasonic, FV-08VKS2*	33	70	14
Kitchen Range Hood	*Venmar, ESV1030BL*	47	100	37.2
Bathroom Exhaust	*Panasonic, FV-08VKM2*	24	50	10.2
Clothes Dryer	N.A.	71	150	-

Heating and Cooling

Heating and cooling equipment was sized according ACCA Manuals J & S (ACCA, 2006). For heating we used a minimally efficient 80% AFUE natural gas furnace. For cooling, we used a SEER 13 split-system air conditioner with a TXV refrigerant flow control. Heating and cooling ducts were located in the attic. The total duct leakage was 6%, evenly split with 3% supply leakage and 3% return leakage.

Field studies by LBNL, Proctor and Parker (2000) (245 systems) and Philips (1998) (71 systems) have shown that existing fans in residential PSC air handlers typically consume 500W or more of electricity and supply about 2 cfm/W. For BPM-driven air handlers we used 6cfm/W and the same airflow rate as the PSC air handlers.