White Paper

Optimum Solutions for Residential Mechanical Ventilation - Phase 2 of CTRSC Research Project

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EXECUTIVE SUMMARY

In the three decades since 1990, residential mechanical ventilation for the whole dwelling has come into much greater focus. Local exhaust in bathrooms and toilet rooms has typically been well accepted by home builders to control buildup of moisture and odor. Whole dwelling unit ventilation, however, can be accomplished many ways, each having pros and cons related to indoor air quality, thermal and moisture comfort and performance, operational verification, initial cost, operating cost, and occupant operation, maintenance, and satisfaction. Climate effects add another layer of complications, making the choices for home builders quite complex and the decisions hard to nail down.

This study examined the operating cost difference between no dwelling unit mechanical ventilation and seven different whole-dwelling unit mechanical ventilation systems (see Section 6.1), with additional consideration of moisture control effects in 11 climate locations. The pros and cons, computer modeling assumptions and analysis results for the ventilation systems are described and discussed, and final recommendations are provided for optimizing residential mechanical ventilation.

All of the dwelling unit mechanical ventilation systems examined are feasible. However, for a particular dwelling unit and location there are both economic and non-economic reasons to prefer one system over another based on best practices related to indoor air quality, building science, optimal occupant comfort, lower first-cost, or lower operating cost. Ultimately, the builder must know their market and choose a system based on market preferences and constraints.

From a high-level perspective without complex caveats, the following table shows the recommended systems and associated climate zones resulting from the seven mechanical ventilation systems modeled:

Ventilation System		Climate Zone
No.	**Description**	
2	Hybrid Central-Fan-Integrated Supply (CFIS) with automatic Exhaust backup	1-5
3	Balanced Energy Recovery Ventilator (ERV) with dedicated ducts	1-8
4	CFIS-33% baseline with occupant-controlled Exhaust	1-5
7	Balanced ERV with AHU interlock	3B. 4C, 5-8

Estimated ventilation system first-cost ranged by a factor of 10 from $190 for single-point exhaust to $1,875 for a ventilating dehumidifier supply with compressor or air-handler unit interlock.

The difference in total annual building energy cost relative to the Vent1-Exhaust reference case ranged from a savings of $37 for the Vent4-CFIS-33% case in Washington DC-Reagan to an increase of $282 for the Vent5-Ventillating Dehumidifier in Miami. The percent cost difference in total annual building energy cost from the Vent1-Exhaust reference case ranged from a savings of 2.5% to an increase of 7.2% for all systems except for the Vent5-Ventilatiing Dehumidifier in Miami and Houston, where costs increased by 18% and 11%, respectively.

If setting aside the first-cost and operating cost disbenefit, the Vent5-Ventilating Dehumidifier as modeled here can be an effective ventilation system for humid locations. As another example, the Vent3-Balanced ERV with dedicated duct system will work in every climate but it is one of the most expensive systems to install and lacks the benefit of whole-house recirculation filtration for particulate removal and mixing for improved indoor comfort. That may lead one to choose the Vent7-Balanced ERV integrated with the central duct system and with central system fan interlock, but that has higher energy consumption and negative humidity control consequences in the humid climates due to evaporation of water from wet coils when the cooling compressor is not operating. On the other end of the spectrum, the Vent1-Exhaust system has the lowest first-cost and also one of the lowest operating costs, however, there are drawbacks in: a) indoor air quality performance since the source of outdoor air is not controlled and the air may be bringing pollutants with it; b) poor ventilation air distribution performance; and c) lack of recirculation air filtration and comfort mixing.

Except for the coldest climates, a middle ground may be the Vent4-CFIS-33% baseline system with occupant-controlled exhaust that has moderate first-cost, low operating cost, good air filtration and comfort mixing characteristics, but, if the occupant does not activate the occupant-controlled exhaust, the resulting average ventilation rate will be lower than what may be specified by code. That may lead one to choose the Vent2-Hybrid CFIS with automatic exhaust backup which increases first-cost by nearly $200 and the recirculation filtration and comfort mixing benefit is reduced.

For balanced mechanical ventilation systems, simple payback for energy recovery ranged from 3.7 years in Fairbanks to 12.3 years in San Jose. Simple payback for installing a separate ERV duct system versus interlocking with the AHU ranged from 2.4 years in Fairbanks to 5.7 years in Seattle.

High efficiency air filtration via the central space conditioning system and ventilation rates were discussed in terms of research gaps and risks that may come with code changes. Without updated return air design considerations and minimum 4" to 5" wide pleated media filter assemblies, using the central space conditioning system to accomplish high efficiency air filtration will likely force most systems outside of their rated range of external static pressure. Mini-split and multi mini-split heat pump systems are especially susceptible to this challenge because their fan systems are designed for no ducting or limited ducting and low filter resistance.

Raising the ventilation rate beyond the current IRC code level would likely force universal supplemental dehumidification to maintain indoor relative humidity below 60% in humid climates, and especially in coastal humid locations.

1 INTRODUCTION

The 2018 International Energy Conservation Code (IECC 2018) and the 2018 International Residential Code (IRC 2018) set limits on building enclosure air leakage – tested to be not more than 5 air changes at 50 Pascal pressure differential (ACH50) in Climate Zones 1-2, and not more than 3 ACH50 for Climate Zones 3-8. Air distribution system duct leakage, which can cause uncontrolled building air exchange, must also be tested to meet limits set by the 2018 IRC and IECC.

Acknowledging the benefit of having less uncontrolled air exchange, but in support of a minimum amount of controlled air exchange for improved indoor air quality, the 2018 IRC requires dwelling unit mechanical ventilation for one- and two-family dwellings (IRC Sections N1103.6 and M1505), and the 2018 International Mechanical Code (IMC) requires dwelling unit mechanical ventilation for Group R-2, R-3, and R-4 occupancies (IMC Section 403.3.2). Where mechanical ventilation is provided it must meet the prescribed requirements. The IECC points to the IRC or IMC for mechanical ventilation requirements except for some damper and fan efficacy requirements (IECC Section R403.6, Table 405.5.2(1)).

States have discretion to adopt codes published by the International Code Council (ICC) in full or in part, or to develop and adopt their own requirements. As such, some states, such as California and Washington, have established mechanical ventilation requirements of their own, differing in provisions such as ventilation rate, distribution of ventilation air, and type of system.

Local exhaust in bathrooms and toilet rooms has typically been well accepted by home builders to control buildup of moisture and odor. However, there are many ways to accomplish whole dwelling unit ventilation, each having pros and cons related to indoor air quality, thermal and moisture comfort, energy performance, operational verification, initial cost, operating cost, and occupant operation, maintenance, and satisfaction. Climate effects add another layer of complications, making the choices for home builders quite complex and the decisions hard to nail down.

This study examined the operating cost difference between no dwelling unit mechanical ventilation and seven different dwelling unit mechanical ventilation systems, with additional consideration of moisture control effects (Rudd 2011). The pros and cons, computer modeling assumptions, and analysis results for the ventilation systems are described and discussed. Final recommendations for optimizing residential mechanical ventilation are provided.

2 ASHRAE STANDARD 62.2 CONSIDERATIONS

The American Society of Heating, Refrigeration and Air Conditioning (ASHRAE) publishes Standard 62.2 titled, "Ventilation and Acceptable Indoor Air Quality in Residential Buildings." The Standard is written as a "code-intended standard." It is not a building code, however, parts of it have been adopted or adapted into the ICC codes and some state codes.

The first version of Standard 62.2 was published in 2003. The current published version is 2019. The Standard is in continuous maintenance and a new version is published every three years. Addenda published between the Standard publications are officially considered part of the Standard.

For the most part, ICC model codes and most state codes refer to the ventilation rate found in the 62.2-2010 version before it increased about 40% on average in the 2013 version (BSC et al. 2014). The State of California refers to the higher rate. While this higher rate may not be a concern in the populous mild, dry climates in California, it certainly is a concern in more severe climates especially those with high outdoor humidity. Higher ventilation rates combined with reduced sensible cooling load in warm, humid climates force the need for supplemental dehumidification (Rudd 2013, Rudd 2014).

For buildings tested for air leakage, different forms of ventilation rate credit (reduction) are allowed for uncontrolled infiltration air, however, air sealing requirements are aimed at precluding air infiltration that comes through secondary spaces with likely contamination, such as attached garages, unconditioned crawlspaces, unconditioned attics, and other dwellings. For multifamily dwellings, the infiltration credit is limited by the amount of exterior wall area.

Another credit allowed against dwelling unit mechanical ventilation rate is for enhanced filtration of recirculated air within the dwelling. A 20% ventilation rate reduction may be applied depending on the filter efficiency level and the amount of air recirculation.

New proposals aimed at the 2021 version of ASHRAE Standard 62.2 would limit the dwelling unit ventilation system to supply-only or balanced systems to limit mechanically induced transfer air not directly from outdoors, such as from adjacent dwelling units or common corridors. An important distinction for exhaust-only systems between single-family detached and multi-family attached dwellings is the extent of a dwelling's ventilation air that is affected by air from adjacent dwelling units or common areas.

3 INDOOR AIR POLLUTANTS

The most recent research on the topic of indoor air pollutants (Logue, 2011) has identified the primary residential indoor air pollutants of concern to be:

1. Respirable particles of size 2.5 micron (one micron equals one millionth of a meter) or less, referred to as PM2.5. Particles of this size are present both outdoors and indoors, but they can be reduced by filtration in the indoor environment.
2. Second-hand tobacco smoke;
3. Acrolein, a gaseous chemical mostly emitted from materials and cooking; and
4. Formaldehyde, a gaseous chemical mostly emitted from materials;

According to Logue, in non-smoking dwellings, PM2.5, acrolein, and formaldehyde make up about 80% of the total health damage risk for 90% of the sample sets evaluated. This was determined using the Disability Adjusted Life Year (DALY) metric which allowed quantification of overall disease damage in terms of equivalent years of life lost to illness or disease.

PM2.5 was by far the most important, having a DALY value 13 times that of acrolein and formaldehyde.

The dwelling unit mechanical ventilation rate in ASHRAE Standard 62.2 was established based only on the collective experience and judgment of the committee members, not on any quantifiable health-specific concerns related to pollutant concentrations. Other than the relatively new ventilation rate credit for enhanced indoor air filtration that was based on the DALY analysis, that is still the case.

4 SOURCE CONTROL

4.1 Source Control by Design

Keeping indoor air pollutants out of the living space in the first place should be the highest priority in the building design process and during occupancy. That is more effective and energy efficient than trying to dilute pollutants by mechanical ventilation. Careful attention should be given to the following practices to avoid or control major sources of indoor air contamination:

- Choose building materials and finishes that are known to reduce pollutant emissions.
- Ensure dry basements, crawlspaces, and walls.
- Provide an airtight separation between living spaces and attached garages, vented crawlspaces, and vented attics.
- Build to limit the need for strong pesticides. Leave little space for pest entry. Use flashing to protect against pests at the foundation-to-wall interface.
- Educate homeowners to use cleaning agents wisely and with adequate ventilation either by exhaust fans or windows.

4.2 Source Control by Local Exhaust

Kitchens, bathrooms, toilet rooms, and laundry rooms are places where pollutants are generated in high concentration. These pollutants include particles, chemicals, moisture, bio-effluent contaminants from people and animals and objectionable odors. When there are activities in these living space areas, pollutants should be exhausted directly to the outdoors before they can negatively impact air quality elsewhere in the home. Typically, this is achieved with range hoods in kitchens, and exhaust fans in bathrooms and laundry rooms. The exhaust fan can be ceiling- or wall-mounted in the space, remote-mounted, and pulling from one location or multiple locations in parallel. While a purposefully airtight separation is paramount, exhaust ventilation of attached garages should also be considered, especially if there is a relatively large area of wall or ceiling area between the garage and living space. Local exhaust must be ducted all the way to outdoors.

Devices such as occupancy and humidity sensors, time delay controllers and light switch interlocks are available to better manage the operation of exhaust fans, but homeowner education is the most effective at ensuring correct operation and usage of these devises. Builders should educate homeowners on the importance of using local exhaust whenever they are actively using kitchens, bathrooms, and laundry rooms. Time delay controllers which keep an exhaust fan energized for a time after the room is vacated are available. These devices are especially beneficial in bathrooms and toilet rooms. Kitchen exhaust should always be used while the range or oven is operating to avoid spreading the cooking emissions into the living space. For gas cooktops and ovens, the gas combustion pollutants add to the cooking pollutants. Also, builders should ensure that the clothes dryer has good airflow (minimal restrictions) and that the exhaust air goes directly to outdoors. It may be necessary to use an inline dryer exhaust booster fan system.

In the 2018 IRC code, local exhaust ventilation rates are shown in Table 1.

Table 1. 2018 IRC local exhaust ventilation rates

	Continuous	Intermittent (on occupant demand)
Kitchen	25 CFM	100 CFM
Bathroom, Toilet Room	20 CFM	50 CFM

4.2.1 <u>Exhaust-Only</u>

An exhaust-only mechanical ventilation system expels indoor air directly to outdoors without any powered makeup air. This tends to depressurize the interior space relative to outdoors. The actual ventilation air is then unpowered makeup air via the paths of least resistance created by gaps, cracks, and openings in the building enclosure. These gaps, cracks, and openings may be directly connected to outdoors, as in the case of gaps around exterior windows and doors, or indirectly connected to outdoors through wall, ceiling, and floor assemblies that may also be connected to garages, vented attics, crawlspaces, and, to a lesser extent, foundations in contact with soil.

Single-Point Exhaust (modeled in System 1)

Single-point whole-house exhaust ventilation most commonly entails a high-quality bath fan installed in a master bathroom, family bathroom, powder/toilet room or laundry area. In some cases, a dedicated fan will be installed in a ceiling location in the central area of the house. These fans are generally quiet, rated for continuous duty, and have low power draw. These fans are typically surface-mounted, on a ceiling or wall, or remote-mounted inline fans. Some have multiple speeds to allow for double-duty use as both the bath/toilet room fan and the whole-house fan.

5 DWELLING UNIT MECHANICAL VENTILATION

The function of dwelling unit mechanical ventilation is to dilute remaining diffuse pollutants after local exhaust has removed concentrated pollutants at their source. Since these remaining dwelling unit pollutants are dispersed, or diffuse, there is no practical way to capture and exhaust them as in a bathroom or over a kitchen cook-top. Rather, dilution is used to reduce the concentration of those pollutants throughout the home.

The intent is that less polluted outdoor air is distributed in a controlled manner to dilute the more polluted inside air. Whole-house ventilation can be operated continuously at a lower rate, or intermittently at a higher rate.

Ventilation equipment must be sized and installed correctly and maintained. Ventilation rates that are too low may result in inadequate fresh air for a healthy living environment resulting in poor odor and moisture control. Ventilation rates that are higher than needed will tend to waste energy, cause homes to be too dry in dry climates or during the wintertime in cold climates, and add excess humidity in warm, humid climates, which if not removed by cooling and dehumidification equipment can result in mold activity.

ANSI/ASHRAE Standard 62.2 establishes the whole-building ventilation rate based on two main factors:

- The number of bedrooms – being used as an indicator of occupancy, considering 2 people in the primary bedroom and 1 person in each secondary bedroom
- The conditioned floor area of the dwelling.

These two parts are added together. As mentioned before, most building codes apply the rate given in the 2010 version of the Standard, as follows:

(7.5 CFM)(Number of bedrooms +1) + (0.01)(Conditioned Floor Area)

The rate given in the current 2019 version of the Standard is as follows:

(7.5 CFM)(Number of bedrooms +1) + (0.03)(Conditioned Floor Area)

5.1 Types Of Dwelling Unit Mechanical Ventilation

There are many ways to accomplish dwelling unit mechanical ventilation. The three main types of systems are fan-powered outdoor air supply, fan-powered indoor air exhaust, and fan-powered balanced. It may be possible to design and implement a functional, reliable, and comfortable dwelling unit ventilation system without fan power, but that would require an engineered solution that is beyond the scope of this study. Balanced ventilation involves an approximately equal amount of supply and exhaust. These types of ventilation systems can be combined in "hybrid" configurations to improve effectiveness and energy efficiency. Home Ventilating Institute (HVI) administers a fan certification program that ensures quiet operation and proper airflow. Note that the heating and cooling system size is not affected much between ventilation systems when the hourly average outdoor ventilation airflow is the same (refer to the Manual J+S columns in Table 6 through Table 8).

The rest of this section describes the operating principles of the dwelling unit mechanical ventilation systems examined in this study.

5.1.1 Supply-Only

Supply whole-house ventilation systems draw outdoor air from a known location and deliver it to the interior living space. Supply mechanical ventilation tends to maintain a slight positive pressure in the conditioned space relative to outdoors. The outdoor air inlet location can be selected to maximize the ventilation air quality. The ventilation air can be filtered, heated, cooled, dehumidified, and actively or passively cleaned before being distributed to the living space.:.

Supply Integrated With The Central Space Conditioning System (modeled System 2 and System 4)
This system provides ventilation air through a duct that extends from a known fresh air location outdoors to the return air side of a central heating and cooling air distribution system. The system relies on negative pressure created in the return air ductwork by the central air handler fan during heating, cooling, or constant fan operation. The fan speed needs to be sufficient to draw in the intended amount of outdoor air. The central-fan-integrated supply (CFIS) system includes a motorized outdoor air damper and an automatic timer control to ensure ventilation air is periodically supplied when heating and cooling have been inactive and to limit outdoor air introduction to a maximum regardless of how long the fan operates. A manual balancing damper should be installed in the outdoor air duct to allow adjustment of the ventilation rate if necessary. This type of system tempers the outdoor air with recirculated indoor air. The air is filtered and sometimes conditioned depending on the coincident cooling and heating activity. The system achieves full distribution of ventilation air using the existing duct network.

Separate Supply Fan
A separate supply fan can be employed to deliver outdoor ventilation air to the conditioned space. Tempering of the outdoor air is essential to avoid comfort complaints and other problems. That can be achieved by selecting a fan size large enough to blend 2- to 3-parts indoor air with 1-part outdoor air depending on climate. If the intent is to deliver the outdoor air into the central system ductwork, then note that introduction of humid outdoor air into the central duct system by a separate supply fan can lead to condensation and mold in the ducts, and introduction of cold outdoor air by a separate supply fan can cause premature furnace heat exchanger. Those problems can be avoided by operating the central system fan at the same time, but then the question becomes, why not just use the CFIS system and avoid the separate supply fan?

Ventilating Dehumidifier Supply Fan (modeled System 5)

In humid climates, a ducted ventilating dehumidifier system can be used to deliver dry outdoor ventilation air to the main supply duct of the central space conditioning system. The dehumidifier fan continuously draws outdoor ventilation air to the dehumidifier appliance where the outdoor air is dehumidified if the incoming air dew-point temperature is higher than the target indoor air dew-point temperature, about 50 ^{o}F. When the outdoor air dew-point temperature is lower than the target indoor air dew-point temperature, the dehumidifier compressor does not need to operate but the central system fan operates at low speed to temper and distribute the continuous outdoor ventilation air.

An application of the ventilating dehumidifier that is not recommended here is to operate the dehumidifier fan constantly for ventilation but the compressor intermittently in response to the conditioned space relative humidity. In humid climates, running the dehumidifier fan constantly but the compressor intermittently causes excessive evaporation of previously condensed moisture from the dehumidifier's wet evaporator coil and drain pan during compressor off periods. That leads to energy inefficiency, wide indoor humidity oscillations, and potential condensation and mold in central system supply ducts.

5.1.2 Balanced

Balanced mechanical ventilation systems provide both exhaust and supply in roughly equal amounts. Indoor air is exhausted to outdoors and outdoor air is supplied to indoors. Balanced mechanical ventilation tends to impose neither a positive or negative pressure relationship between the indoors and outdoors. Dwellings with balanced ventilation tend to have a slightly higher overall outdoor air exchange due to the summing of natural air infiltration from wind and stack effects. The same as for supply ventilation, the outdoor air inlet location can be selected to maximize the ventilation air quality, and the ventilation air can be filtered before being distributed to the living space.

Balanced with Sensible and Latent Heat Recovery (modeled System 3 and System 6)

Balanced ventilation with only sensible heat recovery is accomplished with a heat recovery ventilator (HRV). HRVs use a non-moisture-sensitive heat exchanger to transfer heat between the exhaust air stream and the outdoor air supply stream, affecting only the temperature of the airstreams. With HRVs, no moisture is exchanged between the air streams. In the cold season, less heating will be needed, and in the hot season, less sensible cooling will be needed.

Balanced ventilation with both sensible and latent (moisture) recovery is accomplished with an energy recovery ventilator (ERV). ERVs operate the same as HRVs except that both heat and moisture are exchanged between the dwelling exhaust and outdoor air supply streams, affecting both the temperature and humidity of the airstreams. In the cold, dry season, less heating and humidification will be needed. In the hot, humid season less cooling and dehumidification (both sensible and latent cooling) will be needed. In simple terms, the heat and moisture tend to remain on the side from which they came. These systems can typically recover 50%-80% of the temperature and moisture difference between the dwelling exhaust and outdoor supply air. Note, however, that an ERV can neither cool nor dehumidify the interior space. Another important point about ERVs in humid climates is that, in the spring and fall seasons when indoor relative humidity is usually most elevated, the latent exchange benefit of the ERV is minimized due to relatively small differences between the indoor and outdoor absolute humidity.

HRVs and ERVs require fan energy to move two airstreams, exhaust and supply, through a heat exchanger. HRVs and ERVs can be ducted independently of the central space conditioning system, however, in practice, they are most often ducted into the central space conditioning system. Ducting into the central system requires interlocking controls and additional fan energy to synchronize operation between the central fan and the HRV or ERV. This synchronizing function can be through controls provided by the manufacturer or through setup options available with some thermostats. The extra central fan energy detracts from the sensible and latent heat recovery savings but has the benefit of further tempering the outdoor air and improving air filtration for the entire living space.

Balanced Without Sensible Or Latent Heat Recovery (modeled System 7)

Balanced mechanical ventilation does not necessitate heat recovery. The system may be as simple a CFIS system coupled with an exhaust fan, or a separate supply fan coupled with an exhaust fan. However, if using a separate supply fan, tempering of the outdoor air must be considered.

5.1.3 Hybrid Systems *(modeled System 2)*

Hybrid mechanical ventilation systems create combinations of exhaust, supply, and possibly balanced systems. For example, the hybrid system evaluated in this study involves a CFIS system operating with thermostat calls for heating and cooling automatically supplemented with exhaust mechanical ventilation when there is no call for heating or cooling.

5.1.4 Summary Pros and Cons for Dwelling Unit Mechanical Ventilation System Types

A summary matrix of dwelling unit mechanical ventilation system pros and cons is provided in Table 2. Other factors of initial cost and operating cost are addressed further on in this paper.

Table 2. Matrix of dwelling unit mechanical ventilation systems pros and cons (see Section 6.1 for the ventilation system details)

Ventilation system:	1	2	3	4	5	6	7
	Exhaust, single-point	Hybrid CFIS w/ automatic exhaust backup	Balanced ERV, dedicated ducts	CFIS-33% baseline w/occupant-controlled Exhaust	Ventilating Dehumidifier w/ compressor or AHU interlock	Balanced Supply and Exhaust, no recovery, no interlock	Balanced ERV w/ AHU interlock
Controlled source-path of outdoor air	No	Partial[2]	Yes	Yes	Yes	Yes	Yes
Filtration of outdoor air	No	Partial[2]	Yes	Yes	Yes	Yes	Yes
Whole-house distribution of outdoor air	No	Partial[2]	Yes	Yes	Yes	No	Yes
Tempering of outdoor air	Partial[1]	Partial[2]	Yes	Yes	Yes	No	Yes
Depressurization risk	Yes	Partial[3]	No	No	No	No	No

[1] While some exhaust makeup air will enter the dwelling unit essentially directly through cracks around windows and doors, much will enter indirectly through the materials in walls, ceilings, or floors and will transfer heat and moisture with those materials. Research has shown that entrained particulate and gas-phase contaminants are also transported with air passing through the building enclosure to the indoor environment (Rudd, 2014).

[2] Yes when CFIS is operating, No when exhaust-only backup is operating

[3] No when CFIS is operating, Yes when exhaust-only backup is operating

6 ANNUAL SIMULATION OF MECHANICAL VENTILATION SYSTEMS

This study focused on evaluating the energy consumption differences between mechanical ventilation systems and their impact on heating, cooling, and dehumidification. Seven mechanical ventilation system types were evaluated, plus a no-mechanical ventilation case for reference. The building modeled was from a national production builder plan: 2467 ft^2, 3 bedroom, 1-story, slab-on-grade. The overall result trends and recommendations were expected to be similar and applicable to buildings with basement or crawlspace foundations. The mechanical ventilation airflow rate of 55 CFM continuous was according to the IRC. Refer to Table 4.

The ResSizePro version of the Energy Gauge USA hourly simulation program was used in this study (EnergyGauge.com). One of the useful features of this program is a built-in dehumidifier model based on published results from laboratory testing done at the National Renewable Energy Laboratory. To evaluate moisture control differences between the systems, supplemental dehumidification was applied in all cases to keep the conditioned space at 55% relative humidity set-point (+/- 2% control deadband) when moisture removal by the cooling system would not accomplish that.

6.1 Model Input Descriptions for the Mechanical Ventilation Systems Simulated

A description of the mechanical ventilation systems examined by the simulations, and the underlying assumptions are listed in the following sections.

6.1.1 System 0: No-ventilation

The case of no mechanical ventilation was run as a reference point, although the exhaust-only system was used as the baseline for energy consumption and operating cost comparisons.

6.1.2 System 1: Continuous Exhaust

The continuous exhaust mechanical ventilation system was modeled with a ventilation rate of 55 cubic feet per minute (CFM) at 20 watts (W) power (2.8 CFM/W).

6.1.3 System 2: Hybrid Central-Fan-Integrated Supply with Automatic Exhaust Backup

This system was labeled a hybrid system because it involved CFIS ventilation (outdoor air duct to central system return and motorized damper in outdoor air duct) whenever there was a call for heating or cooling and exhaust ventilation outside of those times. The CFIS ventilation rate was 75 CFM at 0 W because it was coincident with demand for heating and cooling, and the automatic exhaust backup ventilation rate was 55 CFM at 20 W. In the EGUSA program, this system is called Runtime Vent with Backup.

6.1.4 System 3: Continuous Balanced Energy Recovery Ventilation with Dedicated Duct System

The continuous balanced energy recovery ventilation (ERV) system was modeled with an outdoor air ventilation rate of 55 CFM at 40 W (1.375 CFM/W). The recovery efficiency was set to 70% which is representative of an above-average efficiency ERV. The fan power is twice that of the continuous exhaust system since the ERV moves an equal amount of exhaust and supply air. This system was assumed to have a duct distribution system separate from the central space conditioning system so that the central system fan would not have to be operated simultaneously with the ERV. There was one interior exhaust air pickup and three interior outdoor air supply outlets. Considering the pressure drop of the ERV heat exchanger and the duct system, the ERV fan was assumed to be driven by an efficient electrically commutated motor (ECM). If the fan motor were a standard permanent split capacitor (PSC) motor, the power draw would be higher.

6.1.5 System 4: Central-Fan-Integrated Supply Baseline with Exhaust on Occupant Demand

This system involves the same exhaust fan of System 1 except that the exhaust fan was assumed to be operated only on occupant demand for local exhaust. A CFIS baseline system operated automatically in the background a minimum of 33% of the time including time coincident with central space conditioning demand for heating and cooling. The CFIS ventilation rate was 75 CFM, which was based on a 6” outdoor air duct and was 36% more than the IRC continuous rate. The power was determined by the EGUSA program to be 0 W when coincident with demand for heating and cooling and 0.375 W/CFM (2.67 CFM/W) without demand for heating and cooling. The EGUSA program determined the actual power based on the cooling CFM needed for a 15 SEER system sized in accordance with the 8th Edition of the Air Conditioning Contractors of America (ACCA) Manuals J and S. In the EGUSA program, this system is called Runtime Vent with Minimum.

Note that this system has the capacity to provide the full IRC average ventilation rate. That may be accomplished by occupants running either the central fan or the exhaust fan more. It is not possible to schedule exhaust fan operation in the EGUSA program for this system, so exhaust fan operation was assumed to be zero. In practice, the actual amount of exhaust fan operation would not be zero but would depend on the amount of exhaust fan(s) usage by the occupant(s), such that the actual ventilation airflow may be higher or lower than the IRC target rate over a specific time frame. The greater the number of occupants the greater the expected exhaust fan operation. System 4 has the advantages of fully distributing the ventilation air, improving air filtration throughout the house, and homogenizing indoor temperature and humidity comfort conditions.

6.1.6 System 5: Ventilating Dehumidifier Supply with synchronized compressor or central fan operation

The ventilating dehumidifier system involved a ducted dehumidifier intaking 110 CFM of outdoor air 50% of the time for an average hourly ventilation rate of 55 CFM. The outdoor air was delivered to the main supply duct of the central space conditioning system. The system was modeled such that: a) if the outdoor air dew-point temperature was above 50°F then the dehumidifier compressor would operate to dehumidify the ventilation air, and b) if the outdoor dew-point temperature was less than or equal to 50°F then the dehumidifier compressor would not run but the central system fan would operate (if it was not already running for cooling) at low speed coincident with the dehumidifier fan to temper the outdoor air before delivery to the central system supply duct. This system was indirectly modeled within the EGUSA program since it was not directly available.

6.1.7 System 6: Balanced with No Recovery

This system was the same as System 3 except the recovery efficiency was set to zero. This could be a system of efficient independent supply and exhaust fans.

6.1.8 System 7: Balanced Energy Recovery Ventilator with synchronized central fan operation

This balanced ERV system was modeled with an outdoor air ventilation rate of 110 CFM at 50% runtime and 80 W (1.375 CFM/W). The recovery efficiency was set to 70% which is representative of an efficient ERV. This ERV system was assumed to have a duct distribution system integrated with the central space conditioning system. The dwelling exhaust air intake was from the central system return and the outdoor air supply was to the central system main supply duct. The central space conditioning system fan was operated on low speed in sync with the ERV at 300 CFM and 120 W (2.5 CFM/W). The total fan power was modeled as 200 W.

6.2 Climates Used in the Simulations

The graphic in Figure 1 shows the US climate zones as illustrated by the United States Department of Energy, Office of Energy Efficiency & Renewable Energy. Figure 2 shows the International Energy Conservation Code (IECC) climate zone map customized with some major city locations labeled. Table 3 is listing of the cities and respective climate zones modeled in this residential ventilation study. The Washington DC climate was modeled in two locations to capture differences between the more humid area near the Chesapeake Bay and Potomac River, and the further west location in Dulles, Virginia.

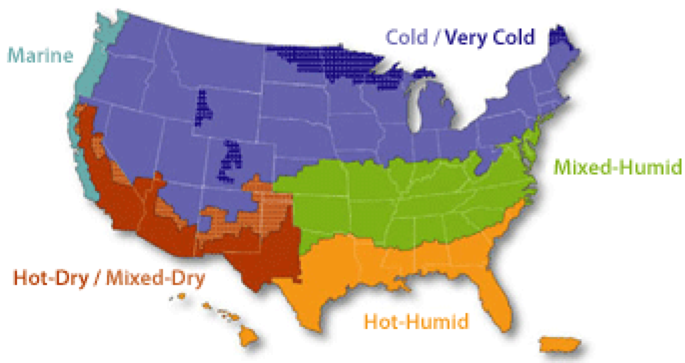

Figure 1. Climate map of the United States; Source: (USDOE)

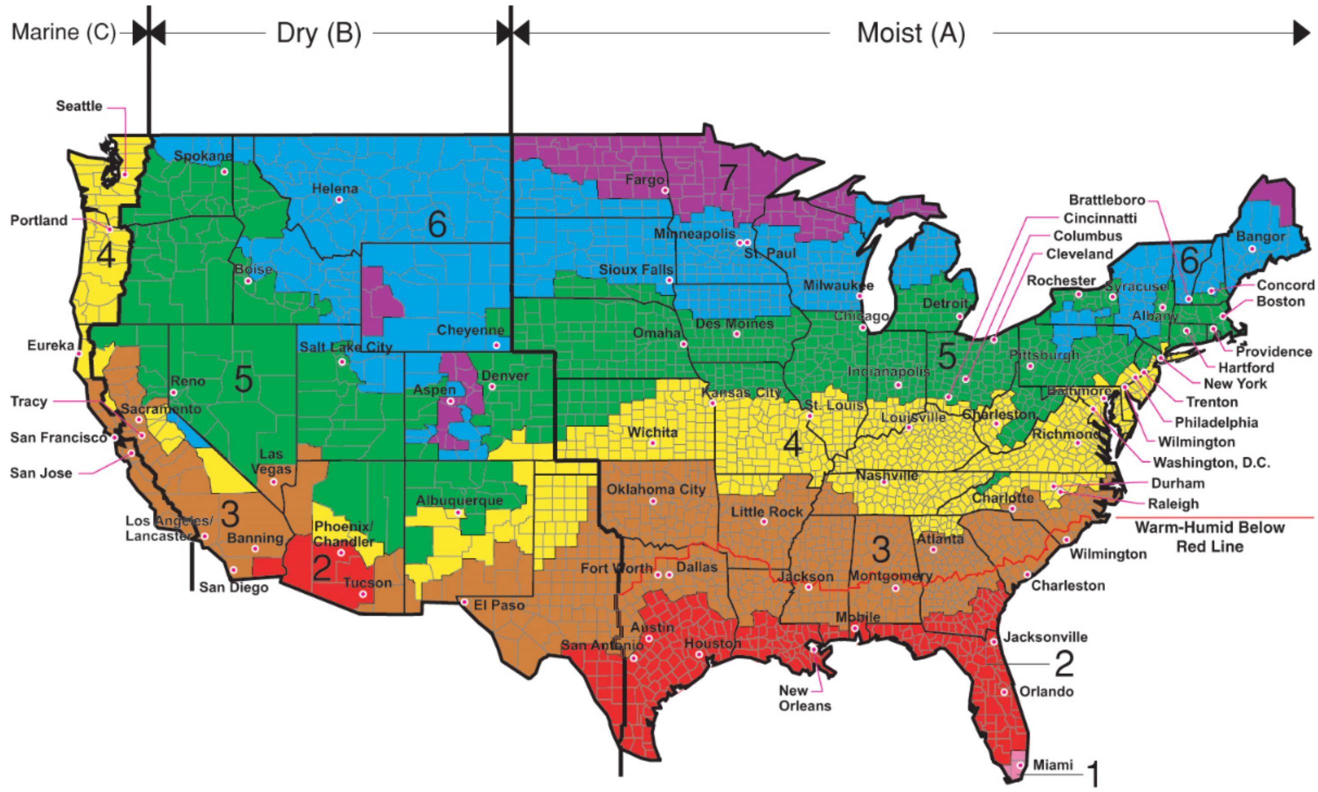

Figure 2. IECC climate zone map customized with some major city locations labeled

Table 3. Listing of US City and Climate Zone Modeled

	City	IECC Climate Zone	Description
1	Miami, Florida	1A	Hot-humid
2	Houston, Texas	2A	Hot-humid
3	Atlanta, Georgia	3A	Mixed-humid
4	Washington DC (Reagan National)	4A	Mixed-humid
5	Washington DC (Dulles-Virginia)	4A	Mixed-humid
6	San Jose/San Francisco, California	3C	Warm-marine
7	Seattle, Washington	4C	Mixed-marine
8	Denver, Colorado	5B	Cold-dry
9	Minneapolis, Minnesota	6A	Cold-moist
10	Fargo, North Dakota	7A	Very Cold-moist
11	Fairbanks, Alaska	8	Very Cold

To illustrate the large differences in outdoor moisture between climates, Figure 3 shows the monthly average outdoor dew-point temperature for each of the modeled climates. Figure 3 also shows shaded area defining upper and lower boundaries of typical indoor dew-point temperatures that should be maintained for comfort conditions. Crossing through that is a dotted line representing expected seasonal variation in indoor air dewpoint temperature.

In the climates of San Jose/San Francisco, Seattle, Fairbanks, and Denver, average monthly outdoor air dew-point temperature never exceeds the typical upper indoor comfort boundary (57 °F). That means that outdoor ventilation air will not contribute to elevated indoor moisture conditions and can always be used to reduce the level of indoor moisture. That is almost true in Minneapolis and Fargo except for the months of July and August when outdoor dew point temperature approaches 60°F.

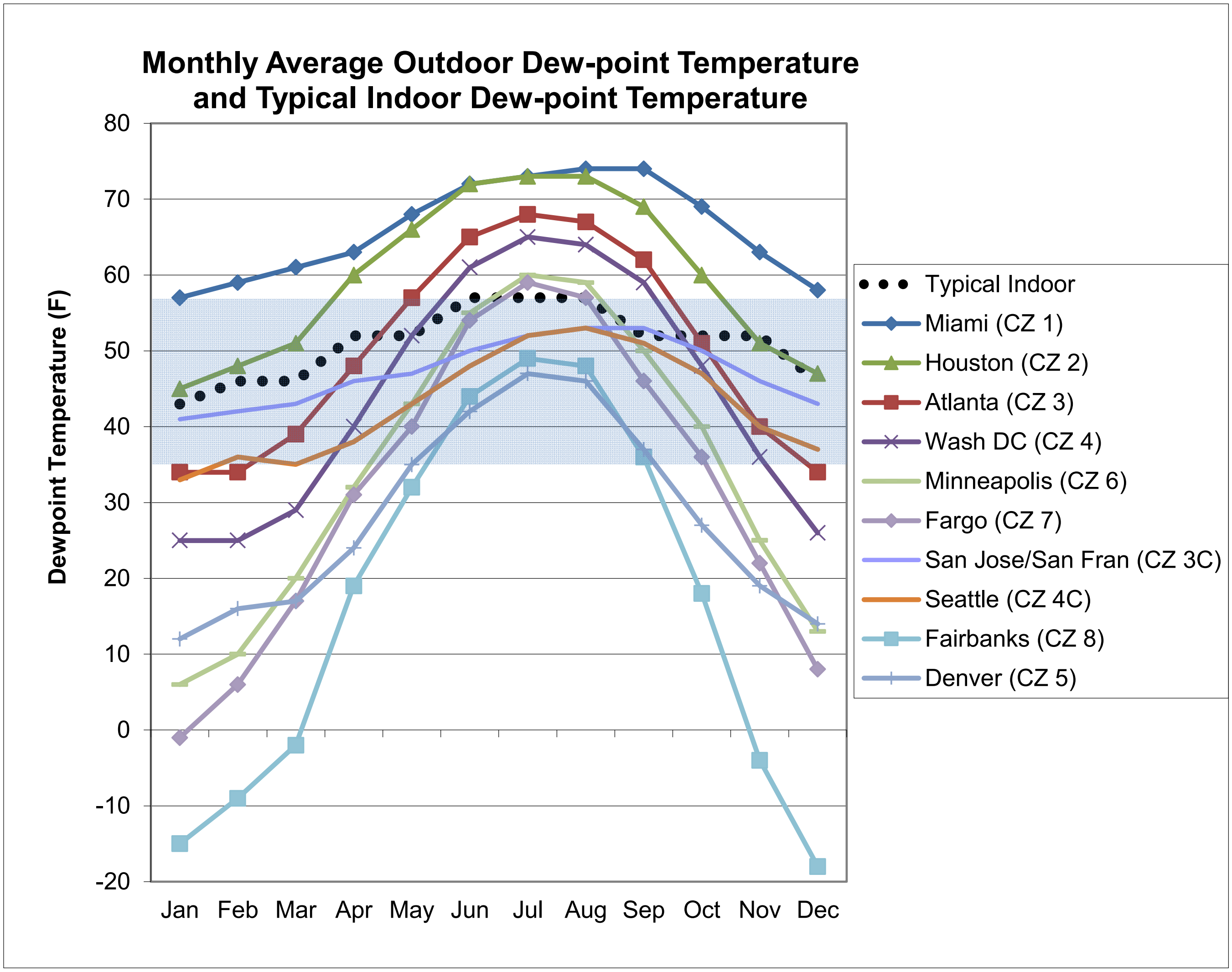

Figure 3. Monthly average outdoor air dew-point temperature for the climates modeled in this study

6.3 Parametric Cases Modeled

A total of 88 cases were modeled in this study. A description of the different parameters is given in Table 4. All parameters were held constant except for the Climate Zone/City (11 locations) and the mechanical ventilation system type (8 types, including a no-ventilation case). The source of the information used for each parameter is also provided in Table 4.

Table 4. Parametric cases modeled

Parameter	Number of modeled options	Comment
Systems	1	*Energy Star minimum values used, as of Oct-2019* Gas furnace: 90% AFUE, South CZ 1-4; 95% AFUE, North CZ 5-8 DX cooling: 15 SEER, 12.5 EER, Single-stage cooling with X13 ECM blower DX dehumidifier: 65 pint/day (31 L/day), EF=2.0 (L/kWh) Domestic water heating: gas, ≥55 gal, EF=0.77
Climates/Locations	11	*IRC R301.1* Climate Zones: 1 (Miami), 2 (Houston), 3 (Atlanta), 3C (San Jose), 4 (Washinton DC, Reagan National and Dulles), 4C (Seattle), 5 (Denver), 6 (Minneapolis), 7 (Fargo), 8 (Fairbanks)
Building foundation type	1	Slab-on-grade
Building size	1	Production builder plan: 2467 ft^2, 3 bedroom, 1-story
Building enclosure thermal efficiency level	1	*2018 IRC TABLE R402.1.2 Insulation and Fenestration Requirements by Component*
Building enclosure air leakage	1	*2018 IRC R402.4.1.2* 5 ach50 in CZ 1-2; 3 ach50 in CZ 3-6
Duct thermal distribution efficiency, total air leakage (to attic)	1	*2018 IRC TABLE R405.5.2(1) and R403.3.4* Thermal distribution efficiency = 0.88, Total leakage =< 4 % of conditioned floor area 2467 ft2*0.04 = 99 cfm
Duct insulation	1	*2018 IRC 403.3.1* R-8 supply and return (all ducts in attic)
Space conditioning thermostat set-points	1	*2018 IRC TABLE R405.5.2(1)* 72 heating, 75 cooling for annual simulation 70 heating, 75 cooling fixed by EGUSA for Manual J8 sizing
Dehumidifier humidistat set-points	1	*EGUSA default dehumidifier model (Energy Factor= 2.0 L/kWh with performance curve coefficients)* 55% (on at 57% off at 53%)
Mechanical ventilation rate	1	*2018 IRC M1505.4.3* (0.01*CFA)+(Nbr+1)*7.5 2467 ft2, 3 bdrm = 55 cfm
Mechanical ventilation systems	8	No Ventilation Vent1 = Exhaust (55 cfm continuous, 2.8 cfm/W=20 W) Vent2 = Hybrid CFI Supply (75 cfm, no minimum) w/ automatic Exhaust backup (55 cfm) Vent3 = Balanced ERV (55 cfm, 1.4 cfm/W=40 W, 70% Sensible Recovery Efficiency) Vent4 = CFI Supply 33% (75 cfm, 33% minimum) w/ occupant controlled Exhaust (55 cfm) Vent5 = Ventilating Dehumidifier Supply (110 cfm, 50% runtime, w/ compressor or central fan interlock) Vent6 = Balanced no recovery (55 cfm@1.4 cfm/W=40 W, 0% SRE) Vent7 = Balanced ERV w/ AHU interlock (ERV: 110 cfm, 80W; AHU: 400 cfm, 0.3 W/cfm, 120 W, 50% runtime) Note: ECM AHU blower at 0.3 W/cfm (3.33 cfm/W @ 0.5 in w.c.), Wilcox 2006 and other. ERV SRE=0.70 is avg and median of HVI Certified Directory
Internal heat generation	1	*2018 IRC TABLE 405.5.2(1)* Internal Gain (Btu/day) = 17,900 + 23.8 * CFA + 4,104 * Nbr 2467 ft2, 3 bdrm = 88,927 Btu/day; 3705 Btu/h
Internal moisture generation	1	*ASHRAE RP-1449 and EGUSA default* 12 lb/day; 0.5 lb/h; 500 Btu/h
Total number of simulation cases	**88**	

6.4 Climate-Based Building Simulation Inputs

The 2018 IRC was the primary resource used for establishing the climate-based simulation inputs shown in Table 5. Refer to Table 4 for a more detailed explanation of the sources of information. The electric and gas utility rates were established by the default values updated for each location in the EnergyGauge software.

Table 5. Climate-based simulation inputs

Parameter[1]	Climate Zone 1 Miami	Climate Zone 2 Houston (IAH)	Climate Zone 3 Atlanta	Climate Zone 3C San Jose	Climate Zone 4 Wash DC (Dulles)	Climate Zone 4C Seattle (Renton)	Climate Zone 5 Denver (Broomfield)	Climate Zone 6 Minneapolis	Climate Zone 7 Fargo	Climate Zone 8 Fairbanks
Wall insulation R-value (nominal)	13	13	20	20	20	20	20	20+5	20+5	20+5
cavity	13	13	20	20	20	20	20	20	20	20
sheathing	0.45	0.45	0.45	0.45	0.45	0.45	0.45	5	5	5
framing factor	0.23	0.23	0.17	0.17	0.17	0.17	0.17	0.17	0.17	0.17
Ceiling insulation R-value	30	38	38	38	49	49	49	49	49	49
Slab insulation R-value (2' down)	0	0	0	0	10, 2 ft	10, 2 ft	10, 2 ft	10, 4 ft	10, 4 ft	10, 4 ft
Window U-value	0.35 (NR)	0.35 (0.40)	0.32	0.32	0.32	0.30	0.30	0.30	0.30	0.30
Window SHGC	0.25	0.25	0.25	0.25	0.30 (0.40)	0.30 (NR)	0.30 (NR)	0.30 (NR)	0.30 (NR)	0.30 (NR)
Building enclosure air leakage (ach50)	5	5	3	3	3	3	3	3	3	3
Duct thermal distribution efficiency, or % Conditioned Floor Area	0.88, 4%	0.88, 4%	0.88, 4%	0.88, 4%	0.88, 4%	0.88, 4%	0.88, 4%	0.88, 4%	0.88, 4%	0.88, 4%
Air distribution system location	attic	attic	attic	attic	attic	attic	attic	attic	attic	attic
SEER, EER	15, 12.5	15, 12.5	15, 12.5	15, 12.5	15, 12.5	15, 12.5	15, 12.5	15, 12.5	15, 12.5	15, 12.5
AFUE	90%	90%	90%	90%	90%	90%	95%	95%	95%	95%
Dehumidifier										
Efficacy (L/kWh)	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0
Indoor set-point (%), On-Off	55, 57-53	55, 57-54	55, 57-55	55, 57-56	55, 57-57	55, 57-58	55, 57-59	55, 57-60	55, 57-61	55, 57-62
Internal heat gain (lumped) (people+lighting+appliances/equip)	88,927 Btu/day; 3705 Btu/h	88,927 Btu/day; 3705 Btu/h	88,927 Btu/day; 3705 Btu/h	88,927 Btu/day; 3705 Btu/h	88,927 Btu/day; 3705 Btu/h	88,927 Btu/day; 3705 Btu/h	88,927 Btu/day; 3705 Btu/h	88,927 Btu/day; 3705 Btu/h	88,927 Btu/day; 3705 Btu/h	88,927 Btu/day; 3705 Btu/h
Internal moisture generation	12 lb/day; 0.5 lb/h	12 lb/day; 0.5 lb/h	12 lb/day; 0.5 lb/h	12 lb/day; 0.5 lb/h	12 lb/day; 0.5 lb/h	12 lb/day; 0.5 lb/h	12 lb/day; 0.5 lb/h	12 lb/day; 0.5 lb/h	12 lb/day; 0.5 lb/h	12 lb/day; 0.5 lb/h
DHW (EF)	0.77	0.77	0.77	0.77	0.77	0.77	0.77	0.77	0.77	0.77
Utility Rate - Electric ($/kWh)[2]	0.1161	0.1101	0.119	0.1831	0.1155	0.0966	0.1217	0.1304	0.1029	0.2127
Utility Rate - Natural Gas ($/Therm)[2]	2.042	1.314	1.634	1.206	1.191	1.025	0.78	0.818	0.737	1.015
Utility Rate - Electric ($/MMBtu)	34.0	32.3	34.9	53.6	33.8	28.3	35.7	38.2	30.1	62.3
Utility Rate - Natural Gas ($/MMBtu)	20.4	13.1	16.3	12.1	11.9	10.3	7.8	8.2	7.4	10.2

Note: Values in parenthesis for window U-value and SHGC are the 2018 IRC code values. However, due to typical glass availabilty in the industry, some unrealistic combininations have been substitued for realistic values.

[1] See Parametric Cases table for explanation of sources for values.

[2] Utility rates were accepted as the EnergyGuage USA program location defaults.

6.5 Annual Simulation Results

The overall tabular results from the mechanical ventilation simulation study are shown in Table 6, Table 7, and Table 8. The tables break out the data into manageable groups of climate zones. Data descriptions from the left column to the right are:

1. The climate zone, representative city in that climate zone, and the dwelling unit mechanical ventilation system label.
2. The Energy Rating Index (ERI), which is a developed rating codified in the IECC that mostly follows the RESNET Home Energy Rating System (HERS) index. The lower the ERI for a rated building, the better the energy performance for that rated building compared to the reference building. The reference building is geometrically a twin but otherwise it is based on the 2006 IECC requirements for that climate zone. If the ERI were 100, the building energy performance would be equivalent to the reference building. If the ERI were zero, the building would produce as much energy as it consumes (typically via solar energy).
3. Heating and cooling system capacity, sensible heat ratio, and airflow as sized based on the Air Conditioning Contractors of America (ACCA) Manuals J and S.
4. Cooling energy consumption and mechanical ventilation system energy consumption during the cooling season. The electrical energy consumption is multiplied by the applicable electric utility rate to arrive at the cost in dollars.
5. Supplemental dehumidification energy consumption and cost. This represents total moisture removed in addition to moisture removed by the cooling system when it was active. In each case, the dehumidifier was controlled to limit indoor RH to 55% with a +/- 2% RH control deadband.
6. Heating energy consumption and mechanical ventilation system energy consumption during the heating season. The gas consumption and electrical energy consumption were multiplied by the applicable gas and electric utility rates to arrive at the cost in dollars.
7. Domestic water heating energy consumption and cost which was the same for all ventilation system cases within each climate zone.
8. Electric appliances and lighting energy consumption and cost which was the same for all ventilation system cases within each climate zone.
9. House totals for electric and gas energy consumption and the assembled total cost in dollars.
10. The house total difference from the Vent1-Exhaust reference case, expressed in dollars and percentage.

The plots in Figure 4 through Figure 6 illustrate the tabular data for faster visualization of the overall results between climates and mechanical ventilation systems. These plots show total cost (Figure 4), difference in total cost (Figure 5) from the Vent1-Exhaust case, and percent difference in total cost (Figure 6).

Table 9 through Table 14 reduce the overall tabular data to facilitate faster comparison of the data between climate zones for each mechanical ventilation system. Note that green shading represents the lowest value (not including the No-Mechanical Vent and Vent4 cases) and orange shading represents the highest value. The tables include:

- Total building energy cost by climate zone and ventilation system
- Difference in total building energy cost from the Vent1-Exhaust case, by climate zone and ventilation system
- Percent difference in total building energy cost from the Vent1-Exhaust case, by climate zone and ventilation system

- Total cooling season and heating season cost by climate zone and ventilation system
- Supplemental dehumidification cost by climate zone and ventilation system

Table 6. Annual simulation tabular results (Miami thru San Jose)

Location and Ventilation System	ERI 2015	Manual J+S: Heating Capacity (kBtu/h)	Manual J+S: Cooling Capacity (kBtu/h)	Manual J+S: Sensible Heat Ratio	Manual J+S: Airflow (CFM)	Electric, Cooling Season: Outdoor Unit (kWh)	Electric, Cooling Season: Indoor Unit (kwh)	Electric, Cooling Season: Vent Fan (kWh)	Electric, Cooling Season: Sub Total (kWh)	Electric, Cooling Season: Sub Total ($)	Dehum. (kWh)	Dehum. ($)	Gas, Heating Season: (Therm)	Gas, Heating Season: Furnace Fan (kWh)	Gas, Heating Season: Vent. Fan (kWh)	Gas, Heating Season: Sub Total ($)	Gas, DHW (Therm)	Gas, DHW ($)	Elec. Appl. (kWh)	Elec. Appl. ($)	House Totals (kWh)	House Totals (Therm)	House Totals ($)	Difference ($)	Difference (%)
Miami (CZ 1)																									
No Vent	60	17.27	28.05	0.75	840	4730	972	0	5702	662	10	1	4.4	2	0	9	89.0	182	5508	640	11222	93.4	1484	-$ 88	-5.6%
Vent1-Exhaust	58	17.70	29.31	0.75	870	5097	1035	171	6303	732	55	6	6.0	3	4	13	89.0	182	5508	640	11873	95.0	1572	--	--
Vent2-Hybrid CFIS-Exh	58	17.70	29.31	0.75	870	5097	1035	110	6242	725	55	6	6.0	3	3	13	89.0	182	5508	640	11811	95.0	1565	-$ 7	-0.4%
Vent3-Balanced ERV	57	18.00	30.59	0.75	930	4998	1018	343	6359	738	27	3	5.3	3	7	12	89.0	182	5508	640	11904	94.3	1575	$ 3	0.2%
Vent4-CFIS-33%-Exh*	60	17.33	29.26	0.75	870	4981	1018	275	6274	728	11	1	4.6	12	7	12	89.0	182	5508	640	11812	93.6	1563	-$ 9	-0.6%
Vent5-vDehumidifier	68	17.95	31.10	0.75	930	5576	1145	2028	8749	1016	0	0	5.1	3	52	17	89.0	182	5508	640	14312	94.1	1854	$ 282	17.9%
Vent6-Bal no recovery	60	18.54	31.10	0.75	930	5477	1105	340	6922	804	176	20	8.9	5	10	20	89.0	182	5508	640	12621	97.9	1665	$ 93	5.9%
Vent7-Bal ERV interlock	59	17.65	30.38	0.75	900	5061	1031	857	6949	807	25	3	5.1	3	19	13	89.0	182	5508	640	12504	94.1	1644	$ 72	4.6%
Houston (CZ 2)																									
No Vent	58	32.65	28.45	0.75	840	2916	590	0	3506	386	119	13	151.4	83	0	208	107.0	141	5508	606	9216	258.4	1354	-$ 90	-6.2%
Vent1-Exh	55	33.60	29.73	0.75	900	3193	639	119	3951	435	230	25	167.8	93	56	237	107.0	141	5508	606	9838	274.8	1444	--	--
Vent2-Hyb Sup-Exh	55	33.51	29.73	0.75	900	3193	639	83	3915	431	230	25	167.8	93	46	236	107.0	141	5508	606	9792	274.8	1439	-$ 5	-0.3%
Vent3-Bal ERV	54	33.37	30.49	0.75	900	3109	624	238	3971	437	164	18	162.0	91	112	235	107.0	141	5508	606	9846	269.0	1438	-$ 6	-0.4%
Vent4-CFIS-33%-Exh*	58	32.88	29.65	0.75	900	3098	624	260	3982	438	127	14	152.1	140	168	234	107.0	141	5508	606	9925	259.1	1433	-$ 11	-0.8%
Vent5-vDeh	62	34.09	28.85	0.75	870	3379	681	1103	5163	568	0	0	156.7	96	574	280	107.0	141	5508	606	11341	263.7	1595	$ 151	10.5%
Vent6-Bal no recovery	58	35.25	31.61	0.75	960	3464	689	236	4389	483	387	43	190.7	108	114	275	107.0	141	5508	606	10506	297.7	1548	$ 104	7.2%
Vent7-Bal ERV interlock	56	33.43	30.68	0.75	930	3148	632	595	4375	482	157	17	159.4	91	281	250	107.0	141	5508	606	10412	266.4	1496	$ 52	3.6%
Atlanta (CZ 3)																									
No Vent	57	34.60	24.13	0.75	720	1617	333	0	1950	232	72	9	289.1	160	0	491	122.3	200	5508	656	7690	411.4	1587	-$ 116	-6.8%
Vent1-Exh	53	35.92	25.29	0.75	750	1777	360	88	2225	265	118	14	328.4	184	87	569	122.3	200	5508	656	8122	450.7	1703	--	--
Vent2-Hyb Sup-Exh	53	35.92	25.29	0.75	750	1777	360	66	2203	262	118	14	328.4	184	69	567	122.3	200	5508	656	8082	450.7	1698	-$ 5	-0.3%
Vent3-Bal ERV	53	35.48	25.62	0.75	780	1714	349	178	2241	267	86	10	308.7	174	172	546	122.3	200	5508	656	8181	431.0	1678	-$ 25	-1.5%
Vent4-CFIS-33%-Exh*	56	36.02	25.73	0.75	780	1741	362	206	2309	275	72	9	293.8	198	233	531	122.3	200	5508	656	8320	416.1	1670	-$ 33	-1.9%
Vent5-vDeh	57	34.95	24.93	0.75	750	1893	386	622	2901	345	0	0	311.6	186	652	609	122.3	200	5508	656	9247	433.9	1809	$ 106	6.2%
Vent6-Bal no recovery	56	37.53	26.35	0.75	780	1869	377	173	2419	288	161	19	360.1	205	177	634	122.3	200	5508	656	8470	482.4	1796	$ 93	5.5%
Vent7-Bal ERV interlock	54	35.48	25.62	0.75	780	1743	356	444	2543	303	84	10	304.3	174	432	569	122.3	200	5508	656	8741	426.6	1737	$ 34	2.0%
San Jose (CZ 3C)																									
No Vent	59	250.40	19.53	0.75	600	599	129	0	728	133	121	22	242.8	130	0	317	129.0	156	5508	1009	6487	371.8	1636	-$ 77	-4.5%
Vent1-Exh	57	26.00	19.85	0.75	600	555	119	68	742	136	16	3	299.0	162	107	410	129.0	156	5508	1009	6535	428.0	1713	--	--
Vent2-Hyb Sup-Exh	57	26.00	19.85	0.75	600	555	119	60	734	134	16	3	299.0	162	85	406	129.0	156	5508	1009	6505	428.0	1707	-$ 6	-0.4%
Vent3-Bal ERV	56	25.67	19.73	0.75	600	587	126	144	857	157	45	8	269.3	147	207	390	129.0	156	5508	1009	6764	398.3	1719	$ 6	0.4%
Vent4-CFIS-33%-Exh*	59	25.91	20.02	0.75	600	604	130	184	918	168	70	13	254.5	177	203	377	129.0	156	5508	1009	6876	383.5	1721	$ 8	0.5%
Vent5-vDeh																									
Vent6-Bal no recovery	60	27.15	20.20	0.75	600	545	116	126	787	144	9	2	328.0	182	224	470	129.0	156	5508	1009	6710	457.0	1780	$ 67	3.9%
Vent7-Bal ERV interlock	58	25.67	19.73	0.75	600	604	130	355	1089	199	44	8	263.7	147	521	440	129.0	156	5508	1009	7309	392.7	1812	$ 99	5.8%

Table 7. Annual simulation tabular results (Washington DC thru Denver)

		Manual J+S				Electric, Cooling Season					Dehum.		Gas, Heating Season				Gas, DHW		Elec. Appl.		House Totals			Difference	
Location and Ventilation System	ERI 2015	Heating Capacity (kBtu/h)	Cooling Capacity (kBtu/h)	Sensible Heat Ratio	Airflow (CFM)	Outdoor Unit (kWh)	Indoor Unit (kwh)	Vent Fan (kWh)	Sub Total (kWh)	Sub Total ($)	(kWh)	($)	(Therm)	Furnace Fan (kWh)	Vent. Fan (kWh)	Sub Total ($)	(Therm)	($)	(kWh)	($)	(kWh)	(Therm)	($)	($)	(%)
Wash DC-Reagan (CZ 4)																									
No Vent	63	29.49	23.97	0.75	720	1467	303	0	1770	204	53	6	385.3	207	0	483	133.0	158	5508	637	7538	518.3	1488	-$ 107	-6.7%
Vent1-Exh	58	30.94	24.55	0.75	750	1588	323	80	1991	230	89	10	438.3	238	96	561	133.0	158	5508	637	7922	571.3	1595	--	--
Vent2-Hyb Sup-Exh	58	30.94	24.55	0.75	750	1586	323	58	1967	227	87	10	438.4	238	68	557	133.0	158	5508	637	7868	571.4	1589	-$ 6	-0.4%
Vent3-Bal ERV	57	30.52	24.34	0.75	720	1540	316	162	2018	233	55	6	412.6	226	189	539	133.0	158	5508	637	7996	545.6	1573	-$ 22	-1.4%
Vent4-CFIS-33%-Exh*	60	31.09	24.80	0.75	750	1566	321	169	2056	237	53	6	394.0	268	172	520	133.0	158	5508	637	8057	527.0	1558	-$ 37	-2.3%
Vent5-vDeh	61	29.84	24.36	0.75	720	1696	347	562	2605	301	0	0	416.1	240	712	606	133.0	158	5508	637	9065	549.1	1701	$ 106	6.6%
Vent6-Bal no recovery	62	32.94	25.18	0.75	750	1662	337	155	2154	249	116	13	483.9	268	195	630	133.0	158	5508	637	8241	616.9	1687	$ 92	5.8%
Vent7-Bal ERV interlock	59	30.52	24.34	0.75	720	1566	321	402	2289	264	55	6	406.8	226	474	565	133.0	158	5508	637	8552	539.8	1631	$ 36	2.3%
Wash DC-Dulles (CZ 4)																									
No Vent	63	31.55	25.19	0.75	750	1136	234	0	1370	158	12	1	440.8	238	0	552	138.8	165	5508	637	7128	579.6	1514	-$ 114	-7.0%
Vent1-Exh	57	33.20	26.66	0.75	810	1233	250	73	1556	180	68	8	500.0	273	102	639	138.8	165	5508	637	7507	638.8	1628	--	--
Vent2-Hyb Sup-Exh	57	33.20	26.66	0.75	810	1233	250	59	1542	178	68	8	500.0	273	73	635	138.8	165	5508	637	7464	638.8	1623	-$ 5	-0.3%
Vent3-Bal ERV	56	32.67	27.23	0.75	810	1192	243	149	1584	183	29	3	471.0	258	202	614	138.8	165	5508	637	7581	609.8	1602	-$ 26	-1.6%
Vent4-CFIS-33%-Exh*	60	33.29	27.19	0.75	810	1217	250	203	1670	193	21	2	450.2	320	200	596	138.8	165	5508	637	7719	589.0	1593	-$ 35	-2.1%
Vent5-vDeh																									
Vent6-Bal no recovery	61	35.27	28.03	0.75	840	1295	262	144	1701	196	118	14	550.5	306	206	715	138.8	165	5508	637	7839	689.3	1726	$ 98	6.0%
Vent7-Bal ERV interlock	58	32.67	27.23	0.75	810	1214	248	370	1832	212	25	3	464.9	258	505	642	138.8	165	5508	637	8128	603.7	1658	$ 30	1.8%
Seattle (CZ 4C)																									
No Vent	68	21.89	20.16	0.75	600	462	100	0	562	54	70	7	410.9	213	0	442	141.9	145	5508	531	6353	552.8	1180	-$ 83	-6.6%
Vent1-Exh	64	23.01	20.48	0.75	600	421	90	49	560	54	29	3	480.1	253	126	529	141.9	145	5508	531	6476	622.0	1263	--	--
Vent2-Hyb Sup-Exh	64	23.01	20.48	0.75	600	421	90	43	554	54	29	3	480.1	253	87	525	141.9	145	5508	531	6431	622.0	1259	-$ 4	-0.3%
Vent3-Bal ERV	63	22.65	20.36	0.75	600	448	96	106	650	63	35	3	443.2	235	244	501	141.9	145	5508	531	6672	585.1	1244	-$ 19	-1.5%
Vent4-CFIS-33%-Exh*	65	23.09	20.60	0.75	630	464	100	142	706	68	43	4	425.9	309	156	481	141.9	145	5508	531	6722	567.8	1231	-$ 32	-2.5%
Vent5-vDeh																									
Vent6-Bal no recovery	68	24.43	20.82	0.75	630	410	87	91	588	57	24	2	524.9	283	259	590	141.9	145	5508	531	6662	666.8	1327	$ 64	5.1%
Vent7-Bal ERV interlock	64	22.65	20.36	0.75	600	459	99	262	820	79	35	3	435.3	235	614	528	141.9	145	5508	531	7212	577.2	1288	$ 25	2.0%
Denver (CZ 5)																									
No Vent	63	35.80	22.10	0.75	660	775	169	0	944	115	0	0	481.0	276	0	409	148.6	116	5508	671	6728	629.6	1310	-$ 75	-5.4%
Vent1-Exh	60	37.22	22.50	0.75	660	751	163	59	973	118	0	0	548.5	318	116	481	148.6	116	5508	671	6915	697.1	1385	--	--
Vent2-Hyb Sup-Exh	60	37.22	22.50	0.75	660	751	163	49	963	117	0	0	548.5	318	86	477	148.6	116	5508	671	6875	697.1	1380	-$ 5	-0.4%
Vent3-Bal ERV	58	36.88	22.37	0.75	660	771	168	121	1060	129	0	0	516.2	301	230	467	148.6	116	5508	671	7099	664.8	1382	-$ 3	-0.2%
Vent4-CFIS-33%-Exh*	62	36.57	22.63	0.75	690	787	197	181	1165	142	0	0	492.8	375	269	463	148.6	116	5508	671	7317	641.4	1391	$ 6	0.4%
Vent5-vDeh																									
Vent6-Bal no recovery	64	39.40	23.00	0.75	690	752	163	113	1028	125	0	0	608.6	360	237	547	148.6	116	5508	671	7133	757.2	1459	$ 74	5.3%
Vent7-Bal ERV interlock	60	36.88	22.37	0.75	660	788	172	300	1260	153	0	0	509.9	301	576	504	148.6	116	5508	671	7645	658.5	1444	$ 59	4.3%

Table 8. Annual simulation tabular results (Minneapolis thru Fairbanks)

		Manual J+S				Electric, Cooling Season					Dehum.		Gas, Heating Season				Gas, DHW		Elec. Appl.		House Totals			Difference	
Location and Ventilation System	ERI 2015	Heating Capacity (kBtu/h)	Cooling Capacity (kBtu/h)	Sensible Heat Ratio	Airflow (CFM)	Outdoor Unit (kWh)	Indoor Unit (kwh)	Vent Fan (kWh)	Sub Total (kWh)	Sub Total ($)	(kWh)	($)	(Therm)	Furnace Fan (kWh)	Vent. Fan (kWh)	Sub Total ($)	(Therm)	($)	(kWh)	($)	(kWh)	(Therm)	($)	($)	(%)
Minneapolis (CZ 6)																									
No Vent	61	42.45	22.61	0.75	690	824	172	0	996	130	6	1	640.6	372	0	573	159.6	131	5508	718	6882	800.2	1552	-$ 95	-5.8%
Vent1-Exh	56	44.30	23.38	0.75	690	846	175	62	1083	141	19	2	715.0	419	113	654	159.6	131	5508	718	7142	874.6	1647	--	--
Vent2-Hyb Sup-Exh	56	44.30	23.38	0.75	690	846	175	50	1071	140	19	2	715.0	419	79	650	159.6	131	5508	718	7096	874.6	1641	-$ 6	-0.4%
Vent3-Bal ERV	55	43.95	23.73	0.75	720	845	175	127	1147	150	8	1	683.3	402	224	641	159.6	131	5508	718	7289	842.9	1640	-$ 7	-0.4%
Vent4-CFIS-33%-Exh*	59	43.01	23.47	0.75	690	858	206	180	1244	162	7	1	654.2	398	241	618	159.6	131	5508	718	7398	813.8	1630	-$ 17	-1.0%
Vent5-vDeh																									
Vent6-Bal no recovery	60	47.45	24.39	0.75	720	870	178	120	1168	152	34	4	794.5	474	231	742	159.6	131	5508	718	7415	954.1	1747	$ 100	6.1%
Vent7-Bal ERV interlock	56	43.95	23.73	0.75	720	863	179	315	1357	177	8	1	676.6	402	561	679	159.6	131	5508	718	7836	836.2	1706	$ 59	3.6%
Fargo (CZ 7)																									
No Vent	61	46.16	21.72	0.75	660	613	128	0	741	76	13	1	796.4	462	0	634	169.7	125	5508	567	6724	966.1	1404	-$ 86	-5.8%
Vent1-Exh	56	46.16	22.31	0.75	660	611	126	56	793	82	33	3	879.0	514	119	713	169.7	125	5508	567	6967	1048.7	1490	--	--
Vent2-Hyb Sup-Exh	56	47.92	22.31	0.75	660	611	126	47	784	81	33	3	879.0	514	81	709	169.7	125	5508	567	6920	1048.7	1485	-$ 5	-0.3%
Vent3-Bal ERV	55	47.79	22.59	0.75	690	621	129	114	864	89	17	2	847.0	497	236	700	169.7	125	5508	567	7122	1016.7	1482	-$ 8	-0.5%
Vent4-CFIS-33%+D.Exh*	58	46.54	22.30	0.75	660	635	166	200	1001	103	15	2	814.8	490	260	678	169.7	125	5508	567	7274	984.5	1474	-$ 16	-1.1%
Vent5-vDeh																									
Vent6-Bal no recovery	60	51.57	23.17	0.75	690	623	128	107	858	88	51	5	976.0	580	244	804	169.7	125	5508	567	7241	1145.7	1589	$ 99	6.6%
Vent7-Bal ERV interlock	56	47.79	22.59	0.75	690	635	132	284	1051	108	17	2	839.9	497	592	731	169.7	125	5508	567	7665	1009.6	1533	$ 43	2.9%
Fairbanks (CZ 8)																									
No Vent	62	51.61	17.72	0.75	540	146	31	0	177	38	22	5	1234.0	714	0	1404	197.8	201	5508	1172	6421	1431.8	2819	-$ 169	-5.7%
Vent1-Exh	57	53.64	17.81	0.75	540	121	26	23	170	36	9	2	1357.0	791	152	1578	197.8	201	5508	1172	6630	1410.0	2988	--	--
Vent2-Hyb Sup-Exh	57	53.64	17.81	0.75	540	121	26	21	168	36	9	2	1357.0	791	100	1567	197.8	201	5508	1172	6576	1554.8	2977	-$ 11	-0.4%
Vent3-Bal ERV	56	53.45	17.79	0.75	540	136	29	50	215	46	11	2	1306.8	764	300	1553	197.8	201	5508	1172	6798	1504.6	2973	-$ 15	-0.5%
Vent4-CFIS-33%+D.Exh*	59	51.77	17.81	0.75	540	143	58	108	309	66	10	2	1268.8	750	343	1520	197.8	201	5508	1172	6920	1466.6	2960	-$ 28	-0.9%
Vent5-vDeh																									
Vent6-Bal no recovery	61	57.76	17.96	0.75	540	114	24	40	178	38	11	2	1488.7	881	311	1765	197.8	201	5508	1172	6889	1686.5	3177	$ 189	6.3%
Vent7-Bal ERV interlock	56	53.45	17.79	0.75	540	141	30	124	295	63	11	2	1297.9	764	752	1640	197.8	201	5508	1172	7330	1495.7	3077	$ 89	3.0%

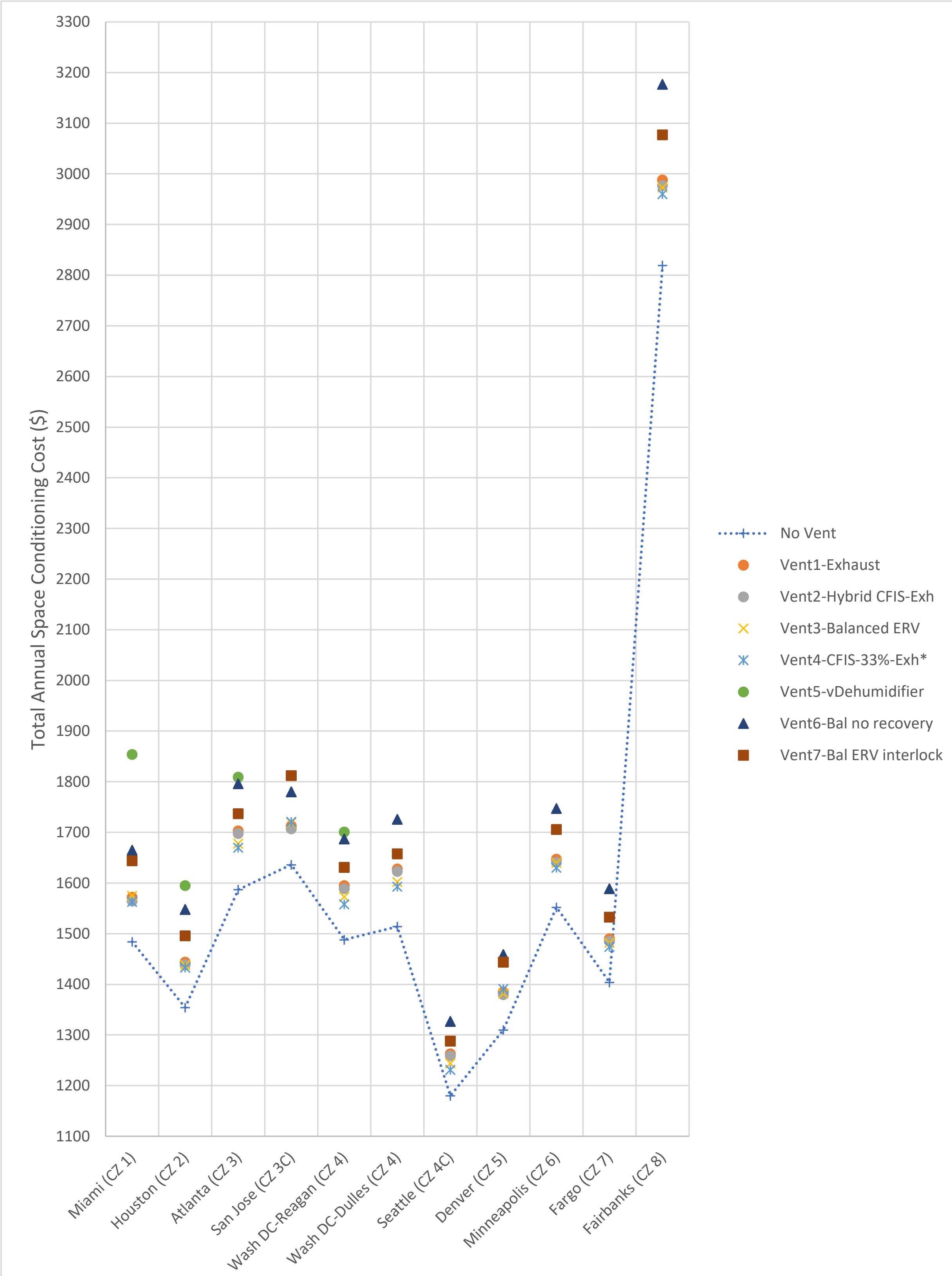

Figure 4. Total annual space conditioning cost by ventilation system type and climate

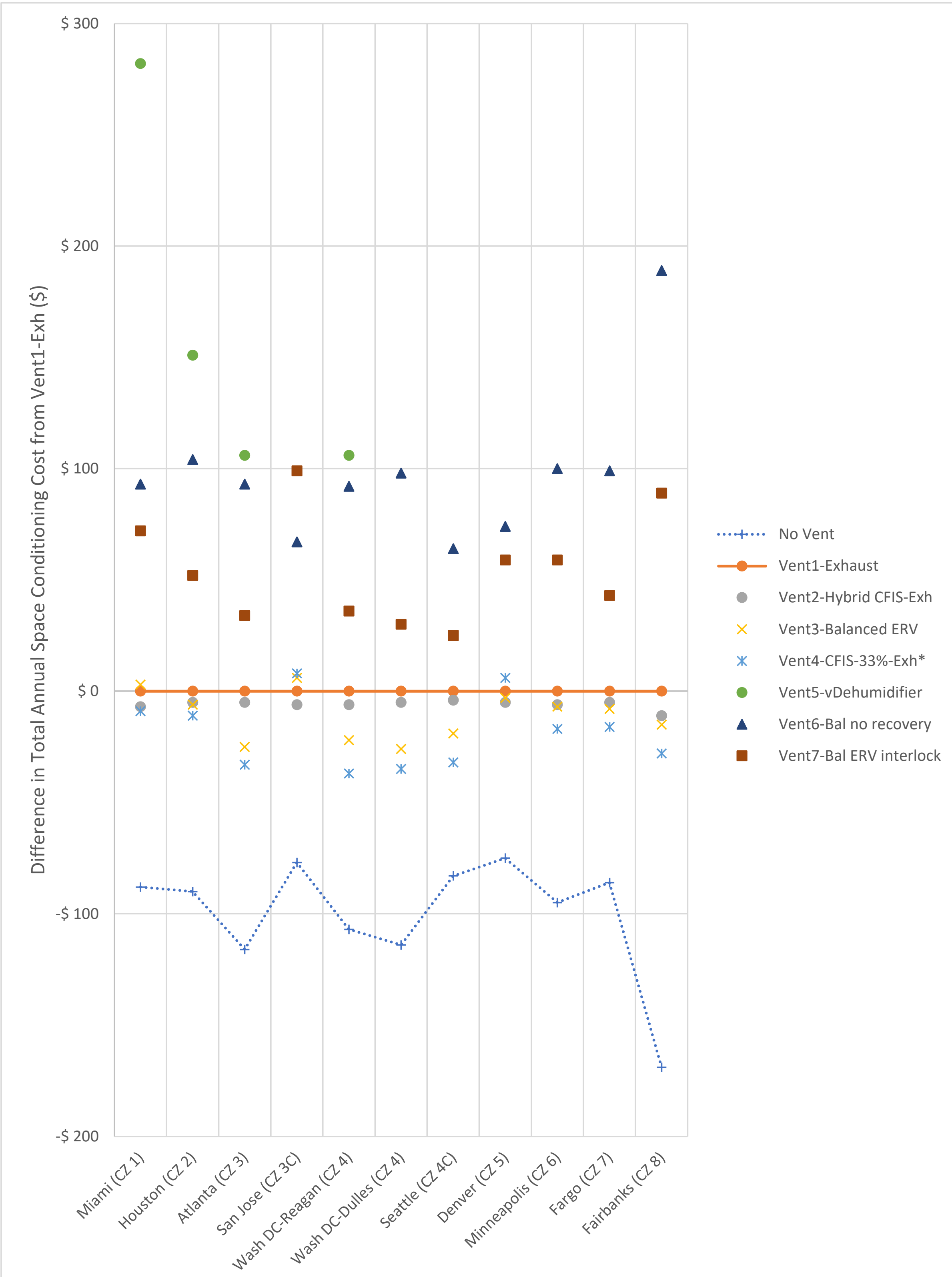

Figure 5. Difference in total annual space conditioning cost with Vent1-Exhaust as reference for each ventilation system type and climate

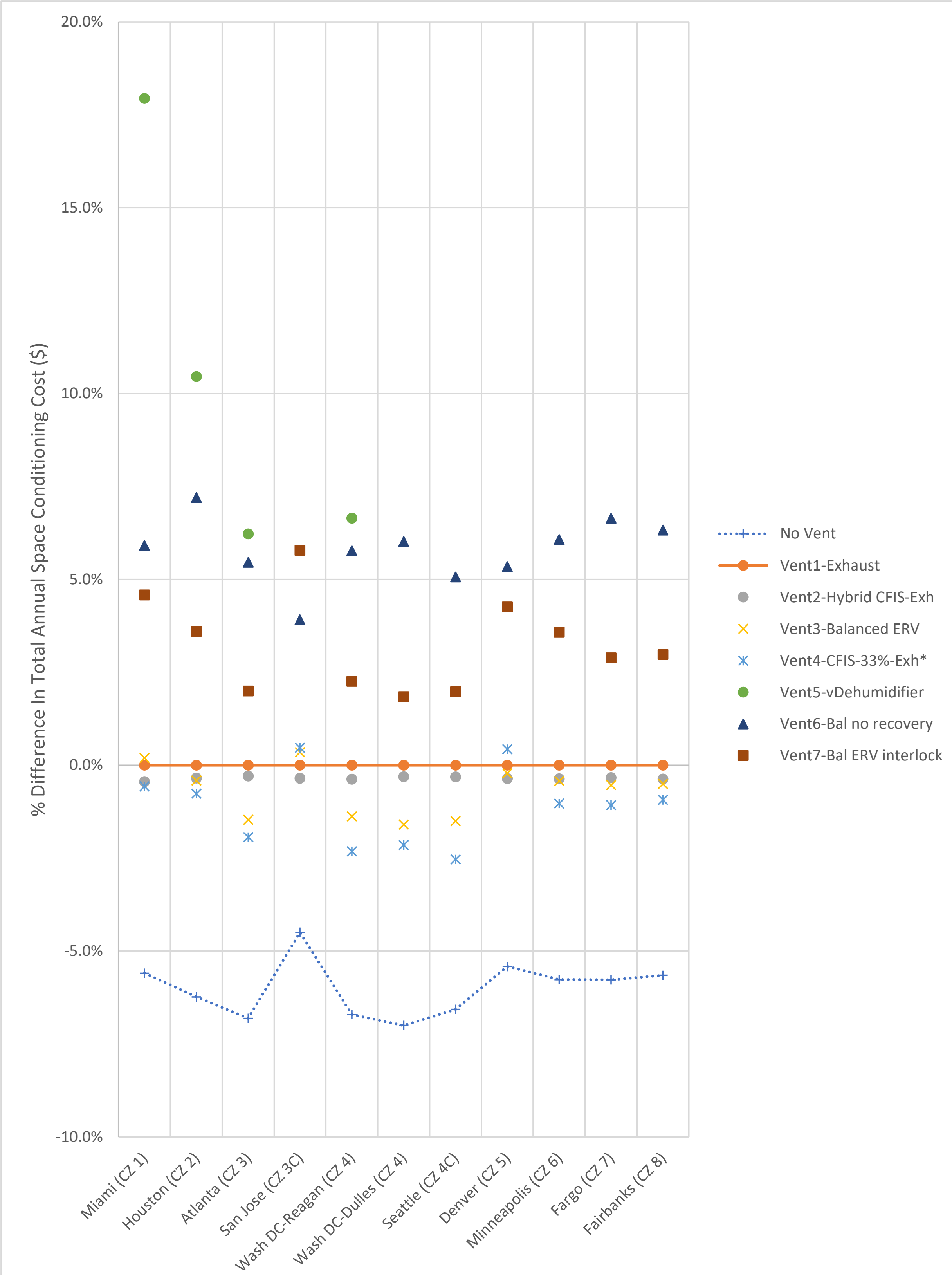

Figure 6. Percent difference in total annual space conditioning cost for each ventilation system type and climate

Table 9. Total annual building energy cost by climate zone and ventilation system

		Total Building Energy Cost							
		No Vent	Vent1 Exhaust	Vent2 Hybrid CFIS+Exhaust	Vent3 ERV 70% Recovery	Vent4 CFIS-33%	Vent5 Ventilating Dehumidifier	Vent6 Balanced noRecovery	Vent7 ERV, AHU interlock
		($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)
CZ 1A	Miami	1484	1572	1565	1575	1563	1854	1665	1644
CZ 2A	Houston	1354	1444	1439	1438	1433	1595	1548	1496
CZ 3A	Atlanta	1587	1703	1698	1678	1670	1809	1796	1737
CZ 3C	San Jose	1636	1713	1707	1719	1721	--	1780	1812
CZ 4A	Wash DC, Reagan	1488	1595	1589	1573	1558	1701	1687	1631
CZ 4A	Wash DC, Dulles	1514	1628	1623	1602	1593	--	1726	1658
CZ 4C	Seattle	1180	1263	1259	1244	1231	--	1327	1288
CZ 5B	Denver	1310	1385	1380	1382	1391	--	1459	1444
CZ 6A	Minneapolis	1552	1647	1641	1640	1630	--	1747	1706
CZ 7A	Fargo	1404	1490	1485	1482	1474	--	1589	1533
CZ 8	Fairbanks	2819	2988	2977	2973	2960	--	3177	3077
	Min:	1180	1263	1259	1244	1231	1595	1327	1288
	Max:	2819	2988	2977	2973	2960	1854	3177	3077
	Avg:	1575	1675	1669	1664	1657	1740	1773	1730

Table 10. Difference in total annual building energy cost from the Vent1-Exhaust case, by climate zone and ventilation system

		Difference in Total Building Energy Cost from Vent1-Exhaust							
		No Vent	Vent1 Exhaust	Vent2 Hybrid CFIS+Exhaust	Vent3 ERV 70% Recovery	Vent4 CFIS-33%	Vent5 Ventilating Dehumidifier	Vent6 Balanced noRecovery	Vent7 ERV, AHU interlock
		($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)
CZ 1A	Miami	-88	--	-7	3	-9	282	93	72
CZ 2A	Houston	-90	--	-5	-6	-11	151	104	52
CZ 3A	Atlanta	-116	--	-5	-25	-33	106	93	34
CZ 3C	San Jose	-77	--	-6	6	8	--	67	99
CZ 4A	Wash DC, Reagan	-107	--	-6	-22	-37	106	92	36
CZ 4A	Wash DC, Dulles	-114	--	-5	-26	-35	--	98	30
CZ 4C	Seattle	-83	--	-4	-19	-32	--	64	25
CZ 5B	Denver	-75	--	-5	-3	6	--	74	59
CZ 6A	Minneapolis	-95	--	-6	-7	-17	--	100	59
CZ 7A	Fargo	-86	--	-5	-8	-16	--	99	43
CZ 8	Fairbanks	-169	--	-11	-15	-28	--	189	89
	Min:	-169	--	-11	-26	-37	106	64	25
	Max:	-75	--	-4	6	8	282	189	99
	Avg:	-100	--	-6	-11	-19	161	98	54

Table 11. Percent difference in total annual building energy cost from the Vent1-Exhaust case, by climate zone and ventilation system

		Percent Difference in Total Building Energy Cost from Vent1-Exhaust							
		No Vent	Vent1 Exhaust	Vent2 Hybrid CFIS+Exhaust	Vent3 ERV 70% Recovery	Vent4 CFIS-33%	Vent5 Ventilating Dehumidifier	Vent6 Balanced noRecovery	Vent7 ERV, AHU interlock
		($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)
CZ 1A	Miami	-5.6%	--	-0.4%	0.2%	-0.6%	17.9%	5.9%	4.6%
CZ 2A	Houston	-6.2%	--	-0.3%	-0.4%	-0.8%	10.5%	7.2%	3.6%
CZ 3A	Atlanta	-6.8%	--	-0.3%	-1.5%	-1.9%	6.2%	5.5%	2.0%
CZ 3C	San Jose	-4.5%	--	-0.4%	0.4%	0.5%	--	3.9%	5.8%
CZ 4A	Wash DC, Reagan	-6.7%	--	-0.4%	-1.4%	-2.3%	6.6%	5.8%	2.3%
CZ 4A	Wash DC, Dulles	-7.0%	--	-0.3%	-1.6%	-2.1%	--	6.0%	1.8%
CZ 4C	Seattle	-6.6%	--	-0.3%	-1.5%	-2.5%	--	5.1%	2.0%
CZ 5B	Denver	-5.4%	--	-0.4%	-0.2%	0.4%	--	5.3%	4.3%
CZ 6A	Minneapolis	-5.8%	--	-0.4%	-0.4%	-1.0%	--	6.1%	3.6%
CZ 7A	Fargo	-5.8%	--	-0.3%	-0.5%	-1.1%	--	6.6%	2.9%
CZ 8	Fairbanks	-5.7%	--	-0.4%	-0.5%	-0.9%	--	6.3%	3.0%
	Min:	-7.0%	--	-0.4%	-1.6%	-2.5%	6.2%	3.9%	1.8%
	Max:	-4.5%	--	-0.3%	0.4%	0.5%	17.9%	7.2%	5.8%
	Avg:	-6.0%	--	-0.4%	-0.7%	-1.1%	10.3%	5.8%	3.2%

Table 12. Total annual cooling season cost by climate zone and ventilation system

		Total Cooling Season Cost							
		No Vent	Vent1 Exhaust	Vent2 Hybrid CFIS+Exhaust	Vent3 ERV 70% Recovery	Vent4 CFIS-33%	Vent5 Ventilating Dehumidifier	Vent6 Balanced noRecovery	Vent7 ERV, AHU interlock
		($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)
CZ 1A	Miami	662	732	725	738	728	1016	804	807
CZ 2A	Houston	386	435	431	437	438	568	483	482
CZ 3A	Atlanta	232	265	262	267	275	345	288	303
CZ 3C	San Jose	133	136	134	157	168	--	144	199
CZ 4A	Wash DC, Reagan	204	230	227	233	237	301	249	264
CZ 4A	Wash DC, Dulles	158	180	178	183	193	--	196	212
CZ 4C	Seattle	54	54	54	63	68	--	57	79
CZ 5B	Denver	115	118	117	129	142	--	125	153
CZ 6A	Minneapolis	130	141	140	150	162	--	152	177
CZ 7A	Fargo	76	82	81	89	103	--	88	108
CZ 8	Fairbanks	38	36	36	46	66	--	38	63
	Min:	38	36	36	46	66	301	38	63
	Max:	662	732	725	738	728	1016	804	807
	Avg:	199	219	217	227	235	558	239	259

Table 13. Total annual heating season cost by climate zone and ventilation system

		Total Heating Season Cost							
		No Vent	Vent1 Exhaust	Vent2 Hybrid CFIS+Exhaust	Vent3 ERV 70% Recovery	Vent4 CFIS-33%	Vent5 Ventilating Dehumidifier	Vent6 Balanced noRecovery	Vent7 ERV, AHU interlock
		($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)
CZ 1A	Miami	9	13	13	12	12	17	20	13
CZ 2A	Houston	208	237	236	235	234	280	275	250
CZ 3A	Atlanta	491	569	567	546	531	609	634	569
CZ 3C	San Jose	317	410	406	390	377	--	470	440
CZ 4A	Wash DC, Reagan	483	561	557	539	520	606	630	565
CZ 4A	Wash DC, Dulles	552	639	635	614	596	--	715	642
CZ 4C	Seattle	442	529	525	501	481	--	590	528
CZ 5B	Denver	409	481	477	467	463	--	547	504
CZ 6A	Minneapolis	573	654	650	641	618	--	742	679
CZ 7A	Fargo	634	713	709	700	678	--	804	731
CZ 8	Fairbanks	1404	1578	1567	1553	1520	--	1765	1640
	Min:	9	13	13	12	12	17	20	13
	Max:	1404	1578	1567	1553	1520	609	1765	1640
	Avg:	502	580	577	563	548	378	654	596

Table 14. Annual supplemental dehumidification cost by climate zone and ventilation system

		Total Supplemental Dehumidification Cost							
		No Vent	Vent1 Exhaust	Vent2 Hybrid CFIS+Exhaust	Vent3 ERV 70% Recovery	Vent4 CFIS-33%	Vent5 Ventilating Dehumidifier	Vent6 Balanced noRecovery	Vent7 ERV, AHU interlock
		($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)	($/yr)
CZ 1A	Miami	1	6	6	3	1	0	20	3
CZ 2A	Houston	13	25	25	18	14	0	43	17
CZ 3A	Atlanta	9	14	14	10	9	0	19	10
CZ 3C	San Jose	22	3	3	8	13	--	2	8
CZ 4A	Wash DC, Reagan	6	10	10	6	6	0	13	6
CZ 4A	Wash DC, Dulles	1	8	8	3	2	--	14	3
CZ 4C	Seattle	7	3	3	3	4	--	2	3
CZ 5B	Denver	0	0	0	0	0	--	0	0
CZ 6A	Minneapolis	1	2	2	1	1	--	4	1
CZ 7A	Fargo	1	3	3	2	2	--	5	2
CZ 8	Fairbanks	5	2	2	2	2	--	2	2
	Min:	0	0	0	0	0	0	0	0
	Max:	22	25	25	18	14	0	43	17
	Avg:	6	7	7	5	5	0	11	5

7 DISCUSSION

Referring to Table 9, total annual building energy cost for all the mechanical ventilation cases ranged from a low of $1231 for the Vent4-CFIS-33% case in Seattle to a high of $3177 for the Vent7-ERV with AHU Interlock case in Fairbanks. Referring to Table 10, the difference in total annual building energy cost relative to the Vent1-Exhaust reference case ranged from a savings of $37 for the Vent4-CFIS-33% case in Washington DC-Reagan to an increase of $282 for the Vent5-Ventillating Dehumidifier in Miami. Referring to Table 11, the percent cost difference in total annual building energy cost from the Vent1-Exhaust reference case ranged from a savings of 2.5% to an increase of 7.2% for all systems except for the Vent5-Ventilatiing Dehumidifier in Miami and Houston, where costs increased by 18% and 11%, respectively.

There were only two cases where the total building energy cost was less relative to the Vent1-Exhaust reference case in every climate zone. Those were the No-Vent case and the Vent2-Hybrid CFIS+Exhaust case. The Vent2-Hybrid system consumed less energy in all climate zones because whenever there was a call for heating or cooling, outdoor ventilation air was drawn in at no additional fan energy cost, filtered, conditioned, and supplied along with the recirculating house air. Exhaust ventilation was then activated in the remaining balance of time.

Comparing the Vent3-ERV system with dedicated ducts to the Vent1-Exhaust reference case, the total building energy cost was $3-$26 less in all climate zones except Miami and San Jose where it was $3 more and $6 more, respectively. In Miami, with 5 ACH50 building tightness and balanced ventilation, the sum of infiltration and balanced ventilation made the total outdoor airflow, averaged over the year, 97 CFM for the Vent3-ERV case and 70 CFM for the Vent1-Exhaust case. The 2018 IRC ventilation rates do not account for the difference in infiltration between balanced and unbalanced ventilation systems. That additional outdoor air exchange and the higher ERV fan power slightly overcame the ERV energy recovery benefit. An interesting point to make for the San Jose marine climate is that even though the outdoor absolute humidity is within the range of comfortable indoor absolute humidity throughout the year, indoor moisture generation was enough to require supplemental dehumidification to control indoor RH to 55% RH (+/- 2% control deadband). That is an example where the ERV was counter-productive because its latent recovery kept moisture indoors when it would have been better to expel it.

To examine the benefit of energy recovery alone, the Vent3-ERV w/dedicated ducts case was compared to the Vent6-Balanced noRecovery case. The total building energy cost savings due to energy recovery alone ranged from $61/yr in San Jose to $204/yr in Fairbanks. Determining the difference in first-cost between those two cases in Table 16 to be $750, the simple payback for energy recovery alone ranged from 3.7 years in Fairbanks to 12.3 years in San Jose. Other advantages of energy recovery would be improved comfort due to ventilation air tempering, and reduced risk of central system supply duct condensation in humid climates.

Most ERV installations are as the Vent7-ERV w/AHU interlock case. To examine the effect of the AHU interlock alone, the Vent7-ERV w/AHU interlock case was compared the Vent3-ERV w/dedicated ducts case. The annual total building energy cost due to the AHU interlock ranged from $44/yr in Seattle to $104/yr in Fairbanks. From Table16, installing a separate duct system would cost about $250 more than the AHU interlock. The simple payback for installing a separate ERV duct system ranged from 2.4 years in Fairbanks to 5.7 years in Seattle.

The Vent5-Ventilating Dehumidifier case was only simulated for the humid climates of Miami, Houston, Atlanta, and Washington DC-Reagan. It was the highest cost system to operate in all those locations. The difference in annual total building energy cost compared to the Vent1-Exhaust reference case ranged from $106 in Atlanta and Washington DC to $282 in Miami. In Miami, the dehumidifier compressor ran most of the time with the dehumidifier fan providing supply ventilation. The dehumidifier also adds the heat of condensation and the compressor heat to the cooling load.

While relatively low cost, some supplemental dehumidification was required to maintain the space conditions below 57% RH (55% RH +/- 2% control deadband) in all climates and all ventilation cases except Denver and the Vent5-Ventilating Dehumidifier case. The highest supplemental dehumidification cost was $43 in Houston for the Vent6-Balanced noRecovery case. Energy recovery in the Vent3-ERV case brought that down to $18 but did not eliminate the need for supplemental dehumidification. Vent4-CFIS-33% brought it down even further to $14 because the outdoor air was directly dehumidified when cooling was active and the net air change rate for this system was slightly lower without the automatic backup exhaust of the Vent2 system. Supplemental dehumidification was needed in the spring and fall seasons when the ERV was mostly ineffective in reducing latent load due to small or negative outdoor to indoor absolute humidity differences. That is illustrated in Figure 7 and Figure 8 which show that the supplemental dehumidifier was operating mostly in the spring and fall seasons to limit the indoor RH to 57%.

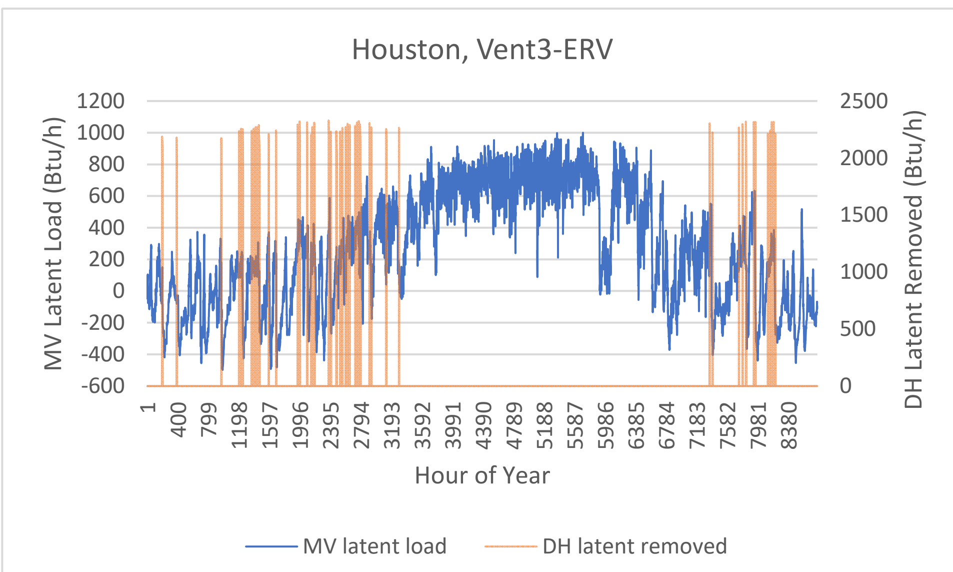

Figure 7. Mechanical ventilation latent load and latent heat removed by the supplemental dehumidifier for the Vent3-ERV case in Houston

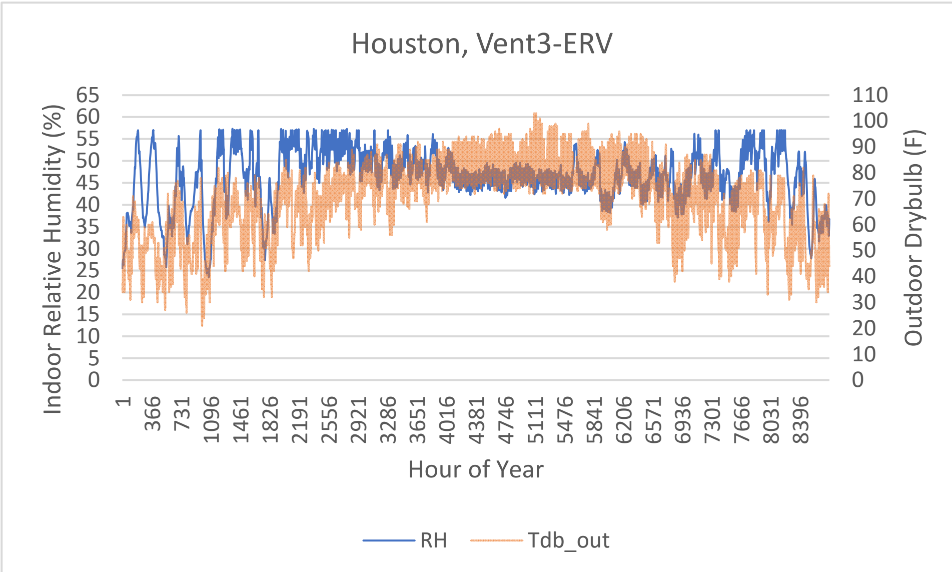

Figure 8. Indoor RH and outdoor drybulb temperature for the Vent3-ERV case in Houston

Figure 9 shows consistent cooling demand in Miami which mostly eliminates the need for supplement dehumidification (Figure 10) to control the indoor RH below 57% (Figure 11).

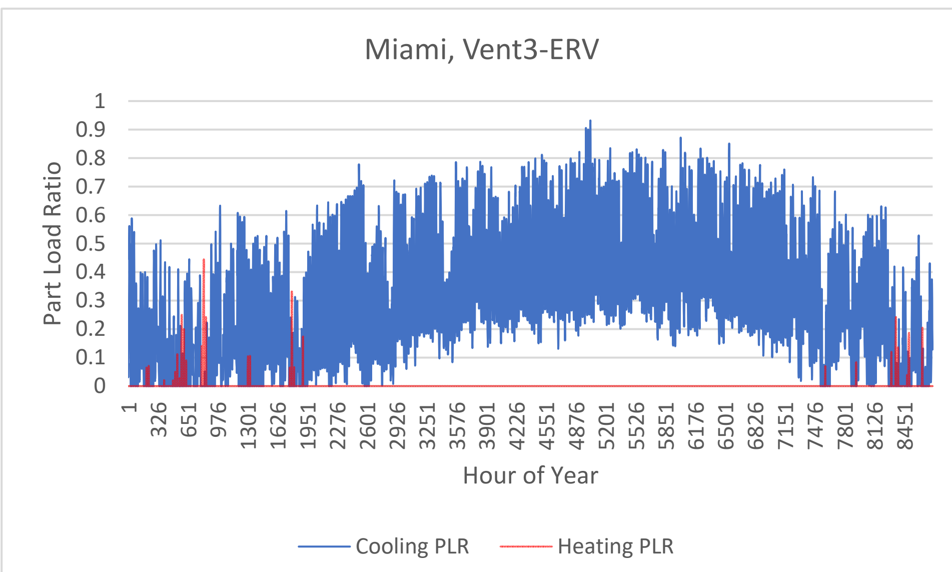

Figure 9. Cooling and heating part load ratios for the Vent3-ERV case in Miami

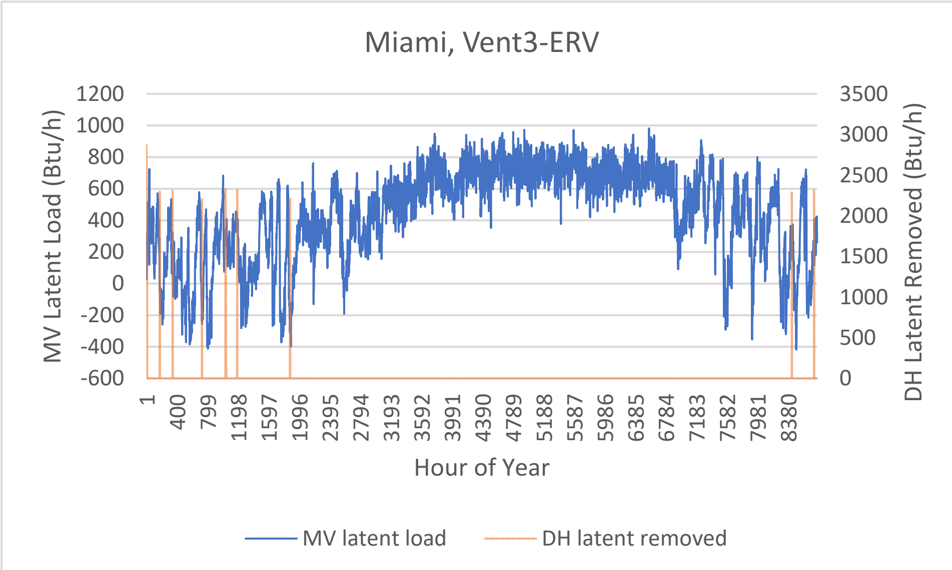

Figure 10. Mechanical ventilation latent load and latent heat removed by the supplemental dehumidifier for the Vent3-ERV case in Miami

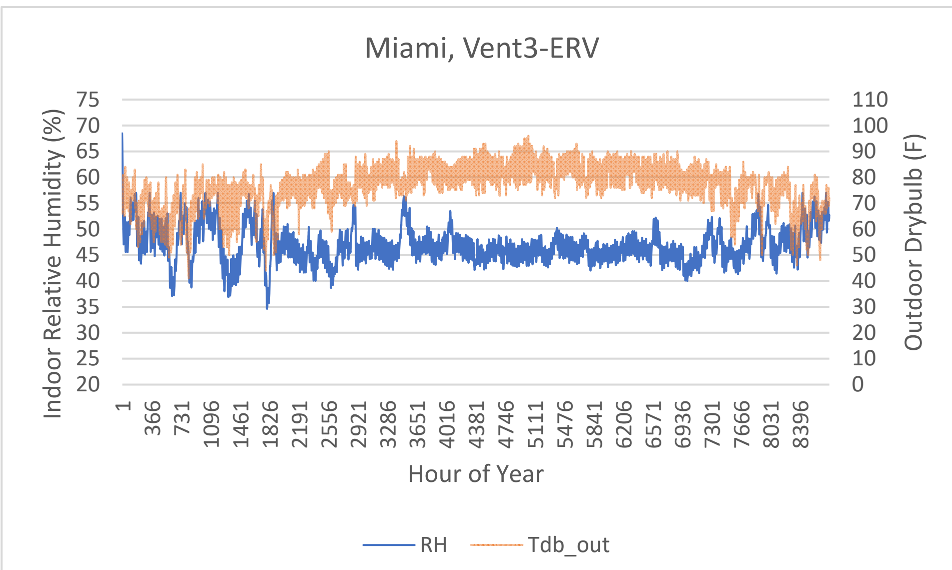

Figure 11. Indoor RH and outdoor drybulb temperature for the Vent3-ERV case in Miami

The matrix in Table 15 compiles the results from this study along with known mechanical system characteristics into a set of recommended mechanical ventilation system applications by climate zone. The footnotes provide additional insight into the reasons for the categorizations.

Table 15. Matrix of recommended ventilation system applications by climate zone (R=Recommended, A=Acceptable, NR=Not Recommended)

	Ventilation System	1A Miami	2A Houston	3A Atlanta	3B San Jose	4A Wash DC	4C Seattle	5 Denver	6 Minneapolis	7 Fargo	8 Fairbanks
1	Exhaust, single-point	NR[1,2]	NR[1,2]	NR[1,2]	A[2]	A[1,2]	A[1,2]	A[2]	A[2]	A[2]	A[2]
2	Hybrid CFIS w/ automatic Exhaust backup	R	R	R	R	R	R	R	A[3,4]	NR[3,4]	NR[3,4]
3	Balanced ERV, dedicated ducts	R	R	R	R	R	R	R	R	R	R
4	CFIS-33% baseline w/occupant-controlled Exhaust	R	R	R	R	R	R	R	A[3,4]	No[3,4]	No[3,4]
5	Ventilating Dehumidifier w/ compressor or AHU interlock	NR[5,6]	A[6]	A[6]	NR[7]	A[6]	NR[6]	NR[7]	NR[6]	NR[6]	NR[7]
6	Balanced Supply and Exhaust, no recovery, no interlock	A[8]	NR[8]	NR[8]	NR[8]	NR[8]	NR[8]	NR[8]	NR[8]	NR[8]	NR[8]
7	Balanced ERV w/ AHU interlock	A[9]	A[9]	A[9]	R	A[9]	R	R	R	R	R

[1] Depressurization moisture concern with vapor retarding (Class I or II) interior surfaces.

[2] Potential for affecting indoor air quality performance due to uncontrolled source-path of outdoor air, lack of outdoor air filtration and lack of ventilation air distribution.

[3] Potential pressurization moisture concern without spray foam or exterior rigid insulation to increase the first condensing plane temperature.

[4] Potential low mixed-air-return temperature concern for gas furnaces. This turns Acceptable to Not Recommended in Climate Zones 7 and 8.

[5] Consistent cooling operation effectively removes moisture such that the high operating cost and added sensible heat gain of the ventilating dehumidifier system makes it un-economic. For times outside of consistent cooling operation, a dehumidifier operated in recirculation-only mode is recommended instead.

[6] The high operating cost and added sensible heat gain of the ventilating dehumidifier system makes it un-economic. Mostly needed for times outside of consistent cooling operation, a dehumidifier operated in recirculation-only mode is recommended instead.

[7] In this climate, outdoor air is consistently drier than indoor air, so dehumidification of outside air is not needed and would be an energy waste. In this climate, more outside air decreases indoor dehumidification demand, whereas in humid climates more outside air increases indoor dehumidification demand.

[8] Discomfort concern from un-tempered outdoor air supply.

[9] In the cooling season in humid climates, operating the air handler unit fan without time for the cooling coil to drain after compressor deactivation results in excessive water evaporation from the wet cooling coil. This negatively affects indoor humidity control and indoor humidity comfort.

8 VENTILATION SYSTEM FIRST-COST ANALYSIS

Material and installation labor cost analysis was conducted to provide an estimate of system cost for comparison against energy savings. The costs were built up by researching trade pricing on equipment and by making hourly installation time estimates multiplied by an hourly rate estimate of $70/h. The Relative Cost column facilitates potential use of the cost information further into the future by simply applying ratios of the cost multiples.

Table 16. Total estimated first-cost derived from material and installation labor cost estimates for each mechanical ventilation system, and relative cost factor

		Total Cost ($)	Relative Cost
1	Exhaust, single-point	190	1.0
2	Hybrid CFIS w/ automatic Exhaust backup	703	4
3	Balanced ERV, dedicated ducts	1,515	8
4	CFIS-33% baseline w/occupant-controlled Exhaust	520	3
5	Ventilating Dehumidifier Supply w/ compressor or AHU interlock	1,875	10
6	Balanced Supply and Exhaust, no recovery, no interlock	765	4
7	Balanced ERV w/ AHU interlock	1,265	7

Estimated ventilation system first-cost ranged by a factor of 10 from $190 for single-point exhaust to $1,875 for a ventilating dehumidifier supply with compressor or air-handler unit interlock.

9 IDENTIFICATION OF RESEARCH GAPS AND POTENTIAL CODE-CHANGE RISKS

9.1 High Efficiency Air Filtration

As mentioned earlier, there has been a major shift in the scientific community's understanding of indoor air contaminants of concern and the relationship to occupant health. The focus has turned significantly toward small particles, mainly particle matter 2.5 micrometer (micron) in diameter and smaller, referred to as PM2.5. With the new focus on PM2.5, there has been much discussion about moving to MERV 13 minimum filtration to replace the current MERV 6 minimum within the ASHRAE 62.2 Standard. The California Building Code has already gone that route.

MERV 6 to 7 filtration has traditionally been recommended by manufacturers for protecting central system thermal conditioning equipment against fouling. Higher MERV filters would be unnecessary for that purpose. Unless measures are taken to increase filter surface area, moving to higher efficiency filtration within the central space conditioning system will have a significant negative effect on central system fan performance, energy consumption, and even fan longevity due to the much greater airflow resistance inherent with the more efficient filters. More expensive 4" to 5" thick wide pleated media filtration will become necessary replacement for standard 1" thick filters. Return air duct design will need to adjust to reduce overall system pressure drop to stay within the manufacturers' specifications and equipment rating.

New filter products have already emerged to allow existing 1" return air filter-grille assemblies to accept wide pleated media filters if there is available height behind the filter-grille to accommodate that. However, more research on this topic would help designers and home builders find the best ways to keep central system fan pressure within bounds when moving to higher filtration efficiency.

The ever-more-popular variable capacity mini-split and multi mini-split heat pump systems are especially susceptible to this challenge because their fan systems are designed for no ducting or limited ducting and low filter resistance. In addition, their popular high efficiency ratings are in large part due to low fan power requirements, so higher efficiency filtration requirements would affect the space conditioning product performance.

Finally, considering the even more recent concerns about virus filtration, more research and education would be prudent to avoid misapplication of high efficiency filtration in homes. The particle size of viruses is smaller than 0.3 micron (0.005 – 0.3), and MERV 13 filters are only 50% efficient at capturing particles between 0.3 to 1.0 micron in size. Therefore, it will be important to fully understand the implications and practicality of trying to achieve high efficiency filtration of very small particles with the space conditioning systems that home builders install.

9.2 Ventilation Rates

This study showed that with the IRC/IECC code level ventilation rate, following the ASHRAE Standard 62.2-2010 ventilation rate, and a reasonable rate of internal moisture generation for a family of four (12 lb/day), there were occurrences of varying duration (shorter in the less humid climates) when RH exceeded the 57% control limit in all locations examined except Denver.

Without considering any infiltration credit, the ASHRAE Standard 62.2 mechanical ventilation rate increased by 80% starting in 2013 for a 2000 ft^2, 3-bedroom house. There was no specific health-based or medical-research-based justification for that. Now that more recent research has shown that PM2.5 is many times more impactful to human health compared to the closest indoor gaseous pollutants of concern in homes, there is even less health-based justification for that 2013 increase in ventilation rate. If the building codes adopt that higher mechanical ventilation rate, humid climate moisture control will become much more of a challenge. That challenge will likely need to be met with the installation of supplemental dehumidification in potentially all of climates zones 1A through 3A and many coastal regions of 4A. That would come with additional construction costs and operating costs.

10 CONCLUSIONS

All of the dwelling unit mechanical ventilation systems examined are feasible. However, for a particular dwelling unit and location there are both economic and non-economic reasons to prefer one system over another based on best practices related to indoor air quality and building science, compatibility of other mechanical equipment, optimal occupant comfort, lower first-cost, or lower operating cost. Ultimately, the builder must know their market and choose a system based on market preferences and constraints.

From a high-level perspective without complex caveats, the following table shows the recommended systems and associated climate zones resulting from the seven mechanical ventilation systems modeled:

Ventilation System		Climate Zone
No.	**Description**	
2	Hybrid Central-Fan-Integrated Supply (CFIS) with automatic Exhaust backup	1-5
3	Balanced Energy Recovery Ventilator (ERV) with dedicated ducts	1-8
4	CFIS-33% baseline with occupant-controlled Exhaust	1-5
7	Balanced ERV with AHU interlock	3B. 4C, 5-8

The difference in total annual building energy cost relative to the Vent1-Exhaust reference case ranged from a savings of $37 for the Vent4-CFIS-33% case in Washington DC-Reagan to an increase of $282 for the Vent5-Ventillating Dehumidifier in Miami. The percent cost difference in total annual building energy cost from the Vent1-Exhaust reference case ranged from a savings of 2.5% to an increase of 7.2% for all systems except for the Vent5-Ventilatiing Dehumidifier in Miami and Houston, where costs increased by 18% and 11%, respectively.

If setting aside the first-cost and operating cost disbenefit, the Vent5-Ventilating Dehumidifier as modeled here can be an effective ventilation system for humid locations. As another example, the Vent3-Balanced ERV with dedicated duct system will work in every climate but it is one of the most expensive systems to install and lacks the benefit of whole-house recirculation filtration for particulate removal and mixing for improved indoor comfort. That may lead one to choose the Vent7-Balanced ERV integrated with the central duct system and with central system fan interlock, but that has higher energy consumption and negative humidity control consequences in the humid climates due to evaporation of water from wet coils when the cooling compressor is not operating. On the other end of the spectrum, the Vent1-Exhaust system has the lowest first-cost and also one of the lowest operating costs, however, there are drawbacks in: a) indoor air quality performance since the source of outdoor air is not controlled and the air may be bringing pollutants with it; b) poor ventilation air distribution performance; and c) lack of recirculation air filtration and comfort mixing.

Except for the coldest climates, a middle ground may be the Vent4-CFIS-33% baseline system with occupant-controlled exhaust has low first-cost, low operating cost, good air filtration and comfort mixing characteristics, but, if the occupant does not activate the occupant-controlled exhaust, the resulting average ventilation rate will be lower than what may be specified by code. That may lead one to the Vent2-Hybrid CFIS with automatic exhaust backup which increases first-cost by nearly $200 and the recirculation filtration and comfort mixing benefit is reduced.

For balanced mechanical ventilation systems, simple-payback for energy recovery ranged from 3.7 years in Fairbanks to 12.3 years in San Jose. Simple-payback for installing a separate ERV duct system versus interlocking with the AHU ranged from 2.4 years in Fairbanks to 5.7 years in Seattle.

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