Whole House Ventilation System Options – Phase 1 Simulation Study

Research Report - 0701

2007-04-05 by Joseph Lstiburek, Betsy Pettit, Armin Rudd, Building Science Corporation, and Max Sherman and Ian Walker, EPB Group

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Abstract:

The final report below includes a summary of the simulation plan and analysis.

About this Report

This report was prepared for:

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WHOLE HOUSE VENTILATION SYSTEM OPTIONS – PHASE 1 SIMULATION STUDY

Final Report

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Team Leader:

BUILDING SCIENCE CORPORATION
Joseph Lstiburek, Ph.D., P. Eng., Principal Investigator
Betsy Pettit, AIA, Project Manager
Armin Rudd, Principal Research Engineer

70 Main Street
Westford, MA 01886
ph. 978.589.5100
fax 978.589.5103
www.buildingscience.com
contact betsy@buildingscience.com

Team Members:

Max Sherman, Ph. D. and Iain Walker, Ph.D., EPB Group

Prepared for
AIR-CONDITIONING AND REFRIGERATION TECHNOLOGY INSTITUTE, INC
4100 N. Fairfax Drive, Suite 200, Arlington, Virginia 22203-1678

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05 March 2007

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1. SUMMARY OF TECHNICAL REVIEW OF LITERATURE (TASK 1)

A comprehensive literature review was made to investigate whole house ventilation system options, various simulation and engineering analysis tools and techniques, and baselines for comparing the current project results.

The literature reviewed included the following:
- Literature listed in the request for proposal (RFP)
- Literature referenced in the project proposal (See REFERENCES)
- Literature listed in the residential ventilation list of the Lawrence Berkeley Laboratory (http://epb.lbl.gov/Publications/ventilation.html)

Additionally the proprietary database of the Air Infiltration and Ventilation Centre, AIRBASE, (http://www.aivc.org) was searched for relevant literature. AIRBASE contains over 15000 references related to air infiltration and ventilation and is the worlds' most authoritative source on the topic.

The deliverable for Task 1 is in the form of two LBNL reports -- one on Residential Ventilation Requirements, and one on Residential Ventilation Technologies, as follows:

McWilliams, J.A. and M.H. Sherman "Review of Literature Related to Residential Ventilation Requirements," June 2005. LBNL-57236. http://epb.lbl.gov/Publications/lbnl-57236.pdf

Abstract

This paper reviews current ventilation codes and standards for residential buildings in Europe and North America. It also examines the literature related to these standards such as occupant surveys of attitudes and behavior related to ventilation, and research papers that form the technical basis of the ventilation requirements in the standards. The major findings from the literature are that ventilation is increasingly becoming recognized as an important component of a healthy dwelling, that the ventilation standards tend to cluster around common values for recommended ventilation rates, and that surveys of occupants showed that people generally think that ventilation is important, but that their understanding of the ventilation systems in their houses is low.

Russell, M, M.H. Sherman and A. Rudd "Review of Residential Ventilation Technologies," August 2005. LBNL-57730. http://epb.lbl.gov/Publications/lbnl-57730.pdf

Abstract

This paper reviews current and potential ventilation technologies for residential buildings with particular emphasis on North American climates and construction. The major technologies reviewed include a variety of mechanical systems, natural ventilation, and passive ventilation. Key parameters that are related to each system include operating costs, installation costs, ventilation rates, and heat recovery potential. It also examines related issues such as infiltration, *duct systems, filtration options, noise, and construction issues. This report describes a wide variety of systems currently on the market that can be used to meet ASHRAE Standard 62.2-2004. While these systems generally fall into the categories of supply, exhaust or balanced, the specifics of each system are driven by concerns that extend beyond those in the standard and are discussed. Some of these systems go beyond the current standard by providing additional features (such as air distribution or pressurization control). The market will decide the immediate value of such features, but ASHRAE may wish to consider related modifications to the standard in the future.*

The second report was accepted as an ASHRAE Research Journal article. The Abstracts are reproduced here in this section. The full texts are provided as attachments to this final report.

2. SUMMARY OF SIMULATION PLAN (TASK 2)

Using the results of the Task 1 Literature Review, a detailed plan was made for conducting simulations and analyses of the performance of whole-house mechanical ventilation systems in new residential buildings. The simulation plan specified the key variables to be studied and defined the criteria to be used in the analysis. Key variables included: ventilation system type and controls; a full range of climate zones, seasons and peak weather; and building description details such as physical properties, operating schedules, internal heat gain rates, and occupancy.

As part of the development of this plan, the project team presented to and received input from the ASHRAE SSPC 62.2 and participants of a DOE Building America Ventilation Expert meeting.

The plan was submitted to the ARTI project monitoring subgroup (PMS) and approved prior to proceeding with the Task 3 Simulation and Analysis. However, in the course of analyzing the simulation results, a number of questions arose illuminating areas where the Simulation Plan fell short, necessitating adjustment of some inputs and assumptions. These changes were also presented to and approved by the PMS. In addition, the PMS requested a several changes: 1) changing to the reference house from a leaky house to a house with the same tighter envelope as the mechanically ventilated houses, 2) doing detailed humidity calculations in all climates, and 3) expanding HRV/ERV simulations to more climates.

The final Simulation Plan is reproduced here in part for clarity because of the revisions, and to more conveniently assist the reader in interpreting the Simulation and Analysis results of Task 3. Appendix I of the Simulation Plan, which provides the details the REGCAP simulation, REGCAP, is included as Appendix D to this report.

Introduction

This simulation plan outlines the pertinent ASHRAE Standard 62.2 requirements that will be simulated using different ventilation technologies. The information required to simulate each approach is summarized together with rationales for selection of particular parameters. The technologies are discussed in more detail in the companion Ventilation Technologies Review report.

The HVI Directory[1] was used to obtain fan power for fans that met the airflow and sound requirements of ASHRAE Standard 62.2 In this plan the specific fan manufacturers and model numbers are given in square parentheses [] for each system.

Approximately 100 different combinations of house size, climate and ventilation technologies will be simulated in all. The REGCAP[2] simulation model will be used to perform minute-by-minute simulations combined with post-processing to answer key questions. The REGCAP model has been used in several previous studies looking at HVAC system performance[3]. REGCAP has a detailed airflow network model that calculates the airflow through building components as they change with weather conditions and HVAC system operation. The airflows include the effects of weather and

[1] HVI. 2005. Certified Home Ventilating Products Directory, Home Ventilating Institute.
[2] Appendix D gives details of the simulation model.
[3] See REGCAP Bibliography at the end of Appendix D.

leak location, and the interactions of HVAC system flows with house and attic envelopes. These airflow interactions are particularly important because the airflows associated with ventilation systems (including duct leakage) interact significantly with the effects of natural infiltration in houses.

Because REGCAP performs minute by minute simulations, the dynamic effects of HVAC systems are captured. This includes issues of cyclic duct losses and latent capacity of air conditioners.

Houses to be simulated

Three house sizes will be simulated to examine the implicit effect of occupant density in the ASHRAE Standard 62.2 requirements. For most of the simulations the medium sized house will be used, and for selected cases smaller and larger houses will be simulated.

1. Small 1000 ft^2 2 bedroom Bungalow.
2. Mid-size 2000 ft^2, 2 story, 3 bedrooms
3. Large 4000 ft^2 2-story, 5 bedrooms.

Two levels of envelope leakage will be examined. The high leakage values in Table 2.4 will be used as a "non-mechanically ventilated ASHRAE Standard 62.2 compliant house". (*In the original simulation plan this was to be the reference house.)* For the mechanical ventilation simulations, the envelopes will be tighter. The tighter envelope will also be used for a set of simulations with no ASHRAE Standard 62.2 compliant mechanical ventilation. This tight but unventilated house will be used as a basis of comparison for the other simulations. From the LBNL air leakage database for typical new construction[4] the Normalized Leakage is NL=0.3 (or about 6 ACH50). The corresponding leakage values are summarized in Table 2.1.

Table 2.1 Envelope Leakage for Mechanically Ventilated Homes

Floor Area (ft^2)	ACH 50	ELA4 (in^2)	m^3/sPa^n	cfm/Pa^n
1000	5.8	43	0.028	61
2000	5.8	86	0.057	121
4000	5.8	173	0.114	243

Building insulation and duct system parameters used to determine the non-ventilation building load will vary by climate as shown in Table 2.2. The envelope characteristics are based on a house that meets IECC requirements and are referred to as the Standard-Performance houses. Exterior surface area for wall insulation scales with floor area and number of stories. The surface area is typically three times the floor area (based on the BSC/Building America data set). Window area is 18% of floor area with windows equally distributed in walls facing in the four cardinal directions.

[4] *Sherman, M.H. and Matson, N.E.,***"Air Tightness of New U.S. Houses: A Preliminary Report "**, Lawrence Berkeley National Laboratory, LBNL-48671, 2002

Higher-Performance Houses

Simulation of higher-performance houses broadens the application of the results for above-code programs such as the USDOE Building America Program. The higher-performance houses will be simulated for all medium house size cases. The differences between the higher-performance and standard-performance houses are:

- Tighter envelopes. The higher-performance houses have half the envelope leakage.
- All ducts inside conditioned space
- Glazing has U=0.35 and SHGC=0.35. It should be noted that in some heating dominated climates, the lower SHGC can increase heating load more than it reduces cooling load.
- Changed thermostat to be constant setpoint: 76°F cooling, 70°F heating
- Changed heating and cooling capacities to match ACCA Manual J loads.

Table 2.2 Building enclosure and duct system parameters for the standard IECC houses and the higher-performance houses

IECC reference house (no mech vent, programmed setpoints on weekdays 68/70, 76/80)

	Bldg leakage (ach50)	Duct leakage (% of flow)	Duct location	Glazing		Insulation			
				SHGC	U-value	Ceiling	Wall	Foundation	Ducts
Houston	5.8	5	out, attic	0.40	0.75	R-30	R-13	slab, none	R-8
Phoenix	5.8	5	out, attic	0.40	0.75	R-30	R-13	slab, none	R-8
Charlotte	5.8	5	out, attic	0.40	0.65	R-30	R-15	slab, R-4	R-8
Kansas City	5.8	2.5	attic, bsmt	0.40	0.35	R-38	R-15	basement, R-10	R-8
Seattle	5.8	5	out, attic	0.40	0.40	R-38	R-21	Crawlspace, R-22 flr	R-8
Minneapolis	5.8	2.5	attic, bsmt	0.40	0.35	R-49	R-21	bsmt wall, R-10	R-8

Higher-performance houses (mech vent, constant setpoints 70, 76)

	Bldg leakage (ach50)	Duct leakage (% of flow)	Duct location	Glazing		Insulation			
				SHGC	U-value	Ceiling	Wall	Foundation	Ducts
Houston	3.0	0	inside	0.35	0.35	R-30	R-13	slab, none	n/a
Phoenix	3.0	0	inside	0.35	0.35	R-30	R-13	slab, none	n/a
Charlotte	3.0	0	inside	0.35	0.35	R-30	R-15	slab, R-4	n/a
Kansas City	3.0	0	inside	0.35	0.35	R-38	R-15	basement, R-10	n/a
Seattle	3.0	0	inside	0.35	0.35	R-38	R-21	Crawlspace, R-22 flr	n/a
Minneapolis	3.0	0	inside	0.35	0.35	R-49	R-21	bsmt wall, R-10	n/a

Meeting ASHRAE Standard 62.2 Requirements

Except for Systems 5 and 5a described below, the simulations will all meet the minimum requirements of ASHRAE Standard 62.2 as follows:

Whole Building Ventilation

Mechanical ventilation sized as follows:

$$Q(cfm) = 0.01A_{floor}\left(ft^2\right) + 7.5(N+1)$$
$$Q(L/s) = 0.05A_{floor}\left(m^2\right) + 3.5(N+1) \quad \textbf{(1)}$$

For the three house sizes:
1000 ft^2 & 2 bedrooms (3 occupants) $\Rightarrow$ 32.5 cfm
2000 ft^2 & 3 bedrooms (4 occupants) $\Rightarrow$ 50 cfm
4000 ft2 & 5 bedrooms (6 occupants) $\Rightarrow$ 85 cfm

Using continuous operation of bathroom exhaust requires a minimum of 20 cfm (Table 5.2 in ASHRAE Standard 62.2), and all of these proposed systems exceed this minimum.

Infiltration Credit

To represent an older leakier house the infiltration credit assumptions are laid out in section 4.1.3 of ASHRAE Standard 62.2 will be used to estimate the envelope leakage. The envelope leakage is calculated using the weather factors from ASHRAE Standard 136 and the airflow requirements from section 4.1.3 of ASHRAE Standard 62.2, i.e.:

ASHRAE Standard 136 airflow = 2×Q (From Equation 1) + 2 cfm/100ft^2 floor area

For the three houses Table 2.3 shows the ventilation rates (in ACH) that the houses would need to have to match this ASHRAE Standard 62.2 requirement. Table 2.4 summarizes the weather factors from Standard 136, and the corresponding envelope leakage to be used in the simulations.

Table 2.3 Summary of ASHRAE Standard 62.2 Mechanical Ventilation Requirements and Envelope Airflow Requirements Based on the Infiltration Credit Estimate from Section 4.1.3

House Floor Area	Number of Stories	Number of Bedrooms	ASHRAE 62.2 Mechanical Ventilation Sizing	Building Envelope Airflow Requirement
ft^2			cfm	ACH
1000	1	2	32.5	0.64
2000	2	3	50	0.53
4000	2	5	85	0.47

Table 2.4 Envelope Leakage to meet ASHRAE Standard 62.2 Infiltration Requirements

Location	Weather Factor From ASHRAE 136	Building Envelope Airflow Requirement From Table 1	Normalized Leakage	Leakage Area	Envelope Leakage Coefficient	ACH 50
		ACH		ft^2	m^3/sPa^n	
1000 ft^2, one story						
Seattle	0.85	0.64	0.75	0.75	0.073	13.9
Phoenix	0.68	0.64	0.94	0.94	0.091	17.4
Minneapolis	0.97	0.64	0.66	0.66	0.064	12.2
Kansas City	0.85	0.64	0.75	0.75	0.073	13.9
Charlotte	0.74	0.64	0.86	0.86	0.084	16.0
Houston	0.81	0.64	0.79	0.79	0.077	14.6
2000 ft^2, 2 story						
Seattle	0.85	0.53	0.62	1.00	0.098	9.3
Phoenix	0.68	0.53	0.77	1.25	0.122	11.6
Minneapolis	0.97	0.53	0.54	0.88	0.086	8.1
Kansas City	0.85	0.53	0.62	1.00	0.098	9.3
Charlotte	0.74	0.53	0.71	1.15	0.112	10.7
Houston	0.81	0.53	0.65	1.05	0.103	9.8
4000 ft^2, 2 story						
Seattle	0.85	0.47	0.55	1.80	0.175	8.3
Phoenix	0.68	0.47	0.69	2.25	0.219	10.4
Minneapolis	0.97	0.47	0.48	1.57	0.153	7.3
Kansas City	0.85	0.47	0.55	1.80	0.175	8.3
Charlotte	0.74	0.47	0.64	2.06	0.201	9.6
Houston	0.81	0.47	0.58	1.89	0.184	8.7

Intermittent Operation

To determine if an intermittent fan meets ASHRAE Standard 62.2 requirements, the exception to section 4.4 shows how to calculate the fan flow rate required to meet the ventilation air requirement (from Equation 1). For the intermittent exhaust system, the daily fractional on-time is 20/24 = 83%. From Table 4.2 in ASHRAE Standard 62.2, the ventilation effectiveness is still 1.0 so the fan flow rate is simply pro-rated by the fractional on-time.

Additional ASHRAE Standard 62.2 requirements

All the fans used to provide mechanical ventilation will be selected to meet the sound and installation requirements of ASHRAE Standard 62.2. From an energy use perspective, the main effect is that fans that meet the 1.0 Sone requirement for continuous operation and 3 Sones for intermittent operation tend to be energy efficient fans that also have power ratings in the HVI directory[5].

[5] HVI. 2005. Certified Home Ventilating Products Directory, Home Ventilating Institute.

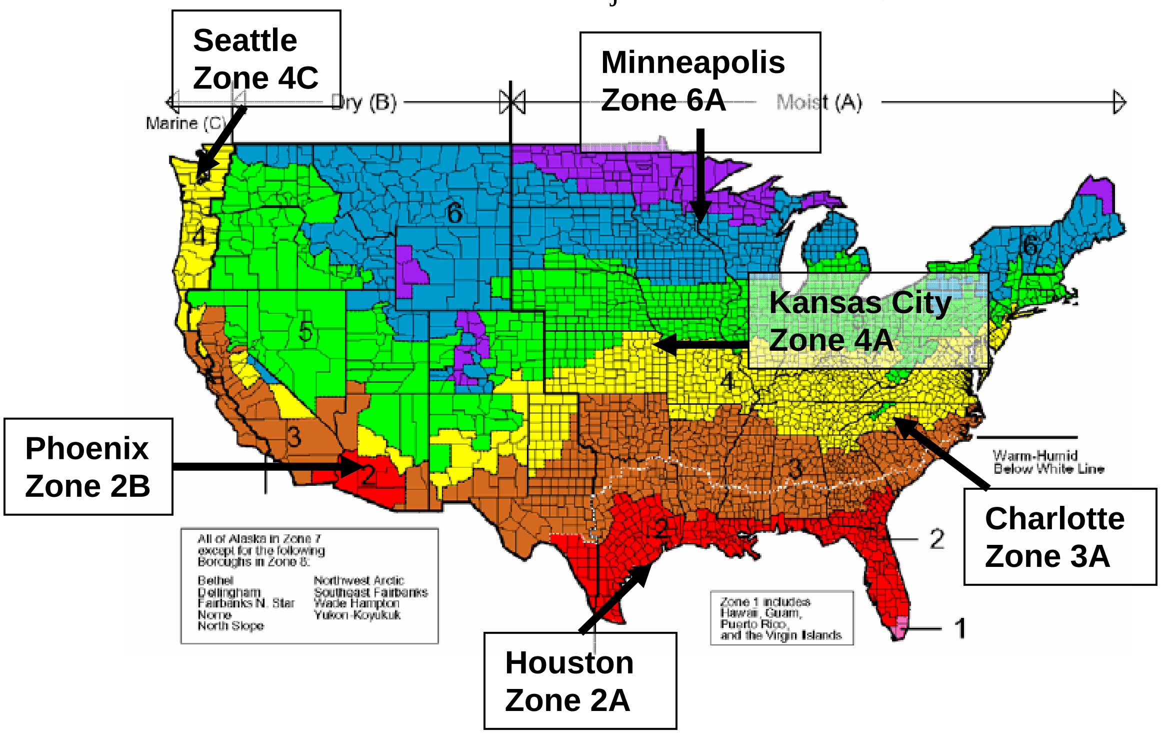
Some simulations will include additional ventilation related airflows and mechanical ventilation operation that contribute to ventilation rates and energy use, but are not considered part of the ASHRAE Standard 62.2 compliance requirements. These are primarily the intermittent occasional operation of kitchen and bathroom fans.

Exceptions to ASHRAE Standard 62.2 Requirements

Standard ASHRAE Standard 62.2 allows an exception to the mechanical ventilation requirement based on climate, occupancy and window operation. These simulations will not examine this exception.

Weather

Six locations will be used that cover the major US climate zones:



TMY2 hourly data files will be used that are converted to minute-by-minute format by linear interpolation. The simulations also use location data (altitude and latitude) in solar and air density calculations. The required weather data for the simulations are:

- direct solar radiation (W/m^2)
- total horizontal solar radiation (W/m^2)
- Outdoor temperature(°C)
- humidity ratio
- wind speed (m/s)
- wind direction (degrees)
- barometric pressure (kPa)

Heating and Cooling Equipment

Equipment sizing will be based on Manual J calculations. Equipment sizing is most important when considering systems that use the central air handler to distribute ventilation air because the outside air is usually supplied as a fraction of total air handler flow. The heating will be supplied by a standard 78% AFUE gas furnace. For cooling, a standard SEER 13 split system air conditioner will be used. All systems will have correct air handler flow and refrigerant charge, so air conditioner capacity and EER will only depend on outdoor temperature.

The duct leakage will be 5%, split with 3% supply leakage and 2% return leakage. For the basement houses in Kansas City and Minneapolis, the leakage is 1.5% supply and 1% return, as it is assumed that the basements are inside conditioned space and half the ducts are in the basement. This level of leakage is lower than typical new construction but use of sealed ducts is a requirement in the IECC and should be a requirement for ducts used for ventilation. It is also the leakage found in BSC houses using ventilation systems that utilize the ducts to distribute ventilation air.

Operation of the heating and cooling equipment will use the weekday set-up and set-back thermostat settings for all Standard-Performance houses and climates in Table 2.5. For the higher-performance houses, the thermostat setpoints will be held constant at 70 °F for heating and 76 °F for cooling.

Table 2.5 Week day Thermostat Settings for Ventilation Simulations

Hour	Heating 68°F/70°F	Cooling 76°F/78°F
1	68	76
2	68	76
3	68	76
4	68	76
5	68	76
6	68	76
7	68	76
8	70	78
9	70	78
10	70	78
11	70	78
12	70	78
13	70	78
14	70	78
15	70	78
16	70	78
17	70	76
18	70	76
19	70	76
20	70	76
21	70	76
22	70	76

23	70	76
24	68	76

On weekends the cooling setpoint is not set up to 78°F and remains at 76°F. The deadband for the thermostat is 0.5°C or 0.9°F. Heating and cooling are available every minute of the year and the operation of heating and cooling equipment is solely decided by the indoor temperature compared to the thermostat settings in Table 2.5.

In the original simulation plan, the cooling setpoints were 77/80 and the switch-over from heating to cooling (and back again) was on selected days for each climate. This led to insufficient air conditioner and furnace operation with periods of indoor temperatures that were either too hot or too cold and less humidity control in shoulder seasons..

Tables 2.6 and 2.7 summarize the equipment efficiencies, capacities, and blower power to be used in the simulations for the standard and higher-performance houses. For the Standard-Performance houses, ACCA Manual J sizing or greater was used. Extra cooling capacity was added in Phoenix and extra heating capacity in Minneapolis (above the Manual J estimates, but in line with ACCA Manual S recommendations) to make sure that the thermostat setpoint was met for every hour of the year. For the higher-performance houses, the equipment sizing was based strictly on ACCA Manual J except for when the smallest commonly available furnace size prevailed over a smaller heating load requirement. In all cases, the blower power used for the Central Fan Integrated (CFI) supply ventilation systems and for fan cycling used for whole-house distribution of ventilation air was the same as the blower power for cooling.

Tables 2.6 Central heating and cooling system parameters including capacity, airflow, and fan power for the standard IECC houses

IECC houses (5.8 ach50, mech vent, ducts outside, programmed setpoints)

	Heating						Cooling				Fan cycling	
	AFUE	Input Capacity (kBtu/h)	Output Capacity (kBtu/h)	Fan Flow (cfm)	Temp Rise (F)	Fan Power (W)	SEER	Total Capacity (kBtu/h)	Fan Flow (cfm)	Fan Power (W)	Fan Flow (cfm)	Fan Power (W)
Houston	0.78	45	35	756	43	378	13	30	1000	500	1000	500
Phoenix	0.78	70	55	1176	43	588	13	42	1400	700	1400	700
Charlotte	0.78	45	35	756	43	378	13	24	800	400	800	400
Kansas City	0.78	50	39	840	43	420	13	30	1000	500	1000	500
Seattle	0.78	27	21	454	43	227	13	18	600	300	600	300
Minneapolis	0.78	70	55	1176	43	588	13	24	800	400	800	400

Tables 2.7 Central heating and cooling system parameters including capacity, airflow, and fan power for the higher-performance houses

Higher-perf houses (3.0 ach50, mech vent, ducts inside, .35 U+SHGC glass, constant setpoints 70/76)

	Heating						Cooling				Fan cycling	
	AFUE	Input Capacity (kBtu/h)	Output Capacity (kBtu/h)	Fan Flow (cfm)	Temp Rise (F)	Fan Power (W)	SEER	Total Capacity (kBtu/h)	Fan Flow (cfm)	Fan Power (W)	Fan Flow (cfm)	Fan Power (W)
Houston	0.78	46	36	650	51	325	13	24	800	400	800	400
Phoenix	0.78	46	36	750	44	375	13	30	1000	500	1000	500
Charlotte	0.78	46	36	650	51	325	13	24	800	400	800	400
Kansas City	0.78	46	36	650	51	325	13	24	800	400	800	400
Seattle	0.78	46	36	650	51	325	13	18	600	300	600	300
Minneapolis	0.78	51	40	800	46	400	13	18	600	300	600	300

Moisture

Because ventilation transports moisture in and out of the building in humid climates changes in indoor air moisture conditions will be estimated for all the simulations. *In the original plan this analysis was to be done for the more humid climates: Charlotte, Kansas City and Houston.*

A mass balance for moisture was used that included five separate lumped moisture capacity zones: house air, attic air, supply ducts, return ducts and moisture storage inside the house. The moisture balance included the removal of moisture due to air conditioner operation, the addition of moisture due to occupants as well as the contributions of outdoor air.

For the air zones the mass balance for moisture is based on the mass flows of air calculated using the airflow model, together with their associated moisture content (humidity ratio). For the moisture storage a mass transport coefficient and total mass storage capacity were used that were determined empirically by comparing predicted humidity variation to measured field data in BSC houses. Both coefficients scale with house size (floor area). The total mass capacity for storage was 60 kg/m^2 of floor area. The mass transport coefficient was 0.003kg/(sm^2).

The occupant generation of moisture was based on values in Table 2.8, taken from draft ASHRAE Standard 160P (Design Criteria for Moisture Control In Buildings) combined with data from NIST[6] on moisture generated by bathing, cooking and dishwashing. The researchers assumed that the exhaust fans operating in bathrooms and kitchens directly exhaust this moisture, so this moisture needs to be subtracted from the total generation rate. For example, the NIST data show that a total of 4 kg/day (for four occupants) are generated by bathing, cooking and dishwashing. Subtracted from the 13.8 kg/day calculated using Table 2.8, the result is 9.8 kg/day of moisture generation. In addition, we assumed that the house was only occupied for 2/3 of the day. The final column in Table 2.8 shows the simulated generation rates. Appendix 2 summarizes some moisture

[6] Emmerich, S., Howard-Reed, C, and Gupte, A. 2005. Modeling the IAQ Impact of HHI Interventions in Inner-city Housing. NISTR 7212. National Institute of Standards and Technology.

generation references that show that the generation rates used in this study are about on the middle of the estimates. *The original plan did not assume the 2/3 occupancy and used 50% more net generation.*

Table 2.8 Internal occupancy based moisture generation rates

	Number of Occupants	Moisture generation rate from 160P			Bathing, Cooking and Dishwashing	Net generation rate, kg/day	Simulated 2/3 Occupancy, kg/day
		L/day	Kg/s x 10^{-4}	lb/h	Kg/day		
1 bedroom	2	8	0.9	0.7	3.2	4.6	
2 bedrooms	3	12	1.4	1.1	3.6	8.5	5.7 – small
3 bedrooms	4	14	1.6	1.3	4.0	9.8	6.5 – medium
4 bedrooms	5	15	1.7	1.4	4.4	10.3	
Additional bedrooms*	+1 per bedroom	+1	+0.1	+0.1		11.3 for 5 bedrooms	7.5 – large

* per additional bedroom

Moisture removal by air conditioner operation will use estimates of latent capacity that include both steady-state and dynamic operation. The model of the coil tracks the quantity of moisture on the coil, sets an upper limit to the amount of moisture on the coil and sets condensation and evaporation rates that determine the mass fluxes to and from the coil.

The mass flux of moisture onto the coil depends on the latent capacity. REGCAP calculates total capacity and EER as functions of outdoor temperature, air handler flow and refrigerant charge. The latent capacity is calculated using the estimate of total capacity and sensible heat ratio (SHR). The SHR is based on the humidity ratio (hr) of air entering the coil. The following empirical correlation between SHR and hr was developed based on manufacturer's published information.

$$SHR_{ss} = 1 - 50(hr - 0.005)$$

Where SHR_{ss} is the steady-state SHR.

This simple, linear SHR model is illustrated graphically in Figure 2.1. The SHR is plotted here as a function of indoor RH for easier interpretation for the indoor temperatures used in the simulations. The SHR is unity up to a lower limit of about 25% RH and decreases linearly to a limit of 0.25 at about 90% indoor RH.

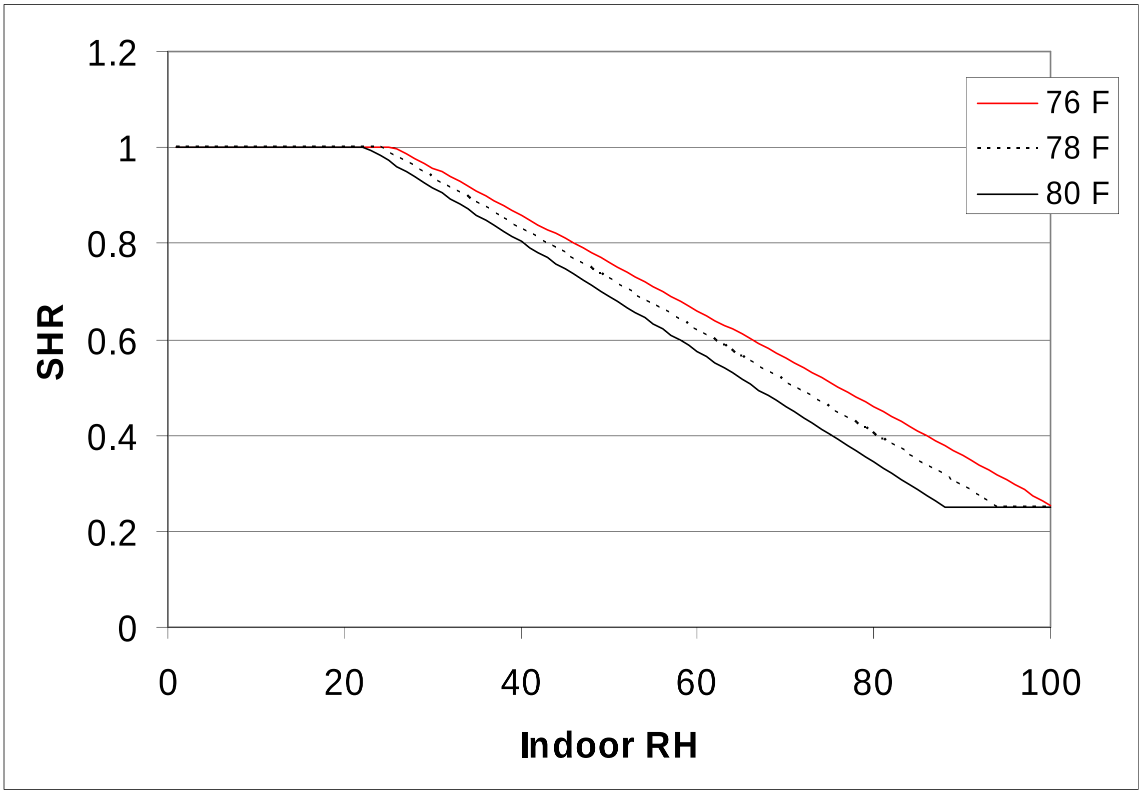


Figure 2.1 Illustration of SHR variability with indoor temperature and humidity

At the beginning of each air conditioner cycle, the system takes three minutes to ramp-up to full latent capacity. Based on work by Henderson[7] we used a linear increase from zero to full latent capacity over the first three minutes of each air conditioning cycle.

Because many of the simulations for this study will include blower fan operation with no cooling the latent model separates condensing and evaporating components with the total mass of moisture on the coil being tracked through the simulations. Condensation occurs when the cooling system operates and evaporation occurs when the coil is wet. Because the steady-state SHR is the net of condensation and evaporation, the moisture removal from the air to the coil during air conditioner operation (m_{cond}(kg/s)) is calculated as:

$$m_{cond} = \frac{(1 - SHR)Q_{total}}{2501000}$$

Where Q_{total} is the total system capacity (W) and 2501000 is the latent heat of condensation/evaporation (J/kg). This is the mass flux of moisture onto the coil and it accumulates until the coil is saturated. This accumulation has a limit, and based on the work of Henderson, the cooling coil stores about 300g/rated ton of moisture. For a three ton system, about 900g (2 lb.) of moisture can be stored on the coil. Once this quantity

[7] Henderson, H.I. and Rengarahan, K. 1996. A model to Predict the Latent Capacity of Air Conditioners and Heat Pump at Part-Load Conditions with Constant fan Operation. ASHRAE Trans, Vol. 102, Pt. 1, pp. 266-274. ASHRAE, Atlanta, GA.

Henderson, H.I. 1998. The Impact of Part-Load Air-Conditioner Operation on Dehumidification Performance: Validating a Latent Capacity Degradation Model. Proc. IAQ and Energy 1998. pp. 115-122.

of moisture is on the coil, any further mass transport of moisture to the coil leaves the system.

When the blower fan is running without air conditioning, the condensed mass on the coil is evaporated. The evaporation rate is based on Henderson's work where a coil takes 30 minutes (1800 s) to dry. The evaporation rate (m_{evap}) is then given by:

$$m_{evap} = \frac{-0.3kg / ratedton \times ratedtons}{1800s}$$

This evaporation rate is maintained until there is no moisture remaining on the coil.

Ventilation Technologies to be Simulated

System 0: Standard House (no whole-house mechanical ventilation)

This is the base case for comparison to the other ventilation methods and was simulated for all six climates and three house sizes. This is the same house as the mechanically ventilated cases (with envelope leakage as given in Table 2.1), except it had no whole-house mechanical ventilation, only bathroom and kitchen source control exhaust.

System 1: Leaky Envelope

This is the base case for existing homes and was simulated for all six climates and three house sizes. The envelope leakage for each of six climates and three houses is given in Table 2.4. This envelope leakage was calculated to produce an ASHRAE Standard 62.2 compliant infiltration rate using ASHRAE Standard 136.

System 2: Continuous exhaust

Continuous exhaust was simulated for the medium sized house in six climate zones using bathroom fans. The airflow required to meet ASHRAE Standard 62.2 for the medium house is 50 cfm (0.0236 m^3/s). A fan meeting this airflow as well as the low noise criterion in the HVI directory has a power consumption of 18.1 W [Panasonic FV-07VQ2].

A second set of continuous exhaust simulations (2a) were performed where the furnace blower operates for at least 10 minutes out of each hour to mix the air in the house. This ten minutes includes operation for heating and cooling, i.e., a system that operates for more than ten minutes to heat or cool will not have additional air handler operation. The heat from the air handler electric motor and heat exchange through exterior ducts were included in the calculations.

For the small house, a Panasonic FV-05VF1 can provide the 32.5 cfm (0.015 m^3/s) using 13.1W. For the large house, a Panasonic FV-08VQ2 can provide 85 cfm (0.04 m^3/s) using 20.5W.

System 3: Intermittent exhaust

Intermittent exhaust was simulated for the medium sized house in six climate zones using bathroom fans. The intermittent exhaust system was on for 20 hours and off for 4 hours during peak space conditioning load (3-7 p.m. for cooling and 1 – 5:00 a.m. for heating). The fan flow was increased from the continuous exhaust case to account for the intermittent operation. The daily fractional on-time is 20/24 = 83%. From ASHRAE Standard 62.2, the ventilation effectiveness is still 1.0, so the correct flow to obtain the average ASHRAE Standard 62.2 value is: 50 × 24/20 = 60 cfm. Using the nearest size greater than the minimum using specific HVI directory entries gives the following: 60 cfm (0.0283 m^3/s) and 24.3 W [Panasonic FV08-VF2].

A second set of continuous exhaust simulations (3a) were performed that ensure that the air handler operates for at least 10 minutes out of each hour to mix the air in the house. This ten-minute period includes operation for heating and cooling, i.e., a system that operates for more than ten minutes to heat or cool will not have additional air handler operation. The heat from the air handler electric motor and heat exchange through exterior ducts were included in the calculations.

System 4: Heat Recovery Ventilator (HRV) & Energy recovery Ventilator (ERV)

As in most field applications of this system, the HRV/ERV was connected to the supply and return of the central forced air duct system and the air handler fan was operated at the same time to avoid short circuiting of ventilation air. For balanced systems, the airflow rate is double the ASHRAE Standard 62.2 continuous airflow requirement (the sum of supply and exhaust). For the medium size house, the airflow rate during operation was 100 cfm (50 cfm supply and 50 cfm exhaust). Based on the recommendations of the project monitoring committee the simulations used the specifications of an actual unit having 138 cfm supply and exhaust. The resulting duty cycle to make the mechanical ventilation airflow equivalent to the other ASHRAE Standard 62.2 compliant systems was 36% (50/138=0.36). For the minute-by-minute simulations, the HRV and ERVs were operated to be on for 21 minutes then off for 39 minutes, or 35% runtime per hour.

An HRV was simulated for the medium sized house in Minneapolis, Kansas City, Seattle and Phoenix *(Originally Minneapolis and KC only)*. HVI listed recovery efficiencies were applied to the airflow through the HRV when calculating the energy use. For these simulations the Apparent Sensible Effectiveness (ASE) was used to determine the temperature of air supplied to the space ($T_{tospace}$).

$$ASE = \frac{T_{out} - T_{tospace}}{T_{out} - T_{fromspace}}$$

The following HRV was selected from the HVI directory:
[Broan Guardian HRV 100H]. At 138 cfm (0.065 m^3/s) it uses 124 W and has:
Apparent Sensible effectiveness = 70%
Sensible recovery efficiency = 62%

An ERV was simulated for the humid climates of Houston and Charlotte for the medium sized house. The ERV had a Total Recovery Efficiency (TRE) that included moisture transport during cooling and Latent Recovery (LR) when heating.

The following ERV was selected from the HVI directory:
[Broan Guardian ERV 100HC]. At 137 cfm (0.065 m^3/s) it uses 126 W and has:

Apparent Sensible effectiveness = 68% (heating)
Sensible recovery efficiency = 60% (heating)
Latent Recovery = 36% (cooling)
Total Recovery Efficiency = 45% (cooling)

System 5: Intermittent Supply with air inlet in central return

Intermittent supply was simulated for the medium sized house in all 6 climates and was provided by a central-fan-integrated-supply that used the air handler to draw air into the return and distribute it throughout the house using the heating/cooling ducts. The outside airflow rate was set at the ASHRAE Standard 62.2 continuous rate (i.e., 50 cfm (0.0236 m^3/s) for the medium house). The outside air duct connected into the central return was only open during air handler operation for heating and cooling and had a damper that closed when the air handler was off. The fan power requirements were determined based on the equipment capacity determined by Manual J load calculations and a nominal 2 cfm/W. For the Standard-Performance house, it was assumed that the central system ducts were in unconditioned space (in the attic) except for the colder climates of Minneapolis and Kansas City where half the ducts were in the basement and half were in the attic. The central system ducts were modeled as inside conditioned space for the Higher-Performance house. Heat generated by operation of the air handler and heat exchange through ducts running in unconditioned space was included in the calculations.

Another set of simulations (5a) were performed that coupled System 5 with additional control capability to assure that the air handler operated 20 minutes per hour to provide supply ventilation when there was little or no heating or cooling and to mix the air throughout the house. A damper control closed the outside air duct when the air handler was on for more than 20 minutes per hour for heating or cooling.

The ASHRAE Standard 62.2 continuous flow rate combined with fractional operating times that were less than continuous resulted in airflows for Systems 5 and 5a that did not meet the requirements of ASHRAE Standard 62.2. However, these intermittent supply systems are in wide use in houses and form a basis for comparison to central-fan-integrated-supply ventilation systems that can be coupled with exhaust systems (treated separately in Systems 6 and 7).

System 6: Intermittent Supply with air inlet in return and continuously operating exhaust.

These simulations were the same as case 5 but with an added exhaust fan. This first exhaust strategy had a continuously operating single-point exhaust (the same as case 2). The intermittent supply occurred when the forced air system was heating or cooling. The outdoor air supply rate was the same as the exhaust fan airflow, therefore, when the supply and return systems were operating simultaneously, the ventilation system was balanced.

A second set of simulations (6a) were performed that coupled System 6 with additional control capability to assure that the air handler operated for 20 minutes out of each hour when there was little or no heating or cooling. This provided balanced ventilation more often and periodically mixed air throughout the house. A damper control closed the outside air duct when the air handler was on for more than 20 minutes per hour for heating or cooling.

System 7: Intermittent Supply with air inlet in return and intermittent exhaust.

These simulations were the same as System 5 but with an added exhaust fan that only operated when the central air handler was off. The intermittent supply occurred when the forced air system was operating for heating or cooling, and the intermittent exhaust occurred when there was no heating or cooling. The supply outdoor airflow was the same as the exhaust flow.

A second set of simulations (7a) were performed that operated the air handler for 20 minutes out of each hour to provide supply ventilation when there was little or no heating or cooling and to mix the air throughout the house. A damper closed the outside air duct when the air handler was on for more than 20 minutes per hour for heating or cooling.

System 8: Continuous Supply

The continuous supply was simulated using the medium house in 6 climates. The continuous supply system used a fan to supply filtered air from outside to the duct system that then distributed the air throughout the house without using the air handler. A mixing ratio of 3:1 for indoor to supply air was used to temper the air. The supply fan was therefore sized to be four times the ASHRAE Standard 62.2 outdoor air requirements, i.e., 200 cfm for the medium sized house. A [Greentek MTF 150] provides 205 cfm at 0.4 in. water and uses 79W.

Because this supply fan will normally be an inline fan located outside the building thermal envelope an exception in ASHRAE Standard 62.2 means that it does not have to meet the low sone requirement. This is fortunate, as the inline fans in the HVI directory either do not have sone ratings or do not meet the low sone requirements in ASHRAE Standard 62.2.

A second set of continuous supply simulations (8a) were performed that operated the air handler for at least 10 minutes out of each hour to mix the air in the house.

House size

Additional simulations were performed for the small and large houses for cases 0, 1, 2 and 6a to examine the effect of house size.

Source Control Ventilation

In addition to the specific technologies that meet ASHRAE Standard 62.2, occasional intermittent operation of kitchen and bathroom fans was included for all simulations. Bathroom fans (except those that operate continuously) operated for half an hour every morning from 7:30 a.m. to 8:00 a.m. These bathroom fans were sized to meet the ASHRAE Standard 62.2 requirements for intermittent bathroom fans. From Table 5.1 in ASHRAE Standard 62.2 this was 50 cfm (25 L/s) per bathroom. For houses with multiple bathrooms the bathroom fans operated at the same time, so the 2000 ft^2 house had a total of 100 cfm (50 L/s) and the 4000 ft^2 house had a total of 150 cfm (75 L/s). Power requirements for these fans were 0.9 cfm/W based on recent California field survey data, i.e. 55W for each 50 cfm fan. Note that this is significantly more power than used by the high efficiency ventilation fans.

Similarly, all simulations had kitchen fan operation. Based on input from ASHRAE Standard 62.2 members and the ARTI project monitoring committee the kitchen fans operated for one hour per day from 5 p.m. to 6 p.m. These kitchen fans were sized to meet the ASHRAE Standard 62.2 requirements for intermittent kitchen fans. From Table 5.1 in ASHRAE Standard 62.2 this was 100 cfm (50 L/s). Unfortunately, very few of the kitchen fans in the HVI directory have power consumption information. The smallest of those that do [Ventamatic Nuvent RH160] has a flow rate of 160 cfm, and uses 99W.

The ventilation fan parameters used in the simulations are summarized in Table 2.9.

Table 2.9 Ventilation fan parameters

Fan	Model	Air flow (cfm)	Static pressure (in. w.c.)	Power (W)
Local exhaust:				
Bathroom exhaust	(used 0.9 W/cfm from a CA study)	50		55
Kitchen exhaust	Ventamatic Nuvent RH160	160		99
Whole-house				
Continuous Exhaust	Panasonic FV-07VQ2	50	0.25	18.1
Intermittent exhaust	FV08-VF2	60	0.25	24.3
Continuous Supply	Greentek MTF 150	205	0.4	79
ERV	Broan Guardian ERV 100HC	137	0.4	126
Central air handler	(used 0.5 W/cfm, indicitive of 0.5 inch w.c. external static pressure)			

Post- Processing

The ventilation rates and other house and ventilation system operating parameters generated by the simulations were used to look at the following:

- Convert minute-by-minute data into hourly sums and averages of airflow, energy use for heating and cooling, air handler power and fan power, indoor Temperature, humidity ratio, and RH.
- Annual energy use estimates separated by heating, cooling, air handler and ventilation fan.
- Air handler operation for ventilation and heating/cooling will be tracked separately.
- Temperature of delivered air for supply systems.
- Annual ventilation rate estimates.
- Look at peak days for peak ventilation load and potential savings for intermittent ventilation.
- Look for periods of low ventilation rates.
- Look at the moisture in the indoor air to see if any of the ventilation techniques may result in excess humidity.

The electric and gas utility rates shown in Table 2.10 were used in post-processing of the simulation data to establish operating cost. The rates were taken from the U.S. Dept. of Energy, Energy Information Administration data from October 2005.

Table 2.10 Electric and natural gas utility rates

Location	Electric rate ($/kW-h)	Gas therm rate ($/therm)
Houston	0.115	1.55
Phoenix	0.095	1.55
Charlotte	0.093	1.55
Kansas City	0.085	1.55
Seattle	0.069	1.55
Minneapolis	0.066	1.55
source: EIA, October 2005		

Ventilation Options not simulated

Open windows. Based on CA survey results (Price and Sherman 2006), this method of providing ventilation is not sufficiently reliable due to uncontrollable variations in occupant behavior and does not meet ASHRAE Standard 62.2 requirements except in very limited cases.

Passive vents. Although popular in Europe, these are not generally available in the US market and do not currently meet ASHRAE Standard 62.2 requirements unless specifically designed and approved on an individual basis.

Summary of moisture generation rate options

An NRCan report (R.L. Quirouette. 1983. Moisture Sources in Houses) gives 1.25L/day per person or 5L/day total for four people account for respiration and perspiration. Other activities add about 2.4 L/day for a total of 7.4 L/day (or about 16 lb/day).

A Canadian Building Digest (CBD 31 – Moisture Problems in Houses) gives a tabular breakdown for various activities (although some - like floor mopping- would seem to be rare events (or much rarer now than then)). The key one is humans that produce 0.18 kg/hour or 4.3 kg/day. This is equivalent to 17.2 kg/day for four people (or 38 lb per day).

The very first Canadian Building Digest (CBD 1) indicates 17 lb/day (plus 2lb/hour on washdays).

CBD reports are available free online from National Research Council Canada website.

Lew Harriman's ASHRAE Humidity Control Design Guide (on p. 166) gives generation rates per person from respiration only of about 0.1 kg/h/person or 9.6 kg/day for a family of 4. This is very close to the numbers we used. If everyone sits still all the time – this number could be halved. However – it still doesn't add up to the numbers used on page

178 that say they include respiration & perspiration and come up with 1.6 to 5 kg/day for 4 people.

A very thorough discussion can be found in "A search for Moisture Sources by Jeff Christian" (in Bugs, Mold and Rot II, 1993. NIBS, Washington DC. The summary is to use 5.5 L/day for people, 11.7 L/day for people + other generation and 20 L/day for a new house with a basement (concrete drying).

Christian, J.E. 1994. "Chapter 8: Moisture Sources:' Moisture Control in Buildings, ASTM Manual 18. ed. H Trechsel. West Conshocken, PA: ASTM 176—182. (Gives 14-15 lb/day (7 kg/day or $8.1x10^{-5}$ kg/s) that matches the Tenwolde summary used in ASHRAE Standard 160P for two adults.)

Moyer, N., Chasar, D., Hoak, D. and Chandra, S. 2004. Assessing Six Residential Ventilation Techniques in Hot and Humid Climates. Porc. ACEEE Summer Study, 2004. American Council for an Enery Efficient Economy, Washington, DC. (Used ~$6x10^{-5}$ kg/s. 0.4lb/hour + additional 0.4 lb/hr in evening.)

3. SUMMARY OF SIMULATION AND ANALYSIS (TASK 3)

Computer simulations were performed to analyze the typical heating, cooling, and ventilation system operating cost, and the associated effective air change rate and indoor environmental conditions. This was done using the methodologies laid out in the Simulation Plan.

A summary analysis is provided in this section of the report with discussion of the major observations. Additional plots with more detail than would appropriately fit into this section are given in Appendix A for the Standard-Performance house, and in Appendix B for the Higher-Performance house.

A recap of the Systems evaluated is given here:

- System 0 was the reference case. It was the same as Systems 2 through 8 except there was no whole-house mechanical ventilation system.
- System 1 had enough envelope leakage to meet the ASHRAE Standard 62.2 requirements without having a whole-house mechanical ventilation system.
- System 2 had constant exhaust ventilation
- System 2a was the same as System 2 but with central fan cycling for a minimum of 10 minutes per hour for whole-house ventilation distribution and thermal comfort mixing.
- System 3 had intermittent exhaust ventilation. Exhaust ventilation was stopped each day for a 4 hour period to avoid ventilating during the highest predicted space conditioning load.
- System 3a was the same as System 3 but with central fan cycling for a minimum of 10 minutes per hour for whole-house ventilation distribution and thermal comfort mixing.
- System 4 was a heat recovery ventilation (HRV) system for Phoenix, Kansas City, Seattle, and Minneapolis, or an energy recovery ventilation (ERV) system for Houston and Charlotte.
- System 5 was intermittent supply ventilation, integrated with the central air distribution system. 50 cfm of outside air was supplied whenever the air handler was on for heating or cooling.
- System 5a was the same as System 5 but with central fan cycling for a minimum of 20 minutes per hour for whole-house ventilation distribution and thermal comfort mixing.
- Systems 6 and 6a were the same as Systems 5 and 5a except that a 50 cfm exhaust fan operated continuously.
- Systems 7 and 7a were the same as Systems 6 and 6a except that the exhaust fan shut off whenever the air handler was on for heating or cooling.

In the discussion of data analysis that follows, the “(a)” graphs are for the Standard-Performance house, while the “(b)” graphs are for the Higher-Performance house specified in the Simulation Plan.

Analysis of Operating Cost, Energy Consumption, and Air Change Rate

As shown in Figures 3.1(a) and 3.1(b), all systems did not provide the same service in terms of annual average air change rate. Systems 5 and 5a provided the lowest average air exchange while System 4 provided the highest. For the same airflow rate and duty cycle, balanced ventilation yielded a higher average air exchange than supply ventilation, and supply ventilation yielded a higher average air exchange than exhaust ventilation. That can be explained in general terms by understanding that wind and stack effects act to depressurize a building most of the time. Exhaust fans increase the building depressurization further but supply fans decrease the depressurization. Balanced fans don’t change the enclosure differential pressure and airflow at all. Because of the non-linear relationship between differential pressure and airflows across the building enclosure, a supply fan changes the pressures across the enclosure less than an exhaust fan, acting more like a balanced fan. The result is that the effect on ventilation air change rate from a supply fan is somewhere between that of an exhaust fan and a true balanced system.

Neither could all of the ventilation systems modeled be expected to provide the same service in terms of ventilation air distribution. Those that operate the central air handler a minimum amount would tend to have better ventilation air distribution, especially during periods of light heating and cooling load.

For the Standard-Performance houses, ventilation systems that operated the central air handler a minimum amount for supplying ventilation air or distributing ventilation air forced an additional amount of outdoor air change due to the specified 5% duct leakage to outside. That included systems: 2a, 3a, 4, 5a, 6a, 7a, and 8a. The effect of that was less in Kansas City and Minneapolis where half of the ducts were located in a conditioned basement. On the other hand, the Higher-Performance houses had all ducts inside conditioned space, so there was no additional outdoor air change associated with central fan use.

Tables 3.0(a) and 3.0(b) show the number of annual hours where outside air exchange was below 0.34 ACH which is equivalent to the 62.2 mechanical ventilation rate plus the default infiltration credit of 0.02 cfm/ft^2. For the Standard-Performance house, System 4 (balanced HRV/ERV) had less than 0.34 ach for less than 50% of the year in all climates, and less than 30% of the year in the three colder climates. Except for the colder climates with supply ventilation, the other systems had less than 0.34 ach for 50% to 80% of the year. For the Higher-Performance house, except for balanced and some supply systems in Minneapolis, ach was less than 0.34 almost all of the time.

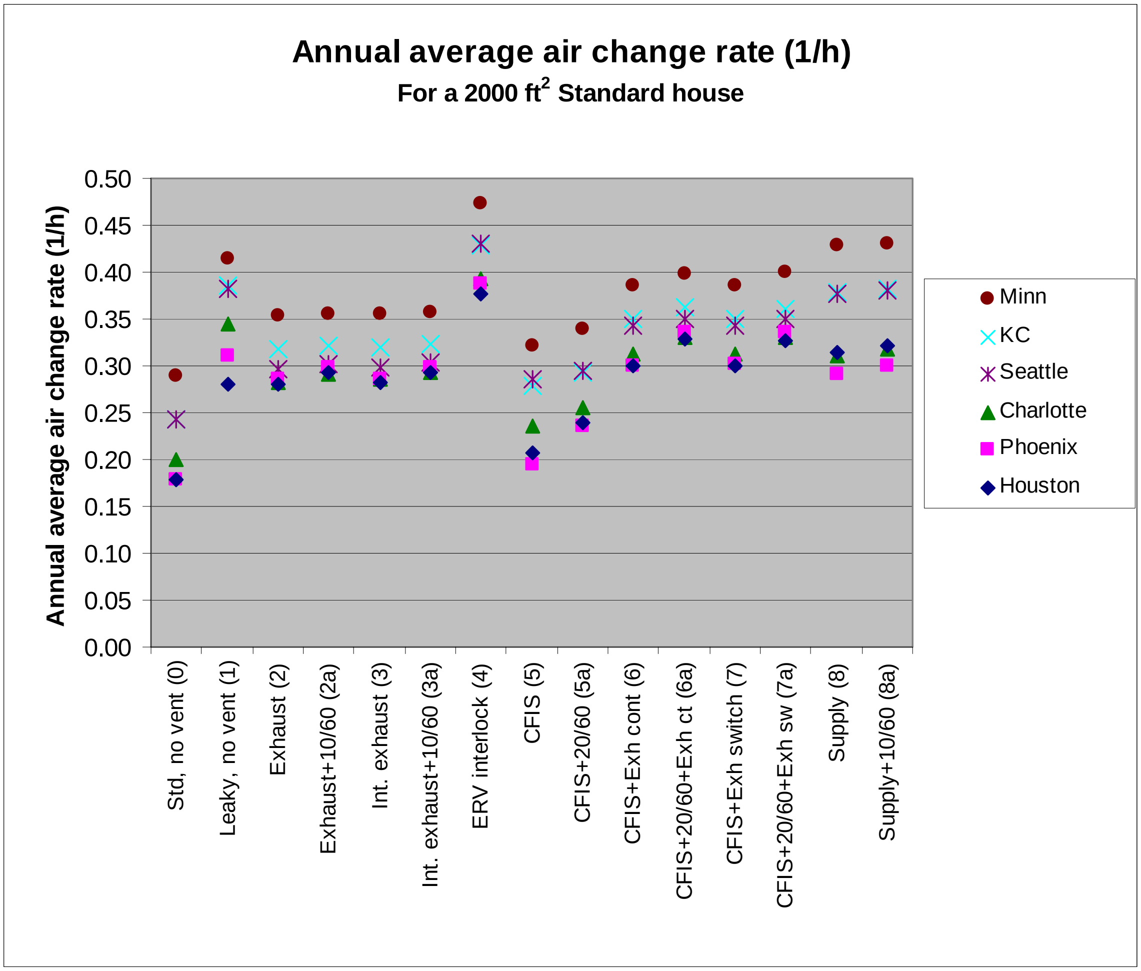


Figure 3.1(a) Annual average air change rate (ach) for Standard-Performance houses

Table 3.0(a) Number of annual hours where outside air exchange was below 0.34 ACH (equivalent to the 62.2 rate + 0.02 cfm/ft2) for Standard-Performance house

Ventilation System Number	Number of annual hours where ACH was below 0.34					
	Houston	Phoenix	Charlotte	Kansas City	Seattle	Minneapolis
0	8029	8268	7695	6965	7785	5773
1	6376	5561	4848	3944	3425	3649
2	6939	6692	6893	5703	6776	4737
2a	6886	6711	6878	5650	6718	4723
3	6929	6824	6896	5673	6746	4738
3a	6840	6822	6869	5629	6707	4698
4	3878	3115	3559	2352	993	1621
5	7309	7712	6687	5865	5806	5059
5a	7630	8025	7037	5934	6637	4801
6	6202	6214	5828	4809	4928	4165
6a	6030	6097	5895	4475	4833	3677
7	6193	6212	5816	4808	4934	4167
7a	6141	6117	5900	4491	4834	3673
8	6124	6856	5697	3649	3207	2837
8a	5910	6786	5579	3528	2990	2737

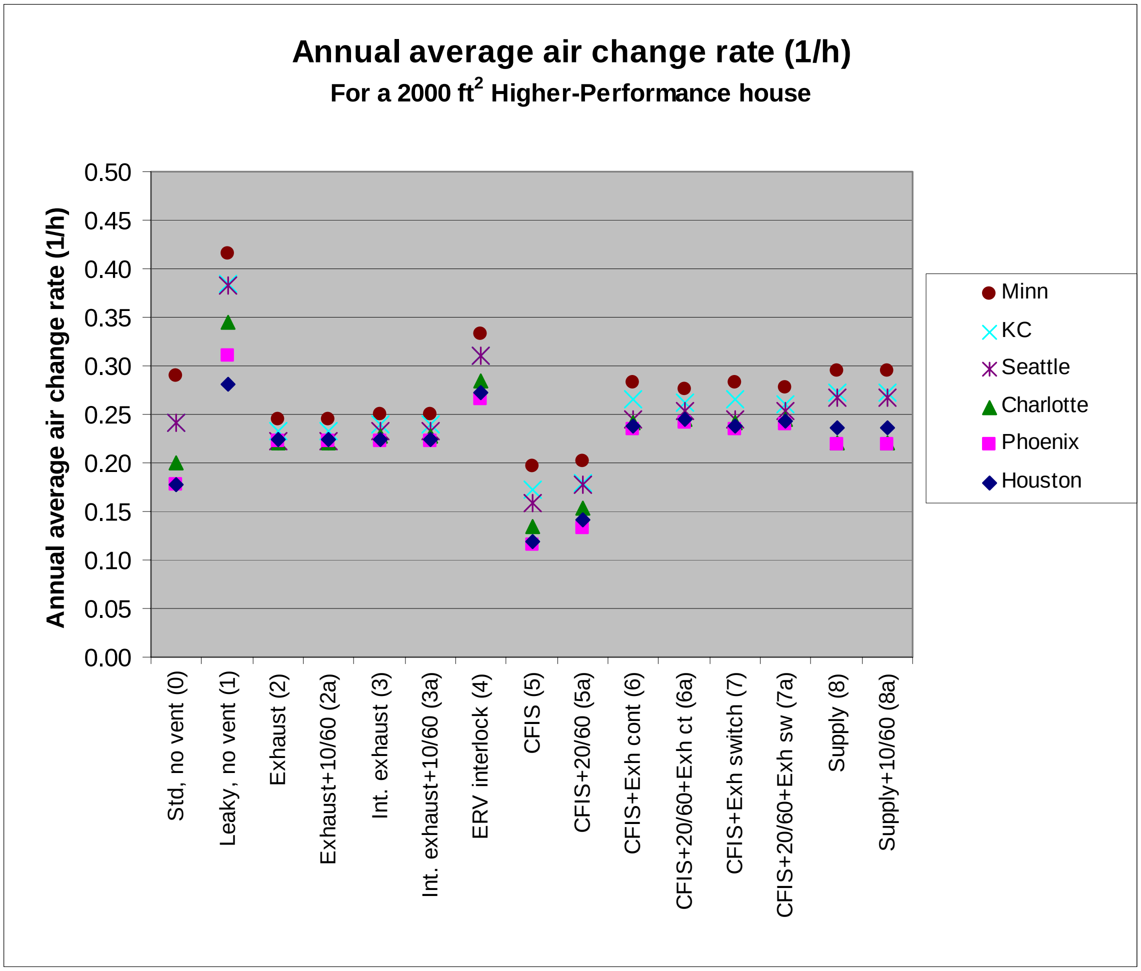


Figure 3.1(b) Annual average air change rate (ach) for Higher-Performance houses (Systems 2 through 8a)

Table 3.0(b) Number of annual hours where outside air exchange was below 0.34 ACH (equivalent to the 62.2 rate + 0.02 cfm/ft2) for Higher-Performance house

Ventilation System Number	Number of hours where ACH was below 0.34					
	Houston	Phoenix	Charlotte	Kansas City	Seattle	Minneapolis
2	8028	8028	8030	8030	8030	7958
2a	8028	8028	8030	8030	8030	7956
3	8027	8026	8030	8030	8029	7906
3a	8027	8026	8030	8030	8029	7905
4	7795	8009	7647	6504	7461	5334
5	8386	8395	8378	8187	8387	7744
5a	8395	8395	8023	8386	8392	8275
6	7990	8028	7973	7488	8020	6737
6a	8026	8027	8023	7963	8023	7506
7	7986	8028	7969	7492	8018	6724
7a	8026	8028	8023	7963	8025	7512
8	8298	8390	8310	7681	8166	6556
8a	8300	8390	8309	7689	8166	6562

Depending on the climate, minimally ASHRAE Standard 62.2 compliant exhaust-only systems provided between 0.06 to 0.11 more annual average air change in Standard-Performance houses, and between 0.05 less to 0.05 more in Higher-Performance houses compared to the Standard house without mechanical ventilation. The higher increase in ACH was in the milder climates. This is shown on Figures 3.2(a) and 3.2(b).

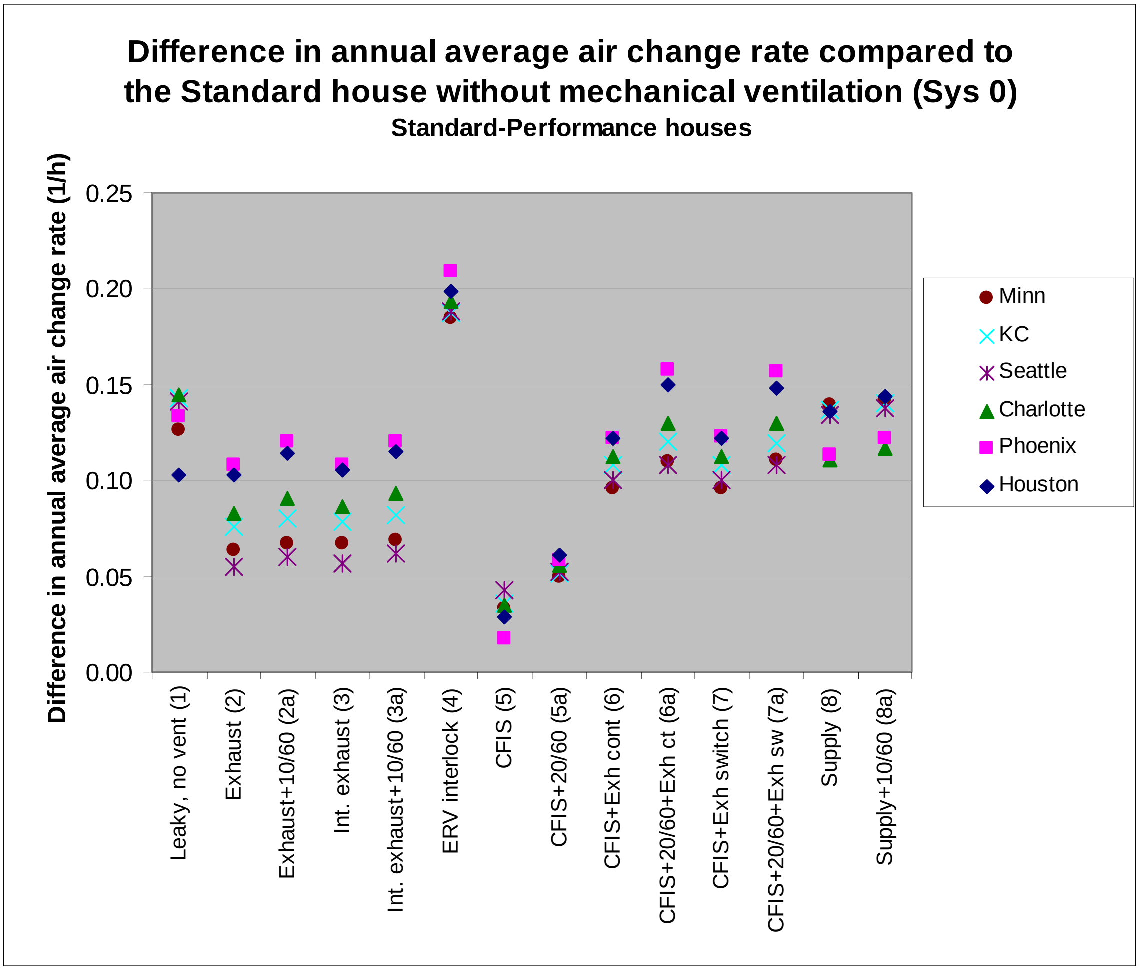


Figure 3.2(a) Difference in annual average air change rate (ach) for Standard-Performance houses compared to the Standard house without mechanical ventilation

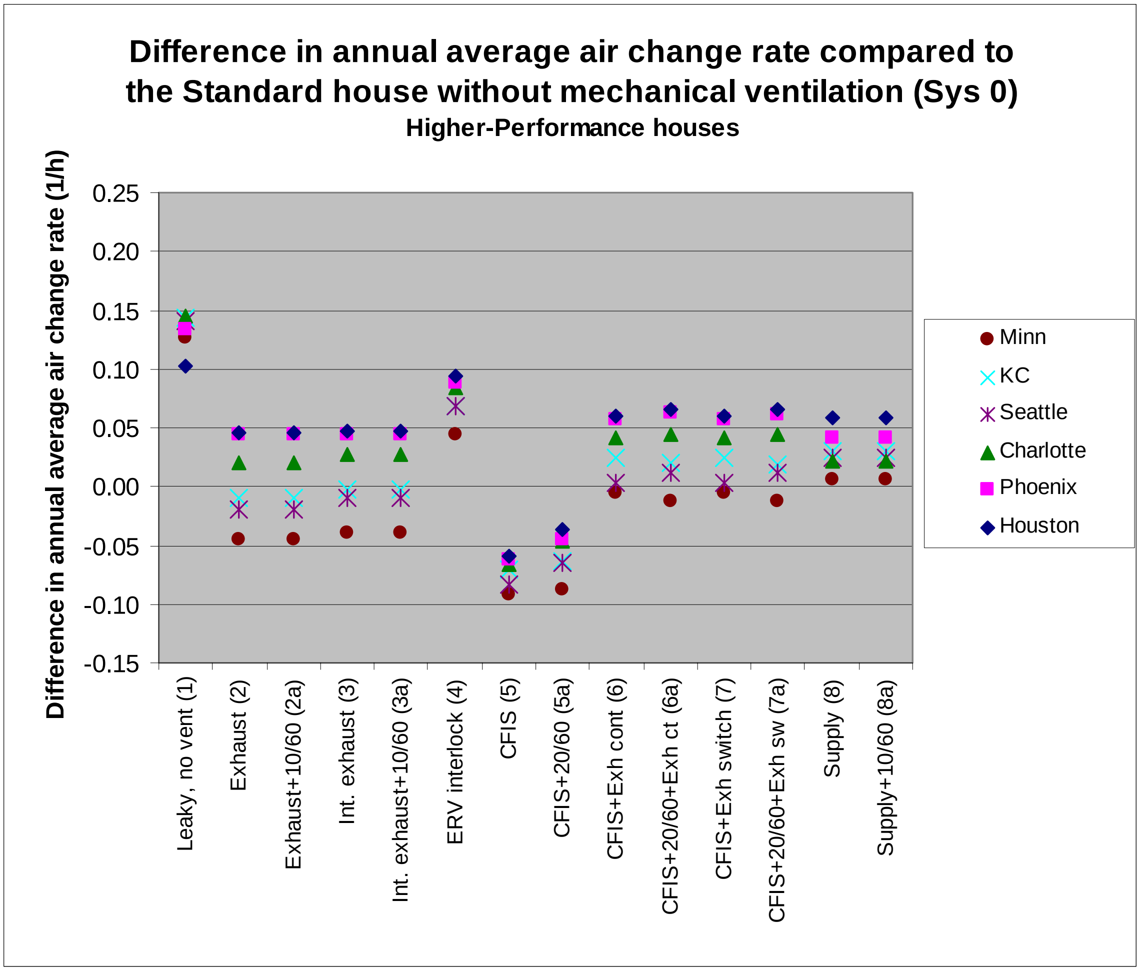


Figure 3.2(b) Difference in annual average air change rate (ach) for Higher-Performance houses (Systems 2 to 8a) compared to the Standard house without mechanical ventilation

Going down to finer resolution, the hourly average air change rate frequency distribution, plotted in bins of 0.02 ach, is shown in Figures 3.3(a) and (b) for Phoenix and in Figures 3.4(a) and (b) for Seattle. Phoenix and Houston represent climates with significant mild periods where the ach differences between the ventilation systems show up the most. Mild periods have less natural infiltration, and less space conditioning system operation which negatively impacts the CFIS ventilation system (System 5). This is much more pronounced for the Higher-Performance house because of the tighter building enclosure. Seattle and Minneapolis represent climates with dominant space conditioning seasons where the ach variation between ventilation systems is least.

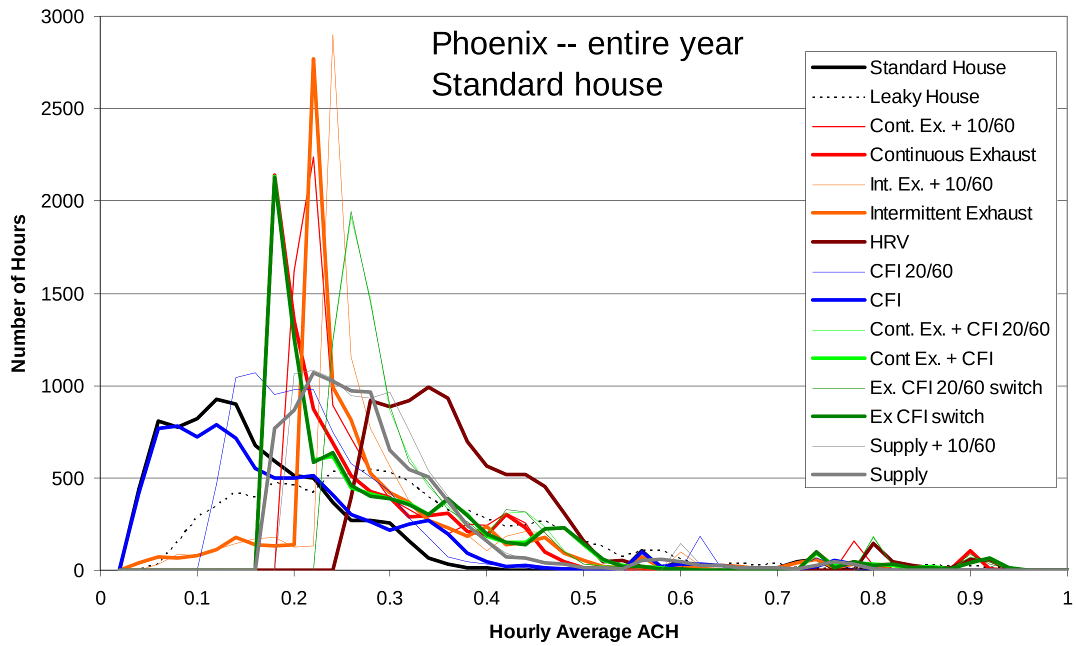


Figure 3.3(a) Hourly average air change rate (ach frequency in 0.02 bins) for Phoenix, for Standard-Performance house

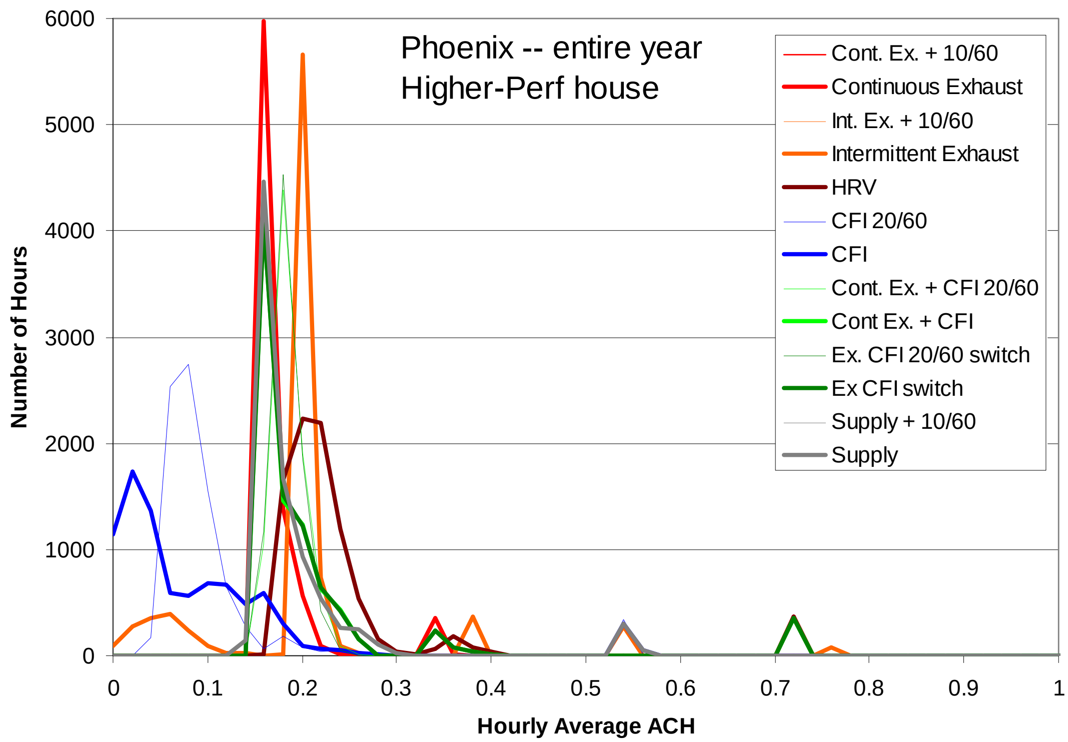


Figure 3.3(b) Hourly average air change rate (ach frequency in 0.02 bins) for Phoenix, for Higher-Performance house

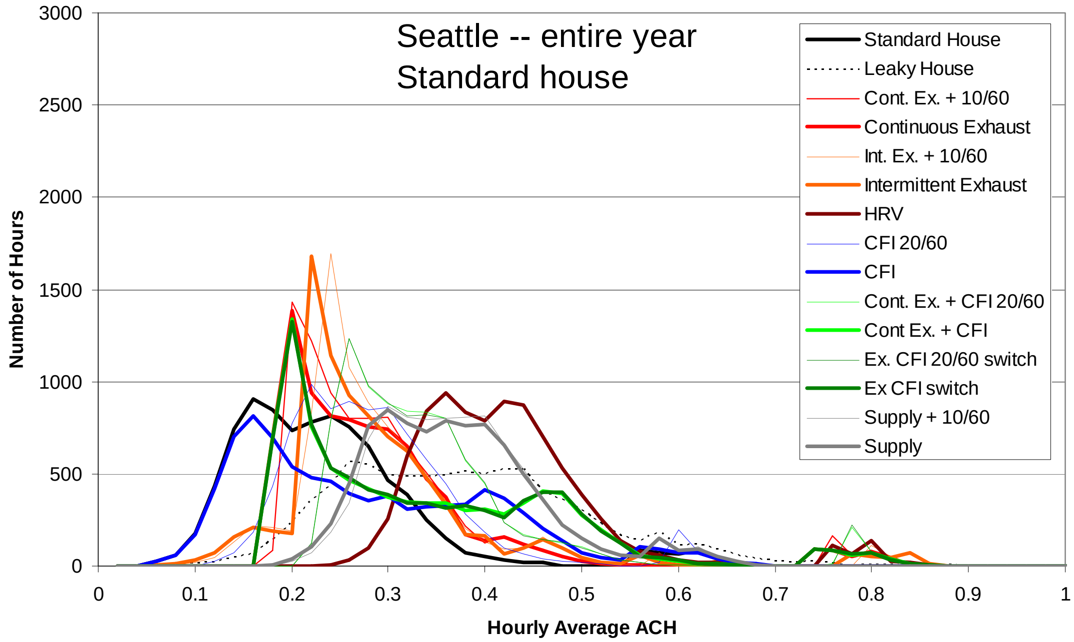


Figure 3.4(b) Hourly average air change rate (ach frequency in 0.02 bins) for Seattle, for Standard-Performance house

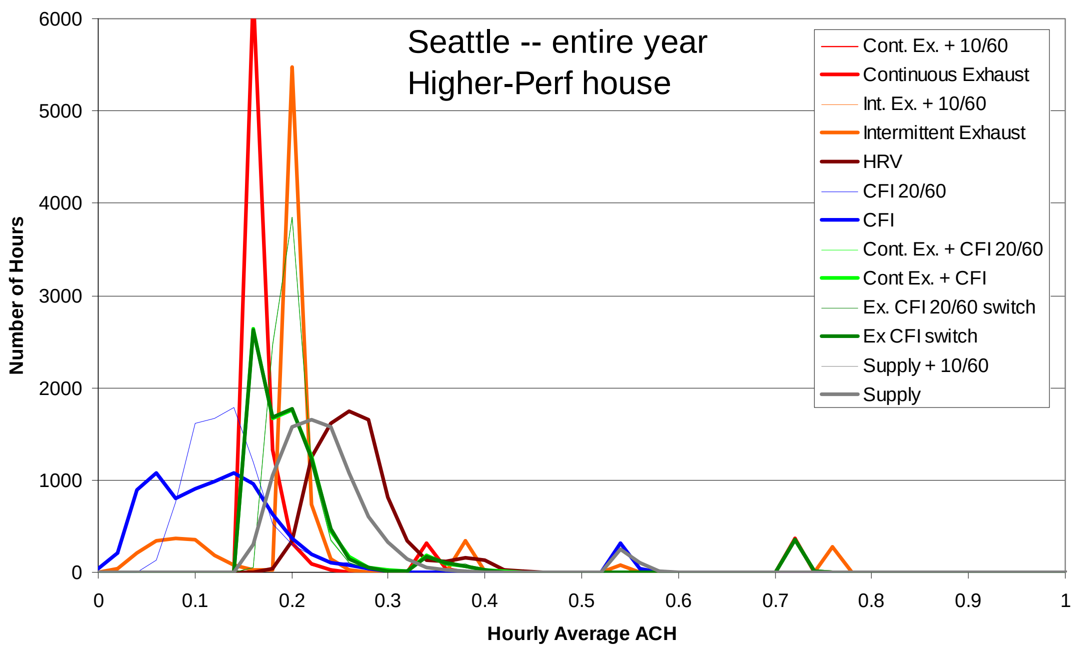


Figure 3.4(b) Hourly average air change rate (ach frequency in 0.02 bins) for Seattle, for Higher-Performance house

Tables 3.1 through 3.6 show the simulation results for each climate, broken out by:

1. natural gas therms for heating
2. heating kW-h which includes heating therms converted to kW-h and central fan electricity for heating
3. heating cost
4. cooling kW-h which includes the compressor and central fan electric consumption
5. cooling cost
6. ventilation kW-h which includes some kitchen and bath local exhaust operation and whole-house ventilation fan operation and central fan operation whenever it was used only for ventilation air distribution
7. ventilation cost
8. total kW-h including the converted heating gas therms
9. total cost
10. annual average air change rate.

Note that the intermittent supply ventilation systems 5 and 5a were modeled with less than the required ASHRAE Standard 62.2 flow rate. The outside airflow was set for the ASHRAE Standard 62.2 continuous flow rate but the fractional operating times were less than continuous. These intermittent supply systems are in wide use in houses and form a basis for comparison to other systems.

Table 3.1(a). HVAC energy-use simulation results for Houston for Standard-Performance house

Description (Sys #)	Operating cost and average air change rate for 2000 ft^2 Standard house									
	Houston									
	heat			cool		vent		total		
	(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	ach
Std, no vent (0)	427	12881	704	3609	415	56	6	16547	1125	0.18
Leaky, no vent (1)	463	13969	763	3778	435	56	6	17805	1204	0.28
Exhaust (2)	442	13312	727	4039	464	216	25	17568	1217	0.28
Exhaust+10/60 (2a)	439	13231	723	4114	473	665	77	18011	1273	0.29
Int. exhaust (3)	440	13276	725	3961	456	270	31	17508	1212	0.28
Int. exhaust+10/60 (3a)	440	13270	725	4056	466	720	83	18047	1274	0.29
ERV interlock (4)	436	13131	718	4039	464	1582	182	18752	1364	0.38
CFIS (5)	438	13216	722	3701	426	56	6	16974	1154	0.21
CFIS+20/60 (5a)	430	12975	709	3880	446	994	114	17850	1270	0.24
CFIS+Exh cont (6)	451	13599	743	4023	463	216	25	17839	1231	0.30
CFIS+20/60+Exh ct (6a)	445	13415	733	4213	484	1133	130	18761	1348	0.33
CFIS+Exh switch (7)	451	13599	743	4019	462	175	20	17794	1225	0.30
CFIS+20/60+Exh sw (7a)	445	13418	733	4222	486	1059	122	18700	1341	0.33
Supply (8)	456	13755	752	3871	445	751	86	18378	1283	0.31
Supply+10/60 (8a)	454	13695	748	3949	454	1201	138	18845	1341	0.32

Table 3.1(b). HVAC energy-use simulation results for Houston for Higher-Performance house

Description (Sys #)	Operating cost and average air change rate for 2000 ft^2 Higher-Performance house									
	Houston									
	heat			cool		vent		total		
	(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	ach
Std, no vent (0)	427	12881	704	3609	415	56	6	16547	1125	0.18
Leaky, no vent (1)	463	13969	763	3778	435	56	6	17805	1204	0.28
Exhaust (2)	389	11674	635	2177	250	216	25	14067	910	0.22
Exhaust+10/60 (2a)	384	11524	627	2230	256	592	68	14345	951	0.22
Int. exhaust (3)	390	11701	636	2126	244	270	31	14096	912	0.23
Int. exhaust+10/60 (3a)	386	11578	630	2166	249	647	74	14391	953	0.23
ERV interlock (4)	374	11215	610	2083	240	1452	167	14750	1016	0.27
CFIS (5)	382	11476	624	1868	215	56	6	13400	845	0.12
CFIS+20/60 (5a)	373	11188	608	1977	227	850	98	14014	933	0.14
CFIS+Exh cont (6)	398	11935	649	2148	247	216	25	14299	921	0.24
CFIS+20/60+Exh ct (6a)	387	11614	631	2263	260	994	114	14871	1006	0.24
CFIS+Exh switch (7)	398	11956	650	2157	248	185	21	14298	919	0.24
CFIS+20/60+Exh sw (7a)	387	11620	632	2264	260	927	107	14811	999	0.24
Supply (8)	401	12022	654	2022	233	751	86	14796	973	0.24
Supply+10/60 (8a)	396	11884	646	2060	237	1135	130	15079	1014	0.24

Table 3.2(a). HVAC energy-use simulation results for Phoenix for Standard-Performance house

Description (Sys #)	Operating cost and average air change rate for 2000 ft^2 Standard house									
	Phoenix									
	heat			cool		vent		total		
	(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	ach
Std, no vent (0)	328	9891	535	6624	629	56	5	16571	1170	0.18
Leaky, no vent (1)	365	10991	594	6903	656	56	5	17951	1256	0.31
Exhaust (2)	343	10349	560	6823	648	216	21	17389	1228	0.29
Exhaust+10/60 (2a)	347	10448	565	6846	650	734	70	18029	1285	0.30
Int. exhaust (3)	343	10331	559	6748	641	270	26	17349	1225	0.29
Int. exhaust+10/60 (3a)	347	10451	565	6777	644	784	74	18012	1284	0.30
ERV interlock (4)	328	9888	535	6886	654	2060	196	18833	1385	0.39
CFIS (5)	332	10014	542	6698	636	56	5	16769	1183	0.20
CFIS+20/60 (5a)	326	9827	531	6769	643	1237	117	17833	1292	0.24
CFIS+Exh cont (6)	347	10466	566	6885	654	216	21	17568	1241	0.30
CFIS+20/60+Exh ct (6a)	346	10424	564	6975	663	1380	131	18779	1357	0.34
CFIS+Exh switch (7)	347	10445	565	6870	653	177	17	17493	1234	0.30
CFIS+20/60+Exh sw (7a)	348	10481	567	6963	661	1312	125	18756	1353	0.34
Supply (8)	351	10566	571	6762	642	751	71	18079	1285	0.29
Supply+10/60 (8a)	348	10478	567	6804	646	1273	121	18555	1334	0.30

Table 3.2(b). HVAC energy-use simulation results for Houston for Higher-Performance house

Description (Sys #)	Operating cost and average air change rate for 2000 ft^2 Higher-Performance house									
	Phoenix									
	heat			cool		vent		total		
	(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	ach
Std, no vent (0)	328	9891	535	6624	629	56	5	16571	1170	0.18
Leaky, no vent (1)	365	10991	594	6903	656	56	5	17951	1256	0.31
Exhaust (2)	318	9563	517	4188	398	216	21	13968	935	0.22
Exhaust+10/60 (2a)	312	9398	508	4219	401	619	59	14236	968	0.22
Int. exhaust (3)	319	9617	520	4125	392	270	26	14013	937	0.22
Int. exhaust+10/60 (3a)	315	9482	512	4154	395	672	64	14308	971	0.22
ERV interlock (4)	292	8807	476	4192	398	1628	155	14628	1029	0.27
CFIS (5)	307	9235	499	4060	386	56	5	13352	890	0.12
CFIS+20/60 (5a)	296	8904	481	4124	392	925	88	13953	961	0.13
CFIS+Exh cont (6)	324	9744	527	4231	402	216	21	14191	949	0.24
CFIS+20/60+Exh ct (6a)	313	9425	509	4311	410	1067	101	14804	1020	0.24
CFIS+Exh switch (7)	323	9732	526	4231	402	177	17	14141	945	0.24
CFIS+20/60+Exh sw (7a)	312	9401	508	4309	409	999	95	14710	1012	0.24
Supply (8)	321	9666	522	4147	394	751	71	14565	988	0.22
Supply+10/60 (8a)	316	9503	514	4178	397	1155	110	14837	1020	0.22

Table 3.3(a). HVAC energy-use simulation results for Charlotte for Standard-Performance house

Description (Sys #)	Operating cost and average air change rate for 2000 ft^2 Standard house									
	Charlotte									
	heat			cool		vent		total		
	(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	ach
Std, no vent (0)	756	22787	1231	1895	176	56	5	24737	1412	0.20
Leaky, no vent (1)	843	25419	1373	1938	180	56	5	27411	1559	0.35
Exhaust (2)	782	23568	1273	2095	195	216	20	25877	1488	0.28
Exhaust+10/60 (2a)	781	23553	1272	2150	200	534	50	26235	1522	0.29
Int. exhaust (3)	781	23544	1272	2037	189	272	25	25851	1487	0.29
Int. exhaust+10/60 (3a)	780	23508	1270	2097	195	588	55	26191	1520	0.29
ERV interlock (4)	778	23438	1266	2092	195	1311	122	26839	1583	0.39
CFIS (5)	777	23426	1266	1925	179	56	5	25406	1450	0.24
CFIS+20/60 (5a)	766	23076	1247	2038	190	727	68	25840	1504	0.26
CFIS+Exh cont (6)	804	24222	1309	2071	193	216	20	26508	1521	0.31
CFIS+20/60+Exh ct (6a)	794	23930	1293	2183	203	873	81	26985	1577	0.33
CFIS+Exh switch (7)	803	24219	1308	2068	192	168	16	26454	1516	0.31
CFIS+20/60+Exh sw (7a)	793	23905	1292	2184	203	796	74	26884	1568	0.33
Supply (8)	799	24095	1302	1992	185	617	57	26703	1544	0.31
Supply+10/60 (8a)	797	24026	1298	2047	190	938	87	27009	1575	0.32

Table 3.3(b). HVAC energy-use simulation results for Charlotte for Higher-Performance house

Description (Sys #)	Operating cost and average air change rate for 2000 ft^2 Higher-Performance house									
	Charlotte									
	heat			cool		vent		total		
	(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	ach
Std, no vent (0)	756	22787	1231	1895	176	56	5	24737	1412	0.20
Leaky, no vent (1)	843	25419	1373	1938	180	56	5	27411	1559	0.35
Exhaust (2)	689	20680	1114	1066	99	216	20	21962	1233	0.22
Exhaust+10/60 (2a)	685	20545	1106	1105	103	535	50	22185	1259	0.22
Int. exhaust (3)	693	20782	1119	1028	96	273	25	22083	1240	0.23
Int. exhaust+10/60 (3a)	687	20617	1110	1056	98	590	55	22263	1263	0.23
ERV interlock (4)	670	20113	1083	1010	94	1427	133	22549	1310	0.28
CFIS (5)	682	20467	1102	905	84	56	5	21428	1191	0.13
CFIS+20/60 (5a)	667	20029	1079	977	91	739	69	21745	1238	0.15
CFIS+Exh cont (6)	708	21247	1144	1035	96	216	20	22498	1260	0.24
CFIS+20/60+Exh ct (6a)	694	20836	1122	1106	103	885	82	22827	1307	0.25
CFIS+Exh switch (7)	708	21235	1144	1030	96	180	17	22446	1256	0.24
CFIS+20/60+Exh sw (7a)	695	20848	1123	1108	103	818	76	22774	1302	0.24
Supply (8)	705	21142	1139	943	88	620	58	22705	1284	0.22
Supply+10/60 (8a)	699	20983	1130	972	90	944	88	22899	1308	0.22

Table 3.4(a). HVAC energy-use simulation results for Kansas City for Standard-Performance house

Description (Sys #)	Operating cost and average air change rate for 2000 ft^2 Standard house — Kansas City									
	heat			cool		vent		total		
	(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	ach
Std, no vent (0)	934	28149	1514	1847	157	56	5	30053	1676	0.24
Leaky, no vent (1)	1048	31579	1699	1909	162	56	5	33545	1866	0.39
Exhaust (2)	967	29155	1569	2069	176	216	18	31418	1761	0.32
Exhaust+10/60 (2a)	964	29059	1563	2086	177	558	47	31704	1788	0.32
Int. exhaust (3)	966	29108	1566	2002	170	274	23	31384	1760	0.32
Int. exhaust+10/60 (3a)	964	29065	1564	2033	173	615	52	31714	1789	0.32
ERV interlock (4)	949	28595	1538	2059	175	1565	133	32220	1847	0.43
CFIS (5)	964	29065	1564	1898	161	56	5	31020	1730	0.28
CFIS+20/60 (5a)	948	28583	1538	1973	168	791	67	31347	1773	0.29
CFIS+Exh cont (6)	997	30051	1617	2047	174	216	18	32315	1809	0.35
CFIS+20/60+Exh ct (6a)	982	29599	1592	2130	181	933	79	32662	1853	0.36
CFIS+Exh switch (7)	996	30030	1616	2049	174	168	14	32247	1804	0.35
CFIS+20/60+Exh sw (7a)	982	29593	1592	2134	181	861	73	32588	1847	0.36
Supply (8)	1011	30470	1639	1943	165	751	64	33165	1868	0.38
Supply+10/60 (8a)	1009	30407	1636	1974	168	1094	93	33475	1897	0.38

Table 3.4(b). HVAC energy-use simulation results for Kansas City for Higher-Performance house

Description (Sys #)	Operating cost and average air change rate for 2000 ft^2 Higher-Performance house — Kansas City									
	heat			cool		vent		total		
	(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	ach
Std, no vent (0)	934	28149	1514	1847	157	56	5	30053	1676	0.24
Leaky, no vent (1)	1048	31579	1699	1909	162	56	5	33545	1866	0.39
Exhaust (2)	1011	30331	1628	1235	105	216	18	31782	1751	0.23
Exhaust+10/60 (2a)	1007	30226	1622	1266	108	472	40	31964	1770	0.23
Int. exhaust (3)	1014	30442	1634	1191	101	273	23	31906	1758	0.24
Int. exhaust+10/60 (3a)	1010	30316	1627	1217	103	528	45	32062	1775	0.24
ERV interlock (4)	986	29575	1587	1213	103	1306	111	32094	1801	0.31
CFIS (5)	1011	30352	1629	1105	94	56	5	31514	1727	0.17
CFIS+20/60 (5a)	990	29713	1594	1146	97	610	52	31469	1744	0.18
CFIS+Exh cont (6)	1046	31385	1684	1221	104	216	18	32822	1806	0.27
CFIS+20/60+Exh ct (6a)	1026	30794	1652	1293	110	755	64	32841	1826	0.26
CFIS+Exh switch (7)	1046	31400	1685	1219	104	166	14	32784	1803	0.27
CFIS+20/60+Exh sw (7a)	1026	30794	1652	1290	110	680	58	32763	1820	0.26
Supply (8)	1053	31601	1696	1151	98	751	64	33503	1857	0.27
Supply+10/60 (8a)	1050	31505	1691	1178	100	1009	86	33692	1876	0.27

Table 3.5(a). HVAC energy-use simulation results for Seattle for Standard-Performance house

Description (Sys #)	Operating cost and average air change rate for 2000 ft^2 Standard house									
	Seattle									
	heat			cool		vent		total		
	(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	ach
Std, no vent (0)	730	22007	1174	256	18	56	4	22319	1196	0.24
Leaky, no vent (1)	829	24988	1333	238	16	56	4	25282	1353	0.38
Exhaust (2)	769	23176	1237	272	19	216	15	23665	1270	0.30
Exhaust+10/60 (2a)	768	23146	1235	286	20	442	30	23874	1285	0.30
Int. exhaust (3)	766	23077	1231	259	18	270	19	23606	1268	0.30
Int. exhaust+10/60 (3a)	765	23062	1230	275	19	496	34	23833	1284	0.30
ERV interlock (4)	756	22790	1216	270	19	1042	72	24103	1307	0.43
CFIS (5)	761	22953	1225	255	18	56	4	23265	1246	0.29
CFIS+20/60 (5a)	748	22555	1203	276	19	543	37	23374	1260	0.29
CFIS+Exh cont (6)	801	24138	1288	264	18	216	15	24618	1321	0.34
CFIS+20/60+Exh ct (6a)	787	23719	1265	284	20	688	47	24691	1333	0.35
CFIS+Exh switch (7)	800	24117	1287	264	18	159	11	24540	1316	0.34
CFIS+20/60+Exh sw (7a)	787	23734	1266	284	20	603	42	24621	1327	0.35
Supply (8)	814	24548	1310	234	16	751	52	25534	1378	0.38
Supply+10/60 (8a)	813	24497	1307	251	17	973	67	25721	1391	0.38

Table 3.5 (b). HVAC energy-use simulation results for Seattle for Higher-Performance house

Description (Sys #)	Operating cost and average air change rate for 2000 ft^2 Higher-Performance house									
	Seattle									
	heat			cool		vent		total		
	(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	ach
Std, no vent (0)	730	22007	1174	256	18	56	4	22319	1196	0.24
Leaky, no vent (1)	829	24988	1333	238	16	56	4	25282	1353	0.38
Exhaust (2)	821	24648	1313	94	6	216	15	24957	1335	0.22
Exhaust+10/60 (2a)	816	24479	1304	101	7	404	28	24984	1339	0.22
Int. exhaust (3)	828	24849	1324	84	6	272	19	25204	1349	0.23
Int. exhaust+10/60 (3a)	823	24702	1316	92	6	456	31	25249	1354	0.23
ERV interlock (4)	788	23648	1260	84	6	1193	82	24925	1348	0.31
CFIS (5)	802	24053	1282	79	5	56	4	24188	1291	0.16
CFIS+20/60 (5a)	793	23804	1268	85	6	495	34	24384	1308	0.18
CFIS+Exh cont (6)	841	25230	1344	90	6	216	15	25536	1365	0.25
CFIS+20/60+Exh ct (6a)	832	24957	1330	97	7	638	44	25691	1380	0.25
CFIS+Exh switch (7)	841	25233	1344	88	6	182	13	25502	1363	0.25
CFIS+20/60+Exh sw (7a)	832	24975	1331	99	7	578	40	25652	1377	0.25
Supply (8)	864	25932	1382	74	5	751	52	26757	1439	0.27
Supply+10/60 (8a)	859	25767	1373	79	5	932	64	26777	1443	0.27

Table 3.6(a). HVAC energy-use simulation results for Minneapolis for Standard-Performance house

Description (Sys #)	Operating cost and average air change rate for 2000 ft^2 Standard house — Minneapolis									
	heat			cool		vent		total		
	(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	ach
Std, no vent (0)	1310	39484	2103	794	52	56	4	40335	2159	0.29
Leaky, no vent (1)	1457	43933	2340	791	52	56	4	44780	2396	0.42
Exhaust (2)	1359	40964	2182	865	57	216	14	42045	2253	0.35
Exhaust+10/60 (2a)	1357	40904	2179	894	59	462	30	42259	2268	0.36
Int. exhaust (3)	1358	40928	2180	835	55	274	18	42037	2253	0.36
Int. exhaust+10/60 (3a)	1356	40874	2177	858	57	519	34	42251	2268	0.36
ERV interlock (4)	1335	40241	2144	880	58	1341	88	42461	2290	0.47
CFIS (5)	1346	40569	2161	811	54	56	4	41436	2218	0.32
CFIS+20/60 (5a)	1334	40211	2142	858	57	654	43	41722	2242	0.34
CFIS+Exh cont (6)	1397	42119	2244	857	57	216	14	43191	2314	0.39
CFIS+20/60+Exh ct (6a)	1384	41730	2223	908	60	799	53	43437	2336	0.40
CFIS+Exh switch (7)	1397	42125	2244	863	57	173	11	43161	2312	0.39
CFIS+20/60+Exh sw (7a)	1384	41724	2223	906	60	732	48	43362	2331	0.40
Supply (8)	1430	43098	2296	795	52	751	50	44644	2398	0.43
Supply+10/60 (8a)	1427	43011	2291	822	54	998	66	44830	2411	0.43

Table 3.6(b). HVAC energy-use simulation results for Minneapolis for Higher-Performance house

Description (Sys #)	Operating cost and average air change rate for 2000 ft^2 Higher-Performance house — Minneapolis									
	heat			cool		vent		total		
	(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	ach
Std, no vent (0)	1310	39484	2103	794	52	56	4	40335	2159	0.29
Leaky, no vent (1)	1457	43933	2340	791	52	56	4	44780	2396	0.42
Exhaust (2)	1312	39476	2102	397	26	216	14	40088	2142	0.24
Exhaust+10/60 (2a)	1309	39376	2097	410	27	387	26	40173	2149	0.24
Int. exhaust (3)	1315	39566	2107	372	25	273	18	40210	2149	0.25
Int. exhaust+10/60 (3a)	1311	39452	2101	386	26	443	29	40280	2155	0.25
ERV interlock (4)	1285	38651	2058	382	25	1182	78	40214	2161	0.33
CFIS (5)	1314	39542	2105	345	23	56	4	39942	2132	0.20
CFIS+20/60 (5a)	1290	38802	2066	375	25	429	28	39604	2119	0.20
CFIS+Exh cont (6)	1363	41001	2183	389	26	216	14	41605	2223	0.28
CFIS+20/60+Exh ct (6a)	1337	40222	2142	414	27	581	38	41216	2207	0.28
CFIS+Exh switch (7)	1362	40989	2182	388	26	163	11	41540	2219	0.28
CFIS+20/60+Exh sw (7a)	1337	40228	2142	414	27	506	33	41148	2203	0.28
Supply (8)	1374	41344	2201	335	22	751	50	42430	2273	0.30
Supply+10/60 (8a)	1372	41278	2198	355	23	922	61	42555	2282	0.30

A comparison of the annual total HVAC cost (heating, cooling, and ventilation) for all six climate locations is shown in Figure 3.5(a) and 3.5(b). The most significant effect between the Standard-Performance house and the Higher-Performance house was seen in the two hot climates of Houston and Phoenix. In those climates, primarily the low SHGC glass and moving ducts out of hot attics reduced annual HVAC cost by about $300 per year. The cost reduction in mixed-humid Charlotte was about $200 per year. In the three colder climates of Kansas City, Seattle, and Minneapolis, there was not a lot of cost difference between the Standard and Higher-Performance house.

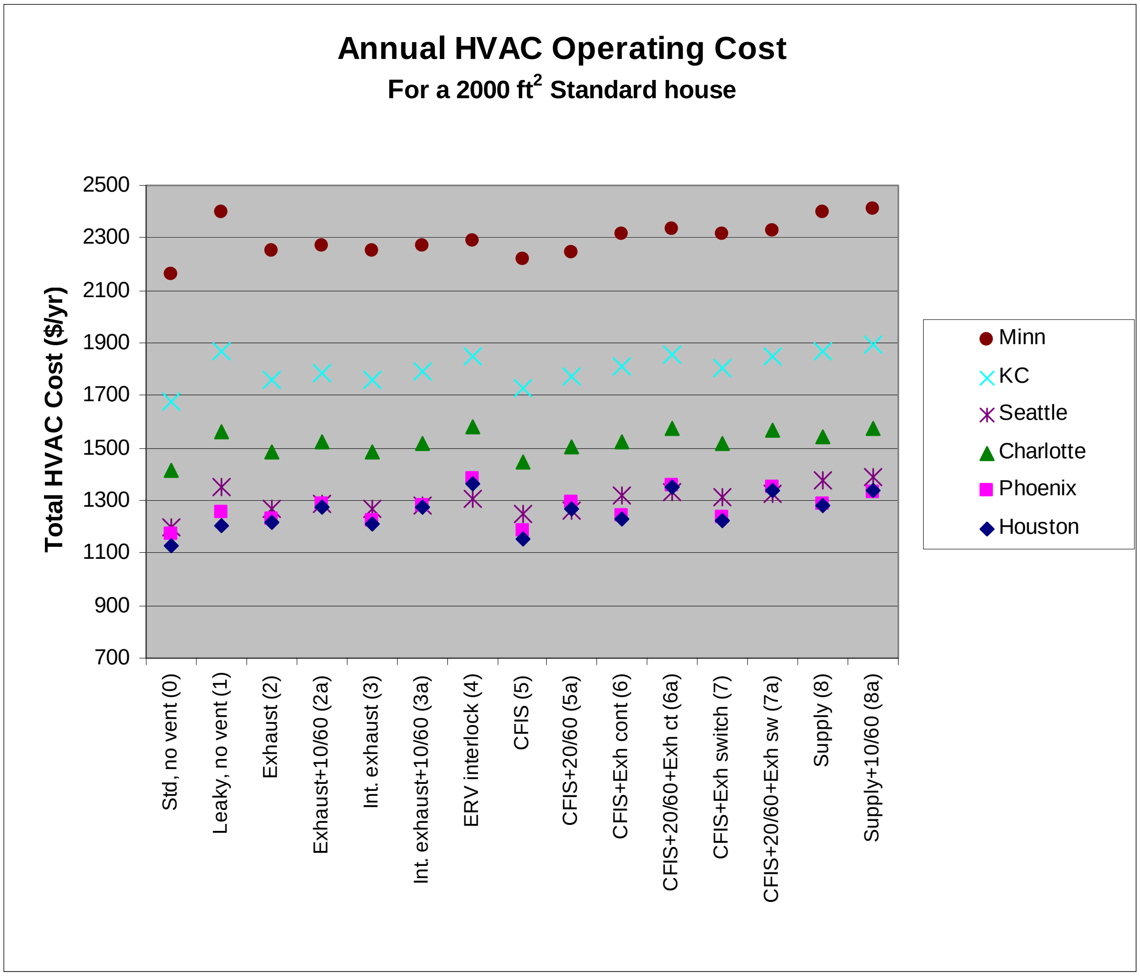


Figure 3.5(a) Comparison of annual HVAC operating cost (heating, cooling, and ventilation) for the Standard-Performance house, for all six climate locations

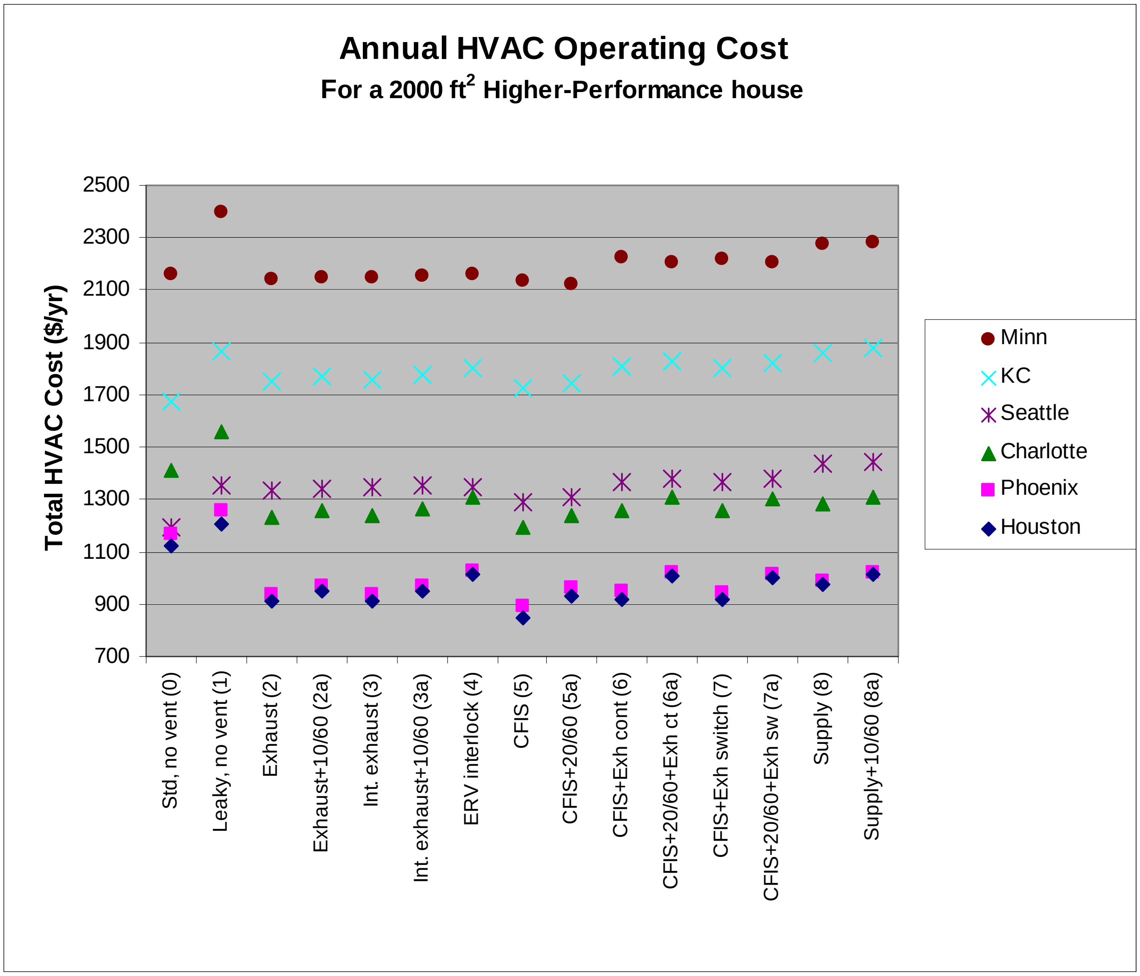


Figure 3.5(b) Comparison of annual HVAC operating cost (heating, cooling, and ventilation) for the Higher-Performance house, for all six climate locations

Contrary to what some might think, it costs far less to condition a Standard or Higher-Performance house in hot climates, even with the highest electric rates, than it does in cold climates. The cost of heating, at the minimum required efficiency of 78% AFUE, overwhelms the cost of cooling at the minimum efficiency of 13 SEER. This is true even in Houston where more could be spent for supplemental dehumidification to better control indoor humidity and the cooling plus dehumidification cost would still be less than heating.

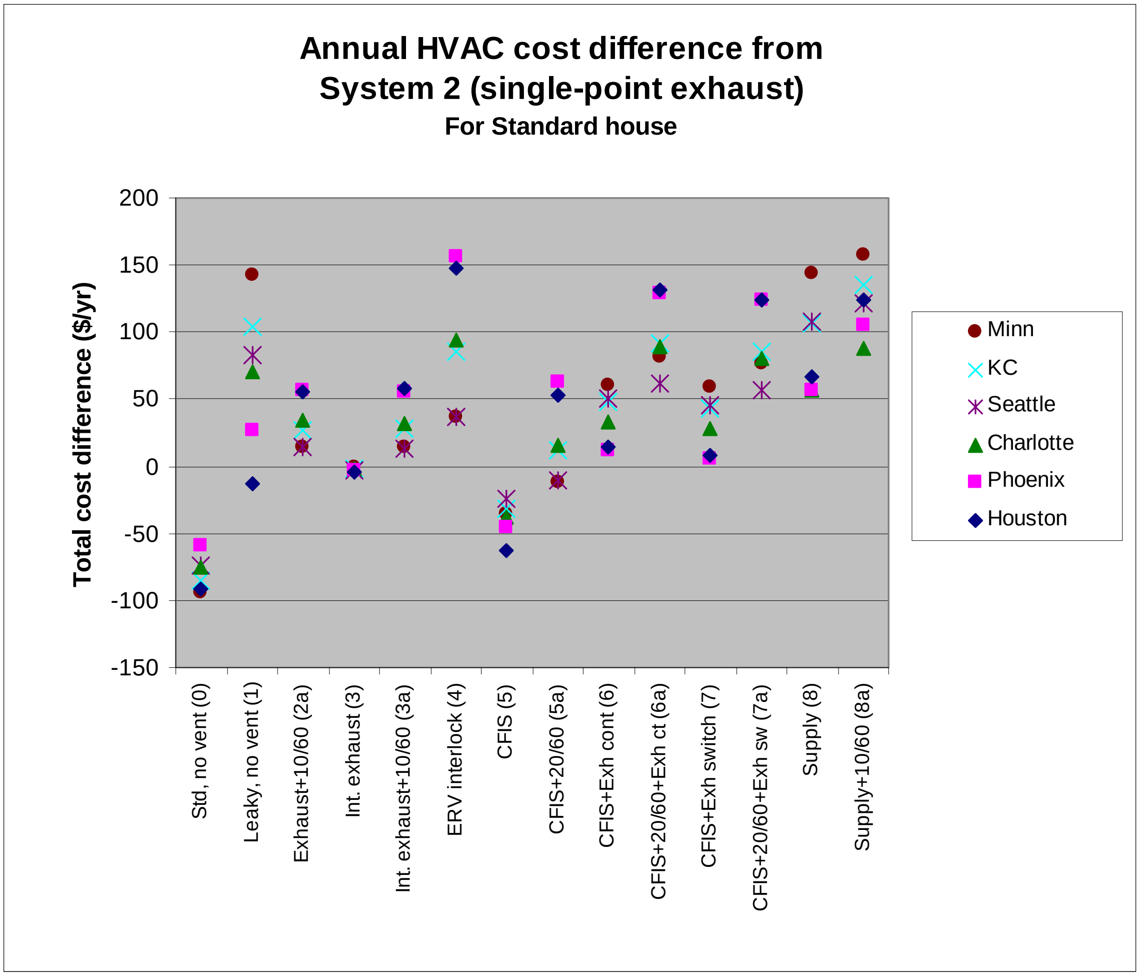


Figure 3.6(a) Total HVAC operating cost difference compared to System 2 (single-point exhaust), for all six climate locations

Referring to Figure 3.6(a), the Standard house without mechanical ventilation cost less to operate than the Standard-Performance house with System 2 (single-point exhaust), in all six climate locations. With minor exception, the Leaky house and all Systems meeting the ASHRAE Standard 62.2 ventilation rate cost from nothing to $150 more to operate than the Standard-Performance house with System 2. System 5 (central-fan-integrated supply ventilation, active only during heating and cooling operation) costs less than System 2 but it also provided from 0.05 to 0.1 annual average ach less ventilation and did not meet ASHRAE Standard 62.2 requirements.

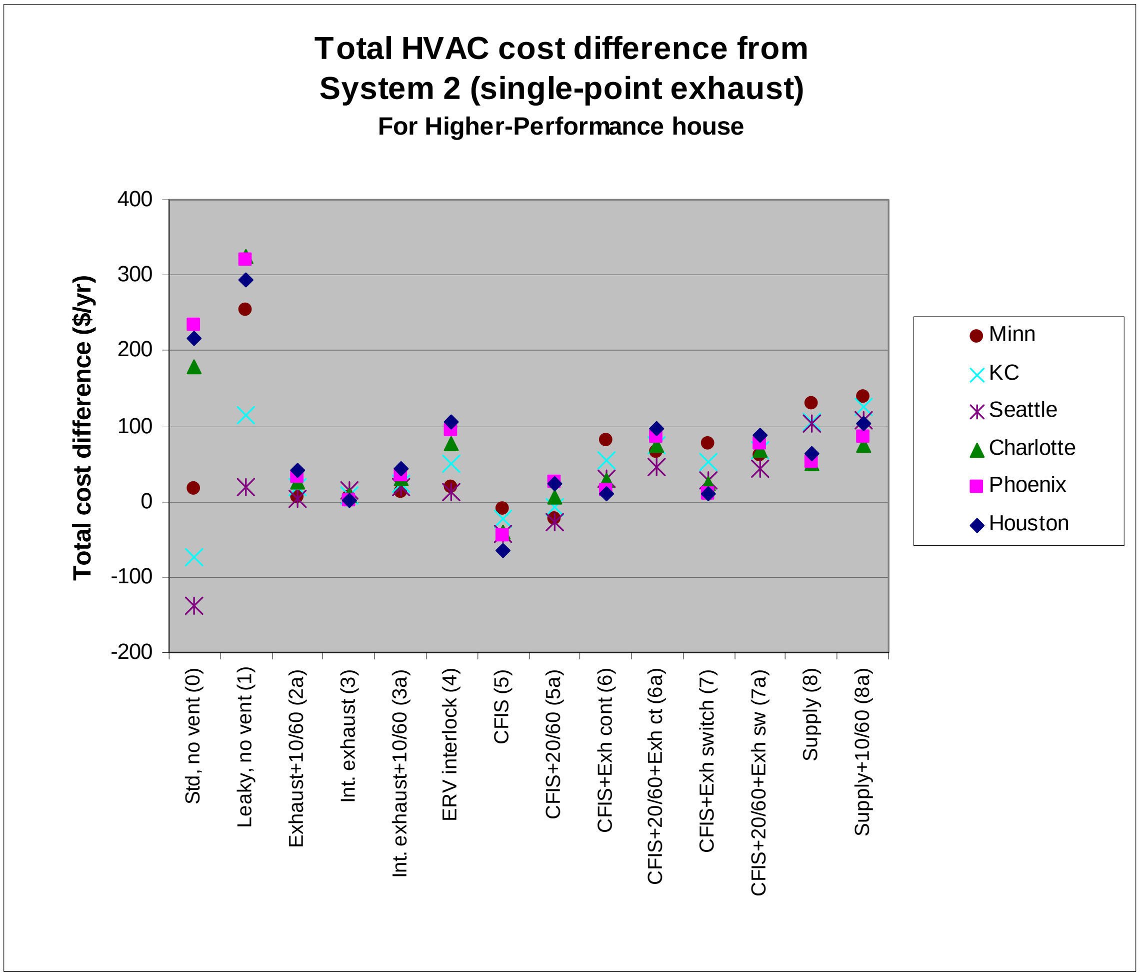


Figure 3.6(b) Total HVAC operating cost difference for the Higher-Performance house compared to System 2 (single-point exhaust), for all six climate locations

Referring to Figure 3.6(b), in Seattle and Kansas City, the Standard house without ventilation cost less to operate than the Higher-Performance house with ventilation System 2 because the low solar heat gain glass increased heating load more than it reduced cooling load in those heating dominated climates. For those locations, that effect was greater than the reduced air change rate effect (5.8 ach_{50} down to 3.0 ach_{50}). In all of the other locations, the Standard house without ventilation cost more to operate than the Higher-Performance house with ventilation System 2. With minor exception, all of the ventilation systems cost from nothing to $100 per year more to operate compared to System 2 (single-point exhaust).

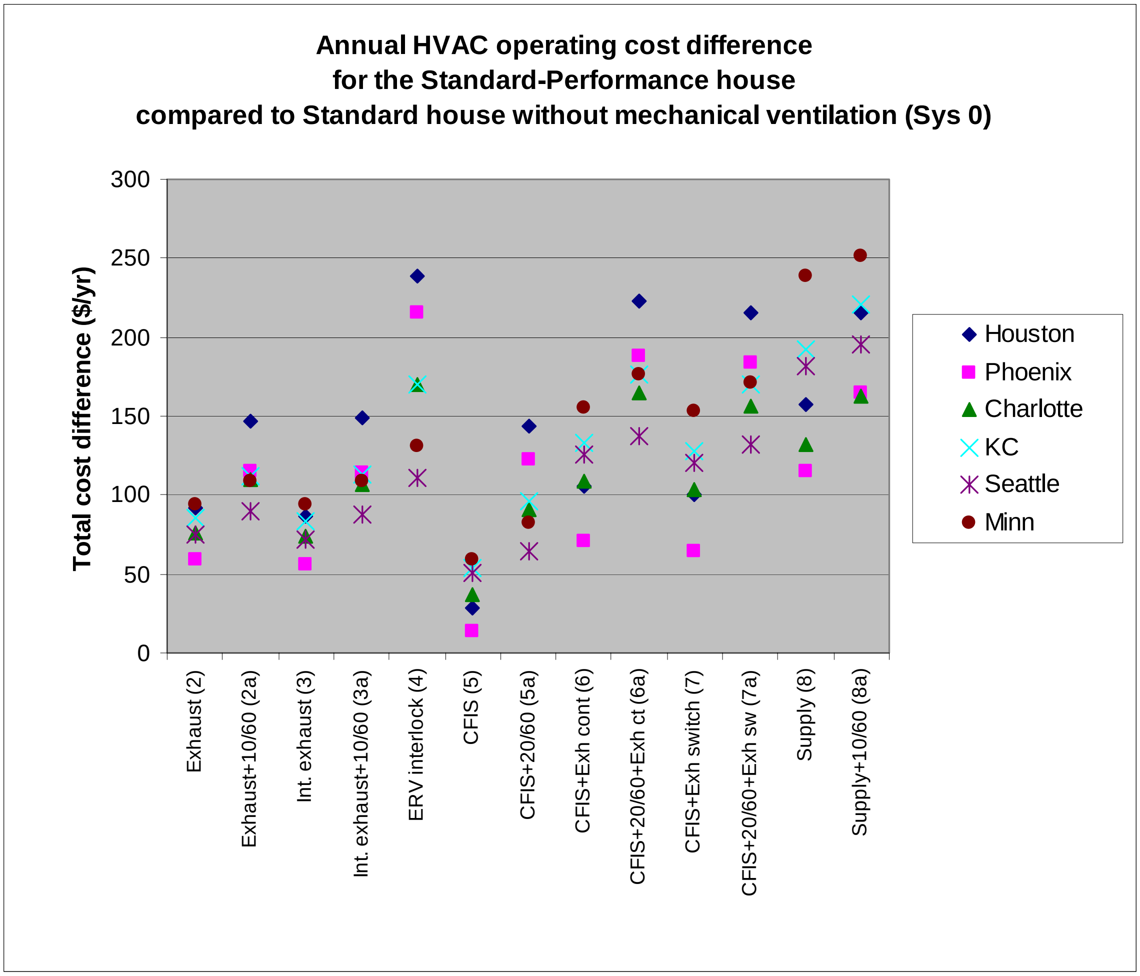


Figure 3.7(a) Annual HVAC operating cost difference compared to the Standard house without mechanical ventilation, for Standard-Performance houses in all six climate locations

Observing Figure 3.7(a), the annual HVAC operating cost for all ventilation systems that met the ASHRAE Standard 62.2 mechanical ventilation flow rate requirements ranged from $50 to $250 more than the Standard house without mechanical ventilation. The highest cost in the hot climates was System 4 because of higher fan energy consumption and the higher resulting air change rate due to a balanced system. The highest cost in the colder climates was System 8 because of higher fan energy consumption and the higher resulting air change rate due to a supply system. Of the ventilation systems that met the ASHRAE Standard 62.2 mechanical ventilation flow rate requirements, Systems 2 and 3 consistently had the lowest cost due to low fan energy consumption and lower air exchange rates.

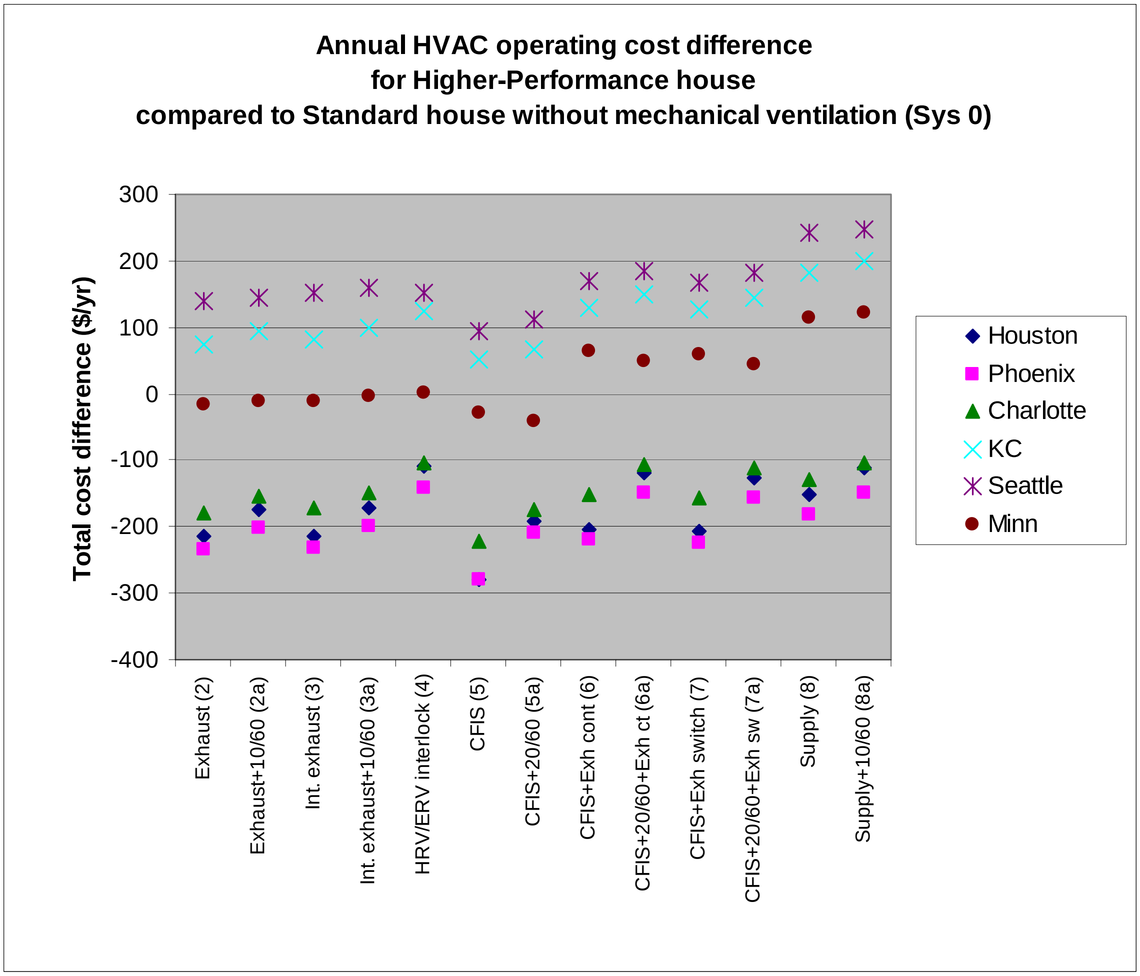


Figure 3.7(b) Annual HVAC operating cost difference for the Higher Performance house compared to the Standard house without mechanical ventilation, for all six climate locations

Looking at Figure 3.7(b), in Phoenix, Houston, and Charlotte, the Higher-Performance houses with whole-house mechanical ventilation had lower operating cost than the Standard house without mechanical ventilation. This was mainly due to the low solar heat gain glass and to locating the ducts inside conditioned space. In Minneapolis, System 5 and 5a had lower operating cost than the Standard house because of the lower air exchange rate. Systems 6 and 7 in Minneapolis showed slightly higher operating cost even though the annual average air change rate was nearly the same and half the ducts were already in a semi-conditioned basement. This was likely due to the constant heating setpoint used for the Higher-Performance house versus the programmed setback used for the Standard house. In Kansas City and Seattle, the Higher-Performance house operating cost was always higher than the Standard house because: 1) the lower SHGC glass reduced the solar gains during the heating season; 2) the constant, rather than programmed, heating setpoint has a more significant effect in heating dominated climates than in cooling dominated climates; and 3) half the ducts were already in a semi-conditioned basement for Kansas City, so the benefit of locating ducts inside conditioned space was only realized for the other half that were in the attic.

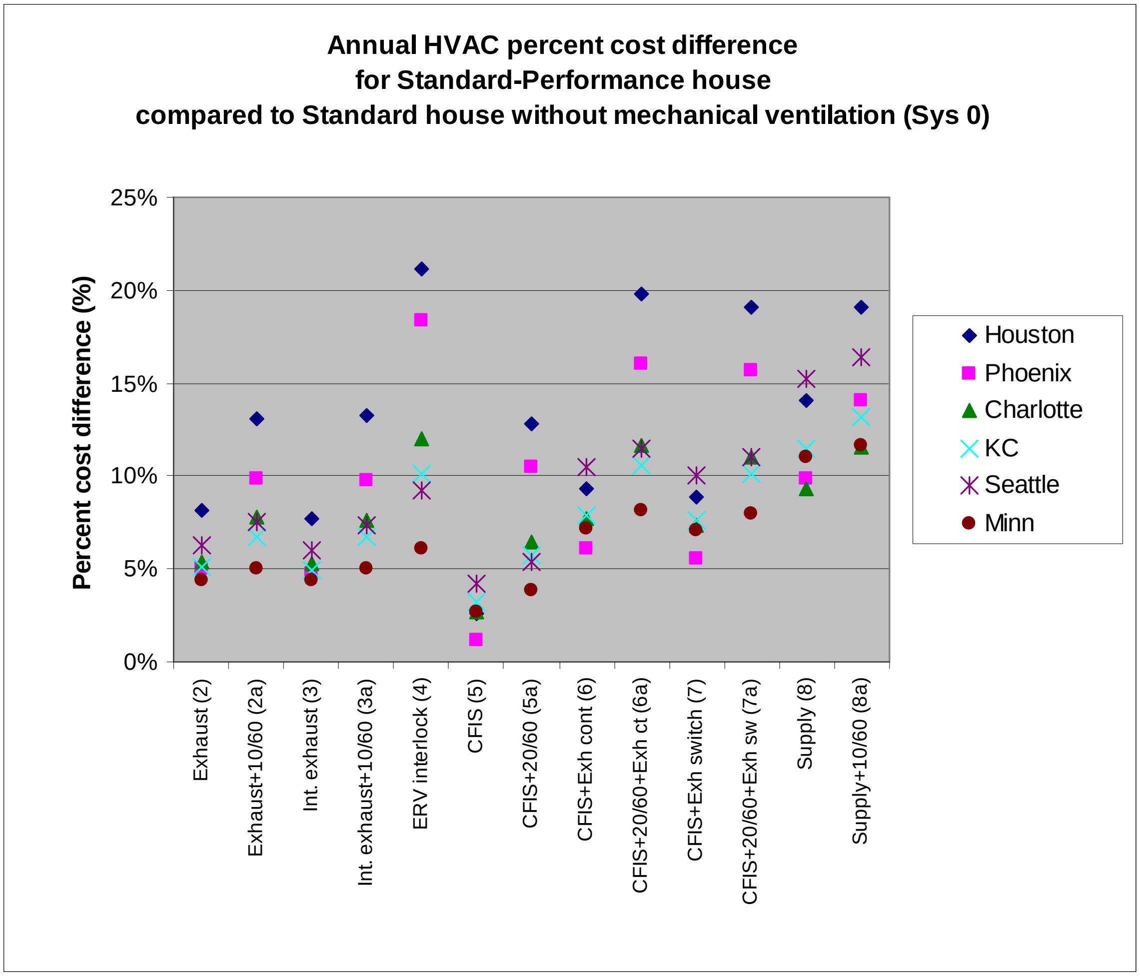


Figure 3.8(a) Annual HVAC percent cost difference for Standard-Performance house compared to the Standard house without mechanical ventilation (System 0), for all six climate locations

As shown in Figure 3.8(a), for Standard-Performance houses with mechanical ventilation compared to a Standard house without mechanical ventilation, it usually costs between 5% to 10% more to provide the mechanical ventilation. In the hot climates of Houston and Phoenix, with the highest electric utility rates, the ventilation systems that provide a minimum amount of whole-house distribution via the central fan cost 10% to 20% more than the Standard house without ventilation. Standard-Performance houses in those locations would benefit the most from ECM air handlers having one-half to one-third of the power draw of the PSC air handlers modeled here for the systems using the central air handler. Alternatively, moving to the Higher-Performance house with mechanical ventilation in those locations shows 10% to 20% savings over the Standard house without mechanical ventilation.

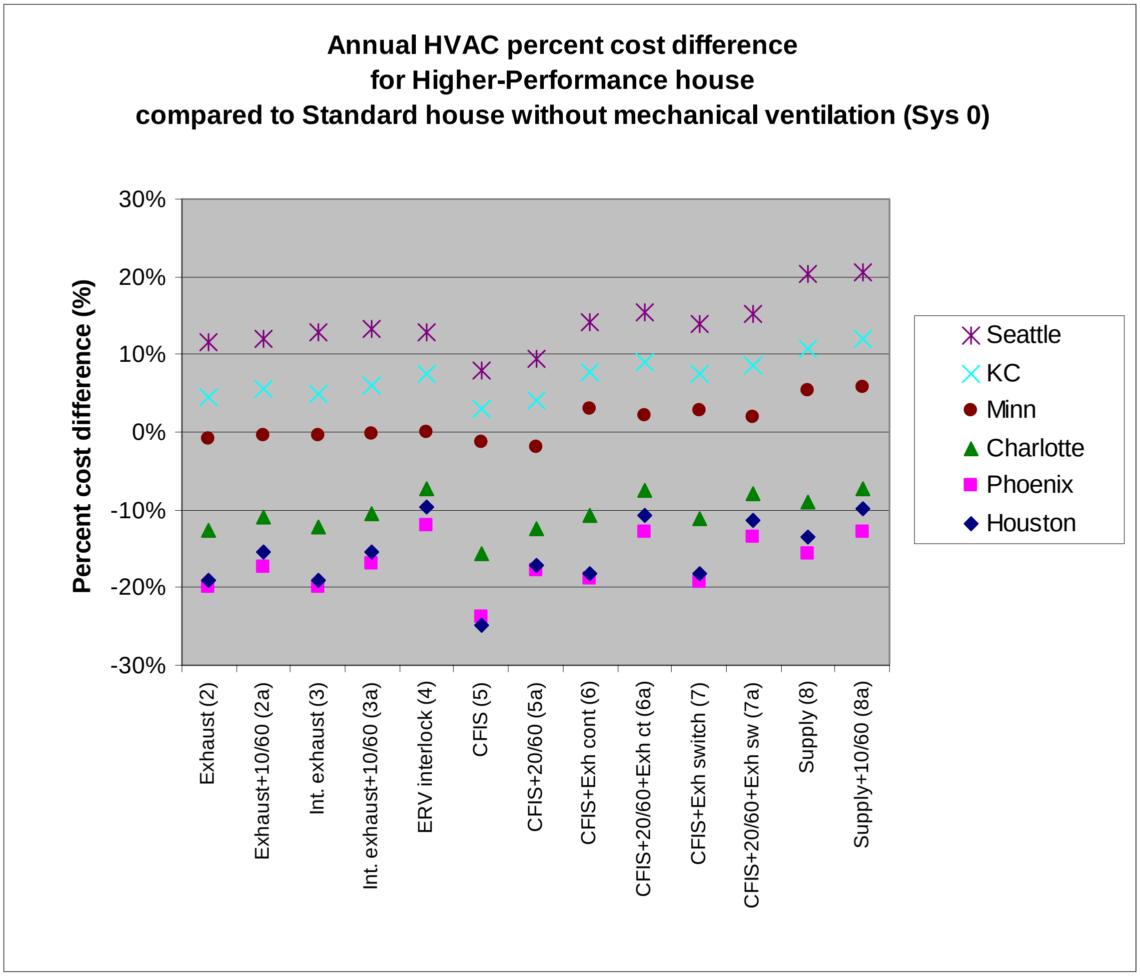


Figure 3.8(b) Annual HVAC percent cost difference for the Higher Performance house compared to the Standard house without mechanical ventilation, for all six climate locations

As shown in Figure 3.8(b), for Higher-Performance houses with mechanical ventilation compared to a Standard house without mechanical ventilation, it generally costs between 20% less to 15% more to provide mechanical ventilation. In the hot climates of Houston and Phoenix, the difference is 10% to 20% less than the Standard house without ventilation, including ventilation systems that provide a minimum amount of whole-house distribution via the central fan. In Minneapolis, the Higher-Performance house with mechanical ventilation costs nearly the same to operate as the Standard house without mechanical ventilation, regardless of the ventilation system.

Cost of Ventilation Air Distribution using the Central Air Handler Fan

Figure 3.9(a) shows the net annual cost of providing whole-house ventilation air distribution and thermal comfort mixing for the Standard-Performance house using the central system fan. When operating the central fan a minimum of 10 minutes per hour, not counting operation that would happen anyway for heating and cooling, the annual cost of central fan cycling ranges from under $20 in the colder climates with low electric rates to about $60 in the hot climates with high electric rates. When operating the central fan a minimum of 20 minutes per hour, the annual cost remained about the same in Seattle and Minneapolis, and about doubled in the other climates. Use of ECM air handler fans would reduce these costs by 50% or more as long as the duct system does not have excessive flow resistance. The dynamics of system sizing, the frequency and length of mild weather periods, the thermostat settings/program, the fan electric contribution to heating or load to cooling, the location of ducts, and duct leakage all affect the result.

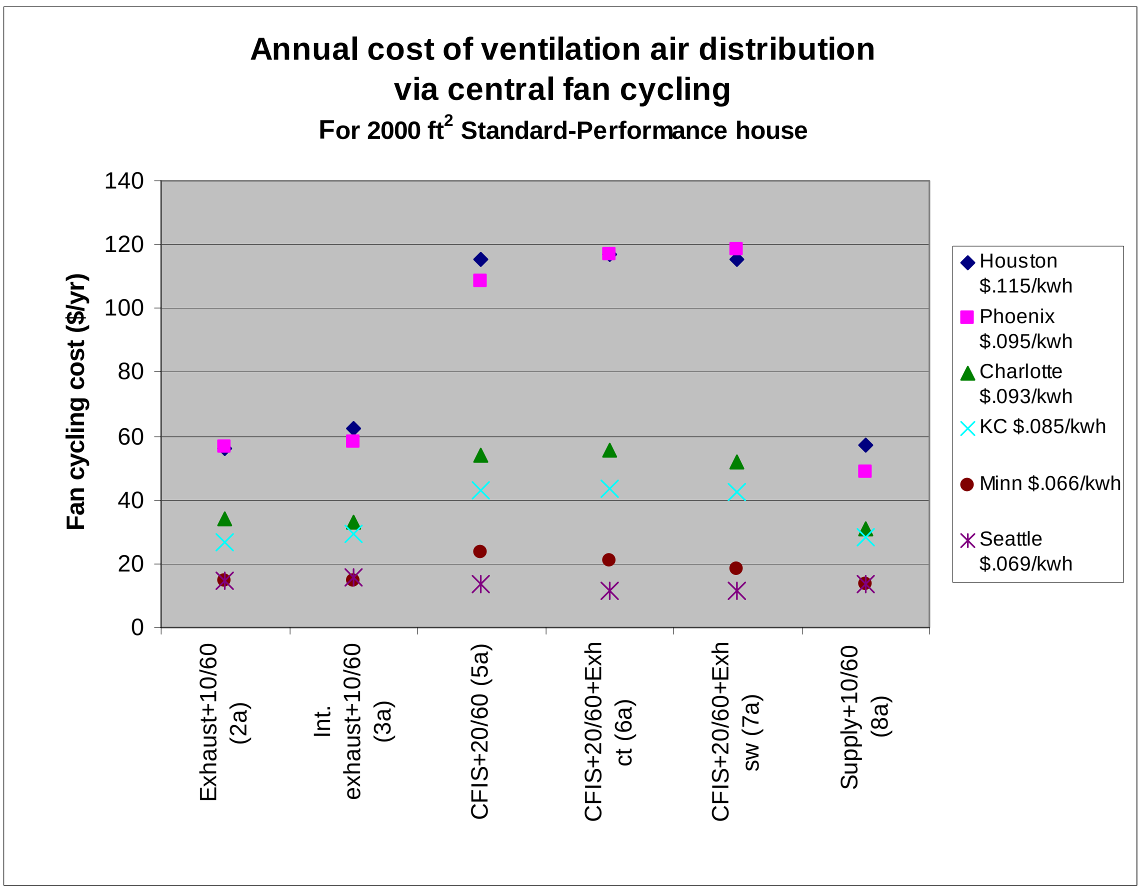


Figure 3.9(a) Annual cost of ventilation air distribution and thermal comfort mixing via the central fan for the Standard-Performance house, for all six climate locations

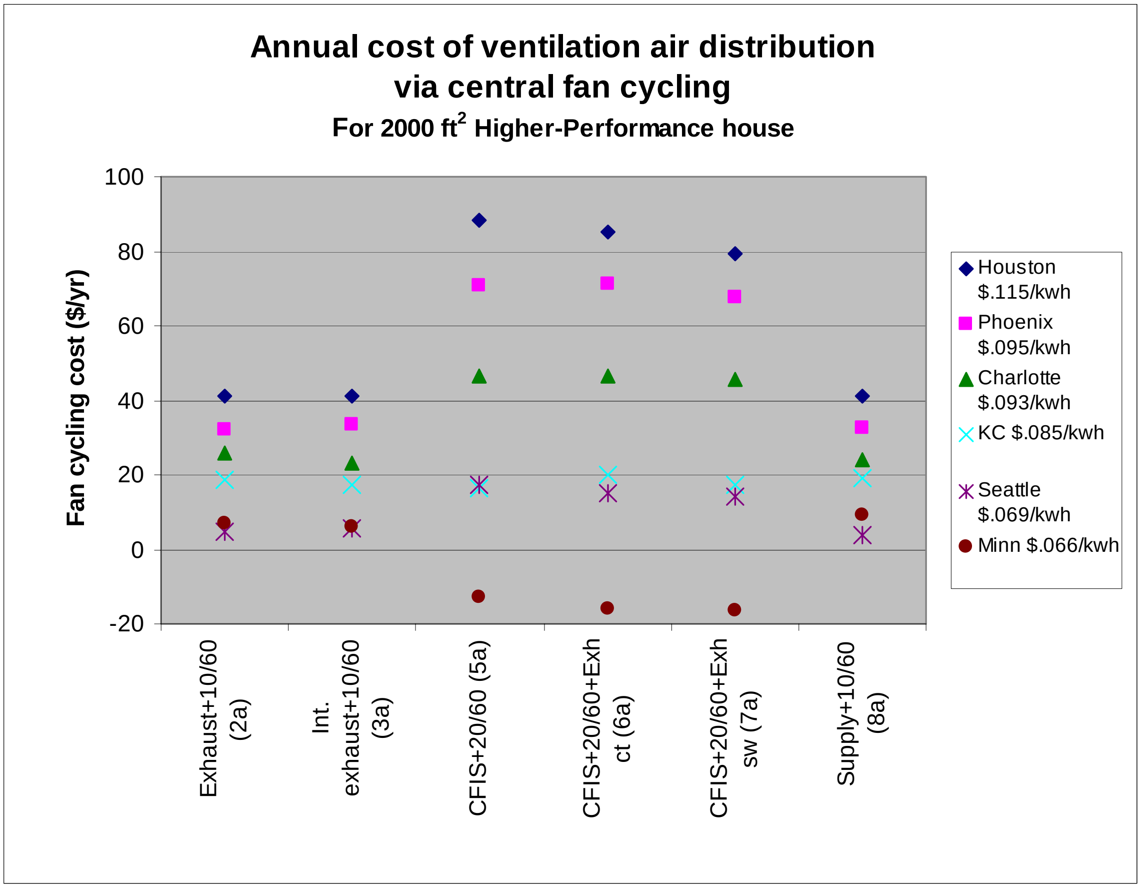


Figure 3.9(b) Annual cost of ventilation air distribution and thermal comfort mixing via the central fan for the Higher-Performance house, for all six climate locations

Figure 3.9(b) shows the net annual cost of providing whole-house ventilation air distribution and thermal comfort mixing for the Higher-Performance house. When operating the central fan a minimum of 10 minutes per hour, not counting operation that would happen anyway for heating and cooling, the annual cost of central fan cycling ranged from under $10 in the colder climates with low electric rates to about $40 in the hot climates with high electric rates. When operating the central fan a minimum of 20 minutes per hour, the annual cost remained about the same in Kansas City, and about doubled in the other climates except Minneapolis. In Minneapolis, it is interesting to note that central fan cycling at 20 minutes per hour produced savings. That is because gas heating was offset by the heat from the central fan, and at 78% efficiency, the cost of natural gas per delivered kW-h was slightly higher than the electric rate in Minneapolis.

Tables 3.7(a) and (b) through Tables 3.12(a) and (b) show the annual hours and annual runtime fractions (fraction of 8760 hours), for heating, cooling, central air handler fan use for supply ventilation and whole-house distribution, and total air handler activity.

HVAC System Hourly Runtime Analysis

Ventilation System Number	Description	Annual runtime hours and runtime fractions for heating, cooling, and AHU fan cycling for ventilation (Standard house) — Houston							
		heat		cool		ah vent/mix		total ah active	
		(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)
0	Ref, no vent, 5.8 ach50	956	0.11	1147	0.13	0	0	2103	0.24
1	Leaky, no mech vent	1036	0.12	1187	0.14	0	0	2223	0.25
2	Exhaust	988	0.11	1256	0.14	0	0	2243	0.26
2a	Exhaust+10/60	982	0.11	1273	0.15	899	0.10	3154	0.36
3	Int. exhaust	985	0.11	1232	0.14	0	0	2216	0.25
3a	Int. exhaust+10/60	984	0.11	1255	0.14	903	0.10	3142	0.36
4	ERV interlock	974	0.11	1253	0.14	2284	0.26	4511	0.51
5	CFIS	980	0.11	1163	0.13	0	0	2143	0.24
5a	CFIS+20/60	963	0.11	1217	0.14	1873	0.21	4052	0.46
6	CFIS+Exh cont	1009	0.12	1241	0.14	0	0	2250	0.26
6a	CFIS+20/60+Exh cont	995	0.11	1299	0.15	1842	0.21	4135	0.47
7	CFIS+Exh switch	1009	0.12	1240	0.14	0	0	2249	0.26
7a	CFIS+20/60+Exh switch	995	0.11	1303	0.15	1835	0.21	4133	0.47
8	Supply	1021	0.12	1206	0.14	0	0	2226	0.25
8a	Supply+10/60	1016	0.12	1227	0.14	893	0.10	3136	0.36
	average of Sys 2 to 8a	992	0.11	1243	0.14				
	average of 10/60 cycling					898	0.10	3144	0.36
	average of 20/60 cycling					1850	0.21	4107	0.47
	average of no cycling							2237	0.26

Table 3.7(a) Annual runtime hours and runtime fractions for heating, cooling, and ventilation/whole-mixing for Standard-Performance house in Houston

Ventilation System Number	Description	Annual runtime hours and runtime fractions for heating, cooling, and AHU fan cycling for ventilation (High-Perf house) — Houston							
		heat		cool		ah vent/mix		total ah active	
		(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)
0	Ref, no vent, 5.8 ach50	956	0.11	1147	0.13	0	0	2103	0.24
1	Leaky, no mech vent	1036	0.12	1187	0.14	0	0	2223	0.25
2	Exhaust	851	0.10	836	0.10	0	0	1687	0.19
2a	Exhaust+10/60	840	0.10	854	0.10	940	0.11	2634	0.30
3	Int. exhaust	853	0.10	815	0.09	0	0	1668	0.19
3a	Int. exhaust+10/60	844	0.10	829	0.09	947	0.11	2620	0.30
4	ERV interlock	817	0.09	797	0.09	2520	0.29	4134	0.47
5	CFIS	836	0.10	725	0.08	0	0	1561	0.18
5a	CFIS+20/60	815	0.09	765	0.09	1986	0.23	3566	0.41
6	CFIS+Exh cont	870	0.10	818	0.09	0	0	1687	0.19
6a	CFIS+20/60+Exh cont	846	0.10	863	0.10	1946	0.22	3655	0.42
7	CFIS+Exh switch	871	0.10	821	0.09	0	0	1692	0.19
7a	CFIS+20/60+Exh switch	847	0.10	864	0.10	1947	0.22	3658	0.42
8	Supply	876	0.10	775	0.09	0	0	1651	0.19
8a	Supply+10/60	866	0.10	787	0.09	951	0.11	2604	0.30
	average of Sys 2 to 8a	849	0.10	811	0.09				
	average of 10/60 cycling					946	0.11	2619	0.30
	average of 20/60 cycling					1960	0.22	3626	0.41
	average of no cycling							1681	0.19

Table 3.7(b) Annual runtime hours and runtime fractions for heating, cooling, and ventilation/whole-mixing for Higher-Performance house in Houston

Ventilation System Number	Description	Annual runtime hours and runtime fractions for heating, cooling, and AHU fan cycling for ventilation (Standard house)							
		Phoenix							
		heat		cool		ah vent/mix		total ah active	
		(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)
0	Ref, no vent, 5.8 ach50	472	0.05	1596	0.18	0	0	2067	0.24
1	Leaky, no mech vent	524	0.06	1659	0.19	0	0	2183	0.25
2	Exhaust	493	0.06	1638	0.19	0	0	2131	0.24
2a	Exhaust+10/60	498	0.06	1642	0.19	739	0.08	2879	0.33
3	Int. exhaust	493	0.06	1619	0.18	0	0	2111	0.24
3a	Int. exhaust+10/60	498	0.06	1624	0.19	737	0.08	2859	0.33
4	ERV interlock	471	0.05	1643	0.19	2313	0.26	4427	0.51
5	CFIS	477	0.05	1606	0.18	0	0	2083	0.24
5a	CFIS+20/60	469	0.05	1620	0.18	1686	0.19	3774	0.43
6	CFIS+Exh cont	499	0.06	1646	0.19	0	0	2145	0.24
6a	CFIS+20/60+Exh cont	497	0.06	1668	0.19	1670	0.19	3835	0.44
7	CFIS+Exh switch	498	0.06	1643	0.19	0	0	2140	0.24
7a	CFIS+20/60+Exh switch	500	0.06	1666	0.19	1669	0.19	3834	0.44
8	Supply	504	0.06	1630	0.19	0	0	2133	0.24
8a	Supply+10/60	500	0.06	1636	0.19	739	0.08	2874	0.33
	average of Sys 2 to 8a	492	0.06	1637	0.19				
	average of 10/60 cycling					739	0.08	2871	0.33
	average of 20/60 cycling					1675	0.19	3815	0.44
	average of no cycling							2129	0.24

Table 3.8(a) Annual runtime hours and runtime fractions for heating, cooling, and ventilation/whole-mixing for Standard-Performance house in Phoenix

Ventilation System Number	Description	Annual runtime hours and runtime fractions for heating, cooling, and AHU fan cycling for ventilation (High-Perf house)							
		Phoenix							
		heat		cool		ah vent/mix		total ah active	
		(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)
0	Ref, no vent, 5.8 ach50	472	0.05	1596	0.18	0	0	2067	0.24
1	Leaky, no mech vent	524	0.06	1659	0.19	0	0	2183	0.25
2	Exhaust	695	0.08	1410	0.16	0	0	2104	0.24
2a	Exhaust+10/60	683	0.08	1415	0.16	807	0.09	2904	0.33
3	Int. exhaust	698	0.08	1387	0.16	0	0	2086	0.24
3a	Int. exhaust+10/60	688	0.08	1392	0.16	807	0.09	2887	0.33
4	ERV interlock	640	0.07	1395	0.16	2387	0.27	4422	0.50
5	CFIS	671	0.08	1360	0.16	0	0	2031	0.23
5a	CFIS+20/60	647	0.07	1377	0.16	1735	0.20	3759	0.43
6	CFIS+Exh cont	708	0.08	1415	0.16	0	0	2123	0.24
6a	CFIS+20/60+Exh cont	684	0.08	1438	0.16	1707	0.19	3830	0.44
7	CFIS+Exh switch	707	0.08	1415	0.16	0	0	2122	0.24
7a	CFIS+20/60+Exh switch	683	0.08	1439	0.16	1703	0.19	3825	0.44
8	Supply	702	0.08	1397	0.16	0	0	2099	0.24
8a	Supply+10/60	690	0.08	1402	0.16	801	0.09	2893	0.33
	average of Sys 2 to 8a	684	0.08	1403	0.16				
	average of 10/60 cycling					805	0.09	2895	0.33
	average of 20/60 cycling					1715	0.20	3805	0.43
	average of no cycling							2104	0.24

Table 3.8(b) Annual runtime hours and runtime fractions for heating, cooling, and ventilation/whole-mixing for Higher-Performance house in Phoenix

Ventilation System Number	Description	Annual runtime hours and runtime fractions for heating, cooling, and AHU fan cycling for ventilation (Standard house)							
		Charlotte							
		heat		cool		ah vent/mix		total ah active	
		(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)
0	Ref, no vent, 5.8 ach50	1691	0.19	756	0.09	0	0	2447	0.28
1	Leaky, no mech vent	1886	0.22	766	0.09	0	0	2652	0.30
2	Exhaust	1749	0.20	824	0.09	0	0	2573	0.29
2a	Exhaust+10/60	1748	0.20	844	0.10	795	0.09	3387	0.39
3	Int. exhaust	1747	0.20	801	0.09	0	0	2548	0.29
3a	Int. exhaust+10/60	1744	0.20	822	0.09	797	0.09	3364	0.38
4	ERV interlock	1739	0.20	821	0.09	2168	0.25	4728	0.54
5	CFIS	1739	0.20	760	0.09	0	0	2499	0.29
5a	CFIS+20/60	1712	0.20	806	0.09	1679	0.19	4198	0.48
6	CFIS+Exh cont	1798	0.21	809	0.09	0	0	2606	0.30
6a	CFIS+20/60+Exh cont	1776	0.20	855	0.10	1645	0.19	4275	0.49
7	CFIS+Exh switch	1797	0.21	808	0.09	0	0	2605	0.30
7a	CFIS+20/60+Exh switch	1774	0.20	855	0.10	1646	0.19	4275	0.49
8	Supply	1788	0.20	784	0.09	0	0	2572	0.29
8a	Supply+10/60	1783	0.20	805	0.09	795	0.09	3383	0.39
	average of Sys 2 to 8a	1761	0.20	815	0.09				
	average of 10/60 cycling					796	0.09	3378	0.39
	average of 20/60 cycling					1657	0.19	4249	0.49
	average of no cycling							2576	0.29

Table 3.9(a) Annual runtime hours and runtime fractions for heating, cooling, and ventilation/whole-mixing for Standard-Performance house in Charlotte

Ventilation System Number	Description	Annual runtime hours and runtime fractions for heating, cooling, and AHU fan cycling for ventilation (High-Perf house)							
		Charlotte							
		heat		cool		ah vent/mix		total ah active	
		(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)
0	Ref, no vent, 5.8 ach50	1691	0.19	756	0.09	0	0	2447	0.28
1	Leaky, no mech vent	1886	0.22	766	0.09	0	0	2652	0.30
2	Exhaust	1506	0.17	413	0.05	0	0	1919	0.22
2a	Exhaust+10/60	1496	0.17	427	0.05	797	0.09	2721	0.31
3	Int. exhaust	1514	0.17	397	0.05	0	0	1911	0.22
3a	Int. exhaust+10/60	1502	0.17	407	0.05	798	0.09	2707	0.31
4	ERV interlock	1465	0.17	388	0.04	2457	0.28	4310	0.49
5	CFIS	1491	0.17	350	0.04	0	0	1841	0.21
5a	CFIS+20/60	1459	0.17	378	0.04	1710	0.20	3546	0.40
6	CFIS+Exh cont	1547	0.18	398	0.05	0	0	1945	0.22
6a	CFIS+20/60+Exh cont	1518	0.17	425	0.05	1676	0.19	3618	0.41
7	CFIS+Exh switch	1547	0.18	396	0.05	0	0	1942	0.22
7a	CFIS+20/60+Exh switch	1518	0.17	427	0.05	1672	0.19	3617	0.41
8	Supply	1540	0.18	364	0.04	0	0	1904	0.22
8a	Supply+10/60	1528	0.17	374	0.04	800	0.09	2703	0.31
	average of Sys 2 to 8a	1510	0.17	396	0.05				
	average of 10/60 cycling					799	0.09	2710	0.31
	average of 20/60 cycling					1686	0.19	3594	0.41
	average of no cycling							1925	0.22

Table 3.9(b) Annual runtime hours and runtime fractions for heating, cooling, and ventilation/whole-mixing for Higher-Performance house in Charlotte

Ventilation System Number	Description	Annual runtime hours and runtime fractions for heating, cooling, and AHU fan cycling for ventilation (Standard house)							
		Kansas City							
		heat		cool		ah vent/mix		total ah active	
		(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)
0	Ref, no vent, 5.8 ach50	1878	0.21	588	0.07	0	0	2466	0.28
1	Leaky, no mech vent	2106	0.24	602	0.07	0	0	2709	0.31
2	Exhaust	1979	0.23	661	0.08	0	0	2640	0.30
2a	Exhaust+10/60	1938	0.22	652	0.07	684	0.08	3274	0.37
3	Int. exhaust	1942	0.22	628	0.07	0	0	2569	0.29
3a	Int. exhaust+10/60	1939	0.22	636	0.07	687	0.08	3261	0.37
4	ERV interlock	1908	0.22	641	0.07	2260	0.26	4809	0.55
5	CFIS	1939	0.22	598	0.07	0	0	2537	0.29
5a	CFIS+20/60	1907	0.22	622	0.07	1468	0.17	3996	0.46
6	CFIS+Exh cont	2005	0.23	637	0.07	0	0	2642	0.30
6a	CFIS+20/60+Exh cont	1974	0.23	664	0.08	1436	0.16	4074	0.47
7	CFIS+Exh switch	2003	0.23	638	0.07	0	0	2641	0.30
7a	CFIS+20/60+Exh switch	1974	0.23	665	0.08	1436	0.16	4075	0.47
8	Supply	2032	0.23	610	0.07	0	0	2642	0.30
8a	Supply+10/60	2028	0.23	618	0.07	681	0.08	3327	0.38
	average of Sys 2 to 8a	1967	0.22	636	0.07				
	average of 10/60 cycling					684	0.08	3288	0.38
	average of 20/60 cycling					1447	0.17	4048	0.46
	average of no cycling							2617	0.30

Table 3.10(a) Annual runtime hours and runtime fractions for heating, cooling, and ventilation/whole-mixing for Standard-Performance house in Kansas City

Ventilation System Number	Description	Annual runtime hours and runtime fractions for heating, cooling, and AHU fan cycling for ventilation (High-Perf house)							
		Kansas City							
		heat		cool		ah vent/mix		total ah active	
		(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)
0	Ref, no vent, 5.8 ach50	1878	0.21	588	0.07	0	0	2466	0.28
1	Leaky, no mech vent	2106	0.24	602	0.07	0	0	2709	0.31
2	Exhaust	2209	0.25	481	0.05	0	0	2690	0.31
2a	Exhaust+10/60	2201	0.25	492	0.06	639	0.07	3333	0.38
3	Int. exhaust	2217	0.25	464	0.05	0	0	2680	0.31
3a	Int. exhaust+10/60	2208	0.25	473	0.05	642	0.07	3323	0.38
4	ERV interlock	2154	0.25	467	0.05	2171	0.25	4792	0.55
5	CFIS	2210	0.25	431	0.05	0	0	2642	0.30
5a	CFIS+20/60	2164	0.25	448	0.05	1387	0.16	3998	0.46
6	CFIS+Exh cont	2286	0.26	472	0.05	0	0	2757	0.31
6a	CFIS+20/60+Exh cont	2242	0.26	501	0.06	1349	0.15	4092	0.47
7	CFIS+Exh switch	2287	0.26	471	0.05	0	0	2758	0.31
7a	CFIS+20/60+Exh switch	2242	0.26	500	0.06	1347	0.15	4089	0.47
8	Supply	2302	0.26	447	0.05	0	0	2749	0.31
8a	Supply+10/60	2295	0.26	457	0.05	638	0.07	3390	0.39
	average of Sys 2 to 8a	2232	0.25	469	0.05				
	average of 10/60 cycling					640	0.07	3349	0.38
	average of 20/60 cycling					1361	0.16	4060	0.46
	average of no cycling							2709	0.31

Table 3.10(b) Annual runtime hours and runtime fractions for heating, cooling, and ventilation/whole-mixing for Higher-Performance house in Kansas City

Ventilation System Number	Description	Annual runtime hours and runtime fractions for heating, cooling, and AHU fan cycling for ventilation (Standard house)							
		Seattle							
		heat		cool		ah vent/mix		total ah active	
		(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)
0	Ref, no vent, 5.8 ach50	2721	0.31	138	0.02	0	0	2859	0.33
1	Leaky, no mech vent	3091	0.35	129	0.01	0	0	3220	0.37
2	Exhaust	2866	0.33	148	0.02	0	0	3014	0.34
2a	Exhaust+10/60	2862	0.33	156	0.02	752	0.09	3770	0.43
3	Int. exhaust	2853	0.33	141	0.02	0	0	2994	0.34
3a	Int. exhaust+10/60	2852	0.33	150	0.02	760	0.09	3761	0.43
4	ERV interlock	2818	0.32	148	0.02	2014	0.23	4981	0.57
5	CFIS	2838	0.32	138	0.02	0	0	2976	0.34
5a	CFIS+20/60	2789	0.32	150	0.02	1621	0.19	4560	0.52
6	CFIS+Exh cont	2985	0.34	144	0.02	0	0	3129	0.36
6a	CFIS+20/60+Exh cont	2933	0.33	155	0.02	1571	0.18	4659	0.53
7	CFIS+Exh switch	2983	0.34	143	0.02	0	0	3126	0.36
7a	CFIS+20/60+Exh switch	2935	0.34	155	0.02	1570	0.18	4661	0.53
8	Supply	3036	0.35	128	0.01	0	0	3164	0.36
8a	Supply+10/60	3030	0.35	137	0.02	733	0.08	3900	0.45
	average of Sys 2 to 8a	2906	0.33	146	0.02				
	average of 10/60 cycling					748	0.09	3811	0.43
	average of 20/60 cycling					1588	0.18	4627	0.53
	average of no cycling							3046	0.35

Table 3.11(a) Annual runtime hours and runtime fractions for heating, cooling, and ventilation/whole-mixing for Standard-Performance house in Seattle

Ventilation System Number	Description	Annual runtime hours and runtime fractions for heating, cooling, and AHU fan cycling for ventilation (High-Perf house)							
		Seattle							
		heat		cool		ah vent/mix		total ah active	
		(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)
0	Ref, no vent, 5.8 ach50	2721	0.31	138	0.02	0	0	2859	0.33
1	Leaky, no mech vent	3091	0.35	129	0.01	0	0	3220	0.37
2	Exhaust	1795	0.20	50	0.01	0	0	1845	0.21
2a	Exhaust+10/60	1783	0.20	54	0.01	627	0.07	2463	0.28
3	Int. exhaust	1810	0.21	45	0.01	0	0	1855	0.21
3a	Int. exhaust+10/60	1799	0.21	50	0.01	620	0.07	2468	0.28
4	ERV interlock	1722	0.20	45	0.01	2517	0.29	4284	0.49
5	CFIS	1752	0.20	41	0.00	0	0	1793	0.20
5a	CFIS+20/60	1734	0.20	45	0.01	1463	0.17	3242	0.37
6	CFIS+Exh cont	1837	0.21	49	0.01	0	0	1886	0.22
6a	CFIS+20/60+Exh cont	1818	0.21	52	0.01	1405	0.16	3275	0.37
7	CFIS+Exh switch	1838	0.21	47	0.01	0	0	1885	0.22
7a	CFIS+20/60+Exh switch	1819	0.21	53	0.01	1405	0.16	3278	0.37
8	Supply	1889	0.22	40	0.00	0	0	1928	0.22
8a	Supply+10/60	1876	0.21	42	0.00	598	0.07	2517	0.29
	average of Sys 2 to 8a	1805	0.21	47	0.01				
	average of 10/60 cycling					615	0.07	2483	0.28
	average of 20/60 cycling					1425	0.16	3265	0.37
	average of no cycling							1862	0.21

Table 3.11(b) Annual runtime hours and runtime fractions for heating, cooling, and ventilation/whole-mixing for Higher-Performance house in Seattle

Ventilation System Number	Description	Annual runtime hours and runtime fractions for heating, cooling, and AHU fan cycling for ventilation (Standard house)							
		Minneapolis							
		heat		cool		ah vent/mix		total ah active	
		(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)
0	Ref, no vent, 5.8 ach50	1882	0.21	316	0.04	0	0	2198	0.25
1	Leaky, no mech vent	2095	0.24	313	0.04	0	0	2408	0.27
2	Exhaust	1953	0.22	341	0.04	0	0	2294	0.26
2a	Exhaust+10/60	1950	0.22	352	0.04	615	0.07	2918	0.33
3	Int. exhaust	1951	0.22	330	0.04	0	0	2281	0.26
3a	Int. exhaust+10/60	1949	0.22	338	0.04	618	0.07	2905	0.33
4	ERV interlock	1919	0.22	346	0.04	2257	0.26	4521	0.52
5	CFIS	1934	0.22	320	0.04	0	0	2254	0.26
5a	CFIS+20/60	1917	0.22	339	0.04	1495	0.17	3752	0.43
6	CFIS+Exh cont	2008	0.23	336	0.04	0	0	2344	0.27
6a	CFIS+20/60+Exh cont	1990	0.23	357	0.04	1460	0.17	3807	0.43
7	CFIS+Exh switch	2008	0.23	339	0.04	0	0	2347	0.27
7a	CFIS+20/60+Exh switch	1989	0.23	356	0.04	1466	0.17	3811	0.44
8	Supply	2055	0.23	313	0.04	0	0	2368	0.27
8a	Supply+10/60	2051	0.23	323	0.04	610	0.07	2984	0.34
	average of Sys 2 to 8a	1975	0.23	338	0.04				
	average of 10/60 cycling					614	0.07	2935	0.34
	average of 20/60 cycling					1474	0.17	3790	0.43
	average of no cycling							2306	0.26

Table 3.12(a) Annual runtime hours and runtime fractions for heating, cooling, and ventilation/whole-mixing for Standard-Performance house in Minneapolis

Ventilation System Number	Description	Annual runtime hours and runtime fractions for heating, cooling, and AHU fan cycling for ventilation (High-Perf house)							
		Minneapolis							
		heat		cool		ah vent/mix		total ah active	
		(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)	(hrs)	(frac)
0	Ref, no vent, 5.8 ach50	1882	0.21	316	0.04	0	0	2198	0.25
1	Leaky, no mech vent	2095	0.24	313	0.04	0	0	2408	0.27
2	Exhaust	2592	0.30	205	0.02	0	0	2798	0.32
2a	Exhaust+10/60	2586	0.30	212	0.02	569	0.07	3367	0.38
3	Int. exhaust	2598	0.30	192	0.02	0	0	2790	0.32
3a	Int. exhaust+10/60	2591	0.30	199	0.02	571	0.07	3361	0.38
4	ERV interlock	2538	0.29	196	0.02	2479	0.28	5213	0.60
5	CFIS	2597	0.30	175	0.02	0	0	2772	0.32
5a	CFIS+20/60	2548	0.29	193	0.02	1241	0.14	3982	0.45
6	CFIS+Exh cont	2693	0.31	199	0.02	0	0	2892	0.33
6a	CFIS+20/60+Exh cont	2642	0.30	213	0.02	1215	0.14	4070	0.46
7	CFIS+Exh switch	2692	0.31	199	0.02	0	0	2891	0.33
7a	CFIS+20/60+Exh switch	2642	0.30	213	0.02	1214	0.14	4069	0.46
8	Supply	2715	0.31	172	0.02	0	0	2888	0.33
8a	Supply+10/60	2711	0.31	182	0.02	566	0.06	3459	0.39
	average of Sys 2 to 8a	2627	0.30	196	0.02				
	average of 10/60 cycling					569	0.06	3396	0.39
	average of 20/60 cycling					1224	0.14	4041	0.46
	average of no cycling							2827	0.32

Table 3.12(b) Annual runtime hours and runtime fractions for heating, cooling, and ventilation/whole-mixing for Higher-Performance house in Minneapolis

Tables 3.13(a) and (b) summarize the runtime data given in Tables 3.7(a) through 3.12(b). The average runtime fractions shown include all of the ventilation Systems 2 through 8a, but not the Standard house without ventilation or the Leaky house without ventilation. The heating and cooling runtimes sometimes varied a lot between the Standard-Performance house and the Higher-Performance house, but the central air handler fan runtimes for ventilation supply and whole-house ventilation air distribution were fairly consistent between the two house types. Previous field monitoring data from Higher-Performance houses with System 5a (CFIS+20/60 minimum) had shown that central fan cycling for ventilation only was needed roughly 15% of the year. The colder climate simulations agree well with that but the warmer climate simulations show about 20% of the year.

As shown in Tables 3.7(a) through 3.12(b), for systems with additional central fan runtime, the air handler total runtime fraction was higher than the controlled minimum of 20 min/h or 10 min/h. That seems to be due to the accumulation of periods where heating and cooling runtime exceeded the minimum fan runtime requirement.

Standard house

	Avg heating runtime frac (Sys 2 to 8a)	Avg cooling runtime frac (Sys 2 to 8a)	Avg air handler runtime fraction for ventilation and mixing only	
			20/60 ventilation	10/60 mixing
Houston	0.11	0.14	0.21	0.10
Phoenix	0.06	0.19	0.19	0.08
Charlotte	0.20	0.09	0.19	0.09
Kansas City	0.22	0.07	0.17	0.08
Seattle	0.33	0.02	0.18	0.09
Minneapolis	0.23	0.04	0.17	0.07

Table 3.13(a) Average annual runtime fractions for heating, cooling, and ventilation/whole-mixing for Standard-Performance house

Higher-Performance house

	Avg heating runtime frac (Sys 2 to 8a)	Avg cooling runtime frac (Sys 2 to 8a)	Avg air handler runtime fraction for ventilation and mixing only	
			20/60 ventilation	10/60 mixing
Houston	0.10	0.09	0.22	0.11
Phoenix	0.08	0.16	0.20	0.09
Charlotte	0.17	0.05	0.19	0.09
Kansas City	0.25	0.05	0.16	0.07
Seattle	0.21	0.01	0.16	0.07
Minneapolis	0.30	0.02	0.14	0.06

Table 3.13(b) Average annual runtime fractions for heating, cooling, and ventilation/whole-mixing for Higher-Performance house

Moisture Analysis

As building enclosures and air distribution duct systems become more efficient, sensible cooling load is reduced but the latent cooling load remains mostly the same. The sensible cooling balance point temperature is raised resulting in less cooling system operation during periods of low sensible cooling load, especially in moderate Fall and Spring swing seasons. Less cooling system operation results in less moisture removal. With the same moisture load and less moisture removal, indoor humidity might increase above comfortable or healthy levels.

For Houston, Figures 3.10(a) and (b) show the indoor relative humidity frequency in 1% bins for the entire year. For the Standard-Performance house, between 40% and 70% indoor relative humidity, clearly less ventilation results in lower indoor humidity. However, the number of hours above 70% relative humidity was about the same for all of the ventilation systems, even for those that did not meet the ASHRAE Standard 62.2 flow rate. That indicates that under those high humidity conditions ventilation was not the driving factor, rather, low sensible heat gain precluded cooling system operation and subsequent moisture removal. Figures 3.11(a) and (b) show the indoor relative humidity frequency for the cooling season only, and the heating season is shown in Figures 3.12(a) and (b). For Houston, a hot-humid climate, the result doesn't change much whether looking at the whole year or just the cooling season because the cooling season is the bulk of the year. Indoor relative humidity frequency for the Standard house without ventilation peaks at about 50%. The leaky house peaks at about 55% relative humidity. Adding ventilation to the Standard-Performance house, at the ASHRAE Standard 62.2 rate, causes indoor relative humidity to peak at about 60%, whereas the Higher-Performance houses peak just above 65% relative humidity. For Systems 5 and 5a, which have less than the ASHRAE Standard 62.2 ventilation rate, it is interesting to note they generally show lower indoor relative humidity except for a second peak at around 80 to 85% relative humidity for the Higher-Performance house in Figure 3.10(b). The second peak may be explained by the Systems with higher ventilation rates causing more cooling demand at times which results in more moisture removal and better humidity control during those times. Looking at the heating season, indoor relative humidity frequency is shifted upwards for the Higher-Performance house less than in the cooling season, peaking just below 60%. Overall, it is clear from this simulation data, and from prior field measurements (Rudd and Henderson 2007), that supplemental dehumidification is required in Houston to control indoor relative humidity outside of heavy cooling periods in Higher-Performance houses.

Interestingly, some of the same trends can be seen in Minneapolis, which is cold and dry in winter but has warm-humid periods in summer. Figures 3.13(a) and (b) clearly illustrate the increase in indoor humidity going from the Standard-Performance house to the Higher-Performance house in Minneapolis. It was not necessary to break out the cooling and heating seasons for Minneapolis because the winter and summer humps were evident. Ventilation system type had little impact on indoor relative humidity in Minneapolis, even for systems without the full ASHRAE Standard 62.2 mechanical ventilation rate.

Humidity frequency plots for the other climate locations can be found in Appendix A and Appendix B.

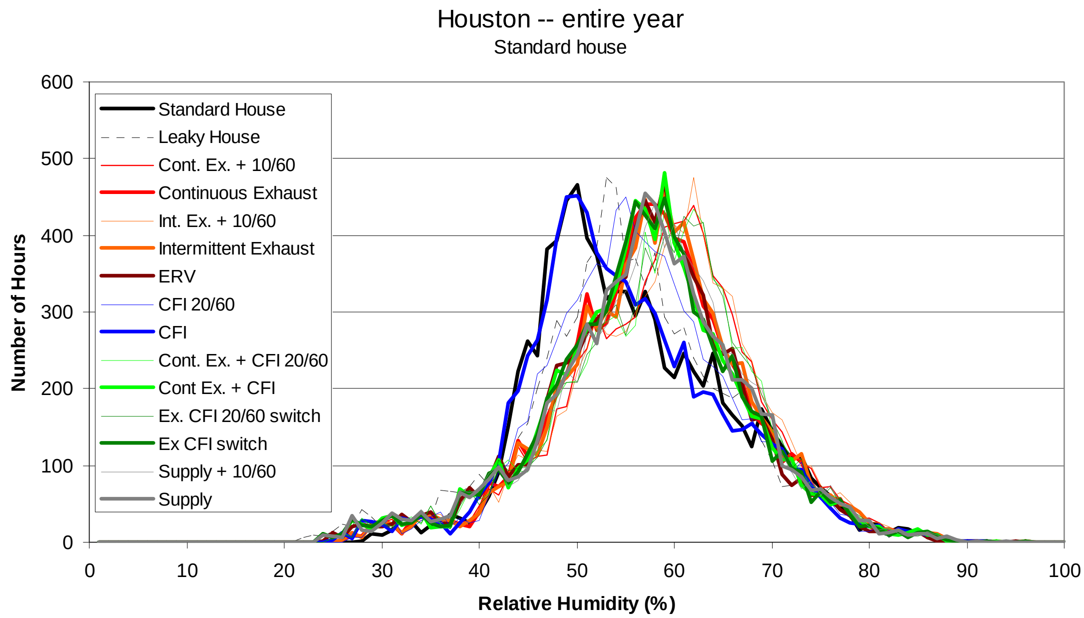


Figure 3.10(a) Frequency distribution of hourly average indoor relative humidity (1% RH bins) for the <u>entire year</u> for the Standard-Performance house in Houston

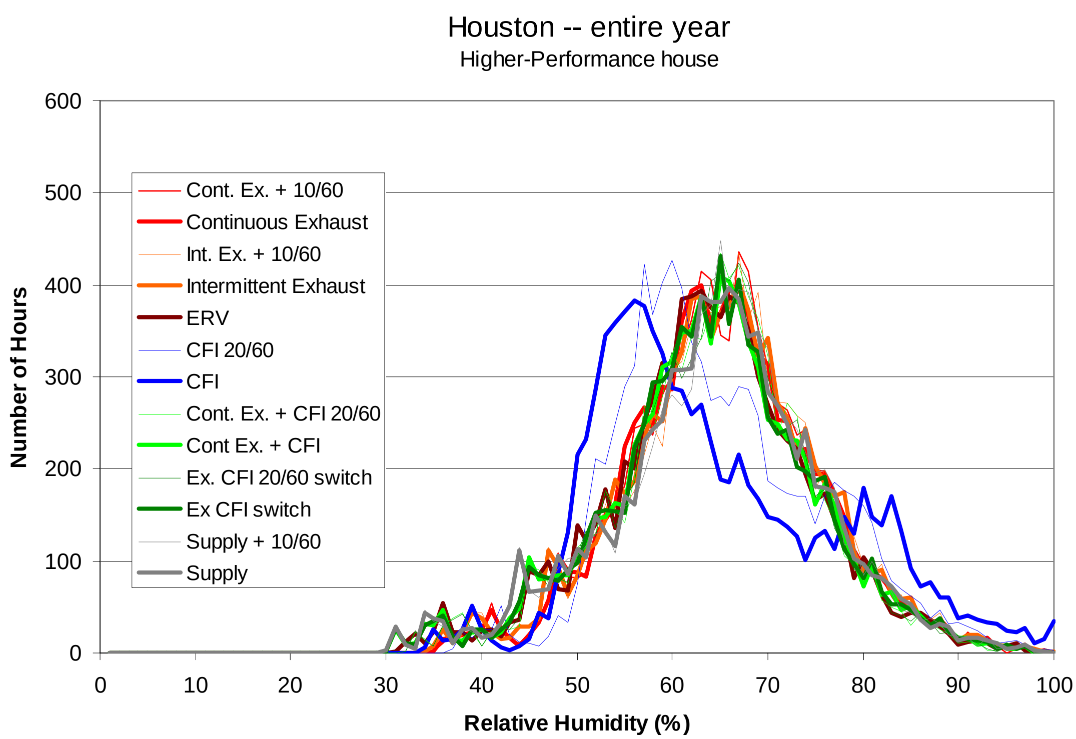


Figure 3.10(b) Frequency distribution of hourly average indoor relative humidity (1% RH bins) for the <u>entire year</u> for the Higher-Performance house in Houston

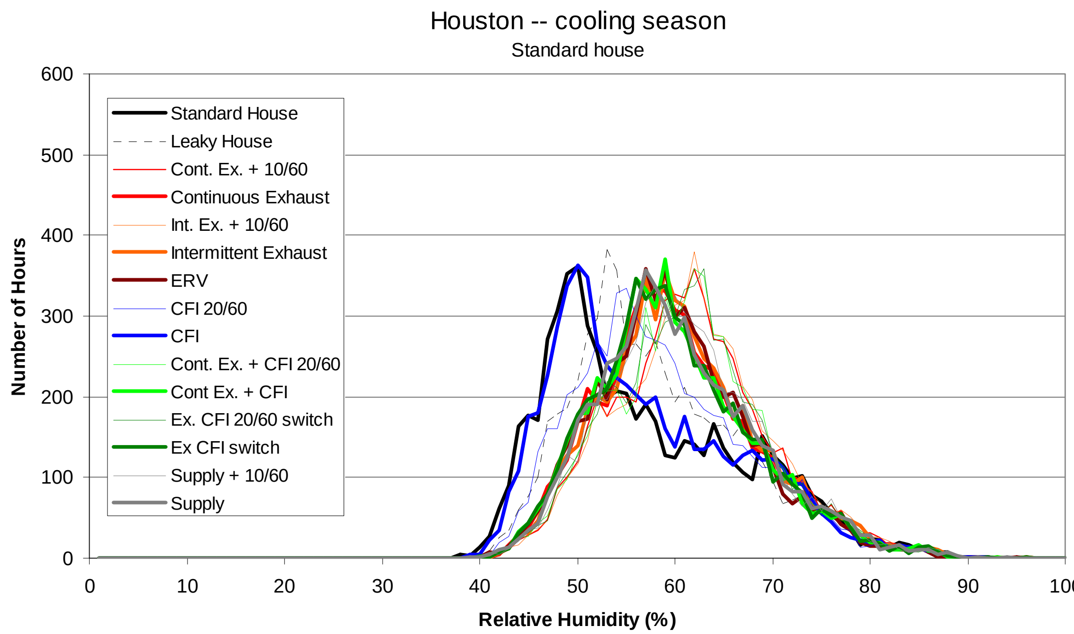


Figure 3.11(a) Frequency distribution of hourly average indoor relative humidity (1% RH bins) for the cooling season for the Standard-Performance house in Houston

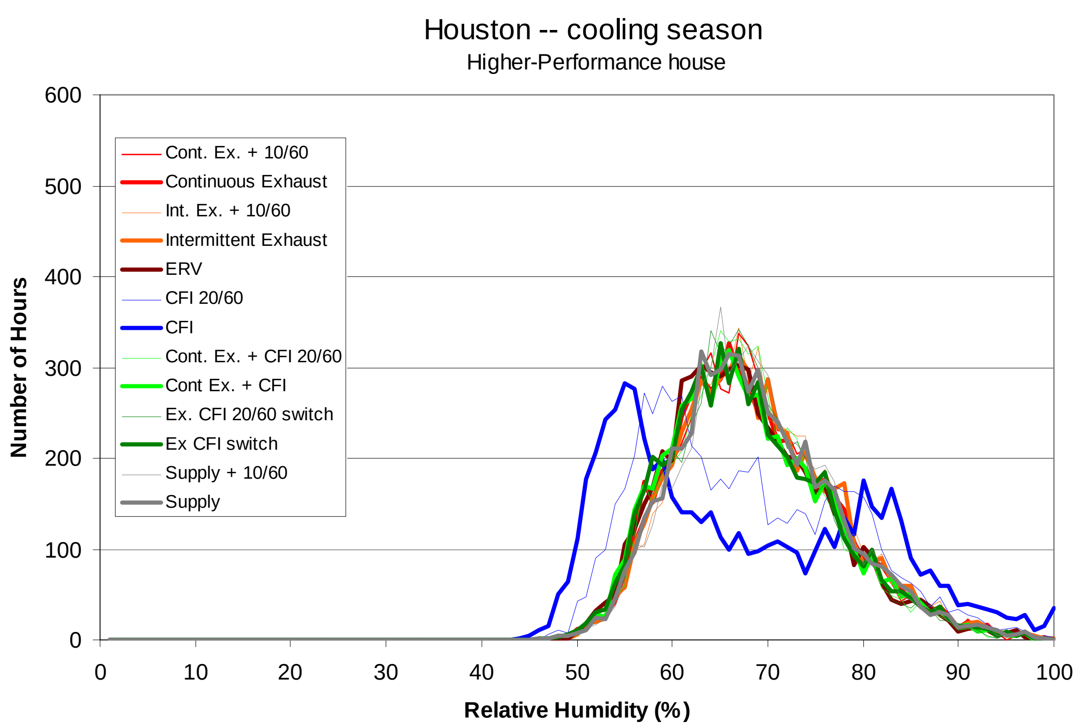


Figure 3.11(b) Frequency distribution of hourly average indoor relative humidity (1% RH bins) for the cooling season for the Higher-Performance house in Houston

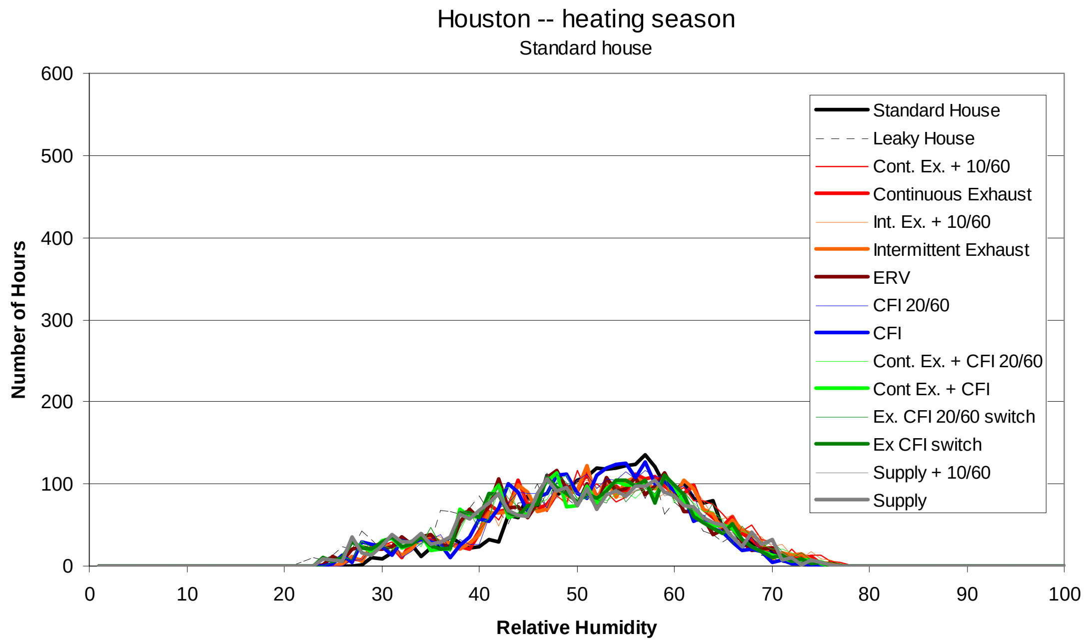


Figure 3.12(a) Frequency distribution of hourly average indoor relative humidity (1% RH bins) for the heating season for the Standard-Performance house in Houston

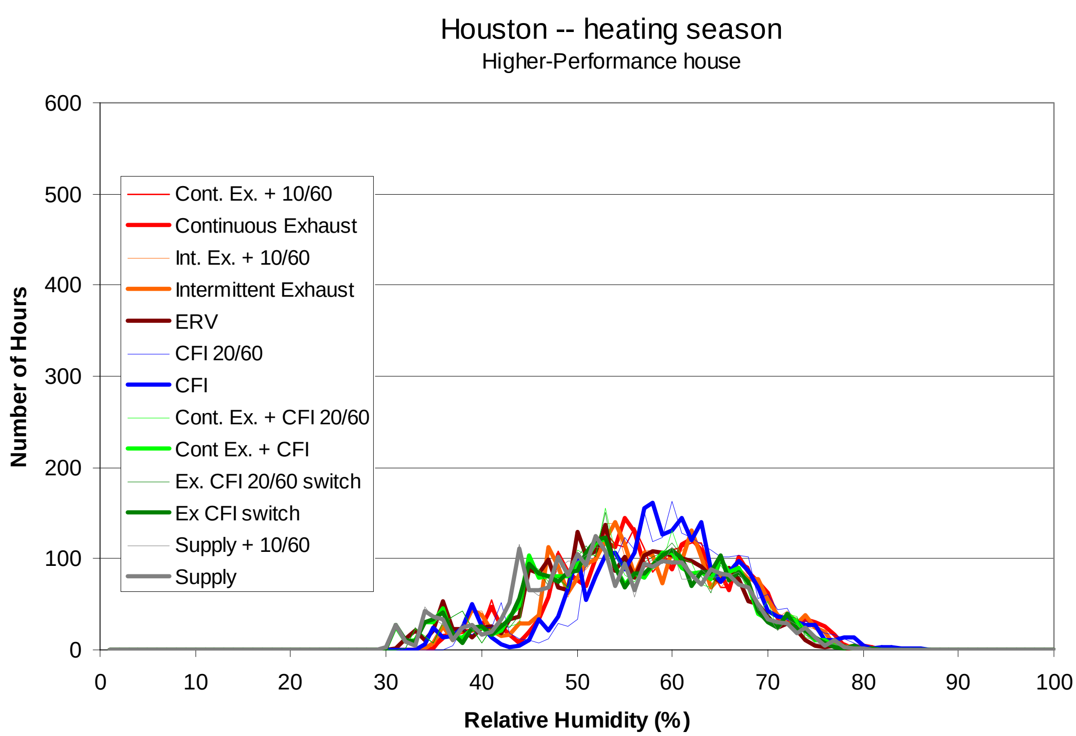


Figure 3.12(b) Frequency distribution of hourly average indoor relative humidity (1% RH bins) for the heating season for the Higher-Performance house in Houston

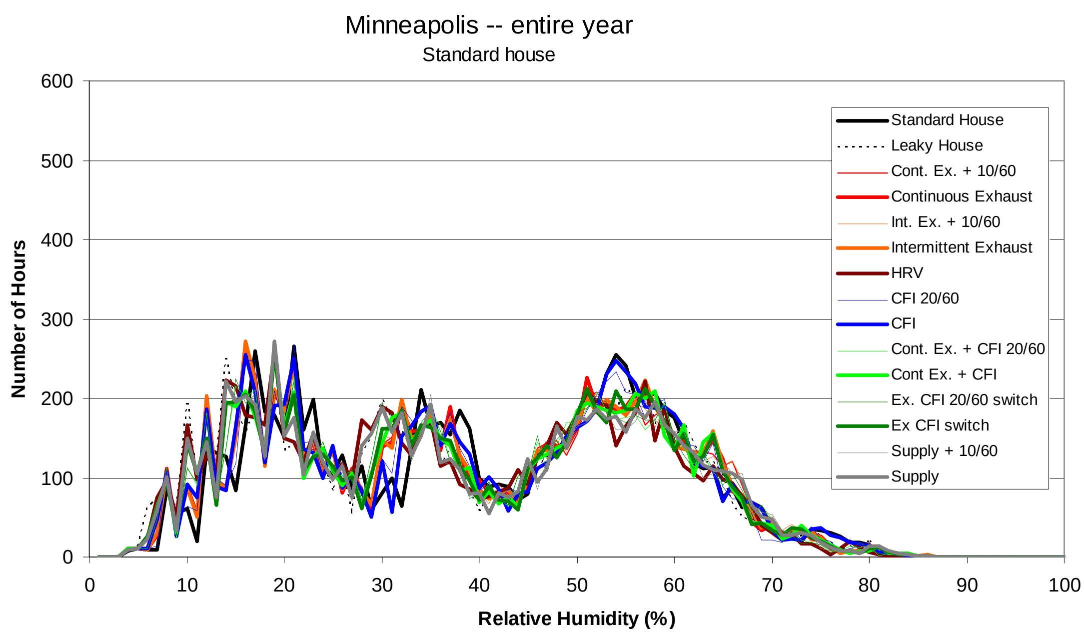


Figure 3.13(a) Frequency distribution of hourly average indoor relative humidity (1% RH bins) for the <u>entire year</u> for the Standard-Performance house in Minneapolis

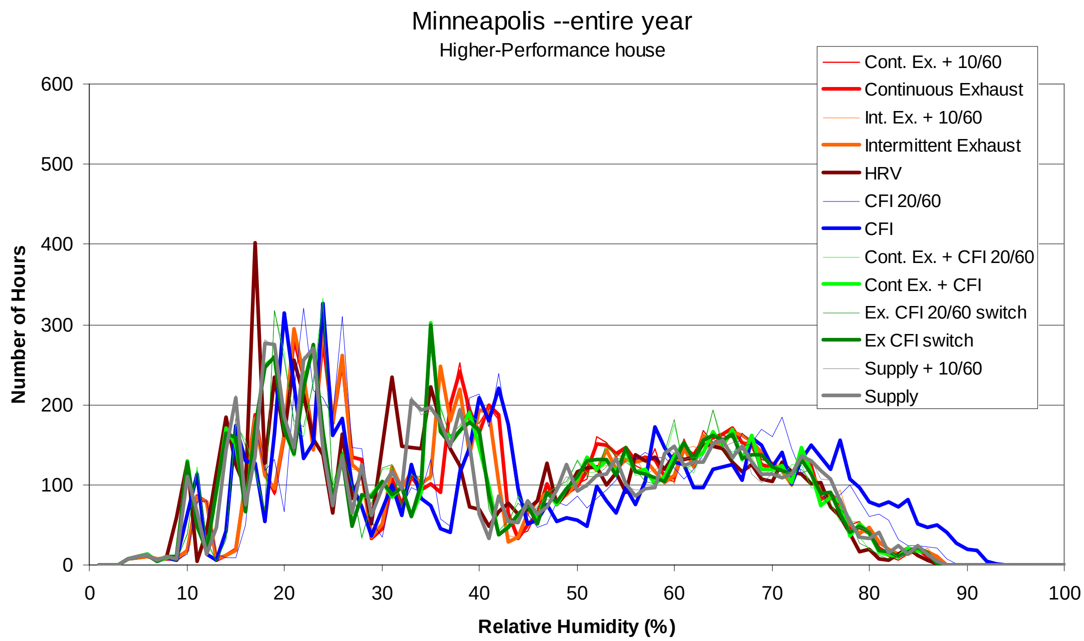


Figure 3.13(b) Frequency distribution of hourly average indoor relative humidity (1% RH bins) for the <u>entire year</u> for the Higher-Performance house in Minneapolis

Tables 3.14(a) and (b) and Figures 3.14(a) and (b) show the number of annual hours and the percentage of annual hours where indoor relative humidity exceeded 60%. For the Standard-Performance house, relative humidity was over 60% in Houston about 35% of the year, while in all other climates it was over 60% relative humidity less than 20% of the year. Those numbers essentially double for the Higher-Performance house.

Table 3.14(a) Number of annual hours where indoor relative humidity exceeded 60% RH for Standard-Performance house

Ventilation System Number	Number of annual hours where RH was above 60%					
	Houston	Phoenix	Charlotte	Kansas City	Seattle	Minneapolis
0	2659	128	1496	1499	1532	1138
1	2542	40	1561	1506	728	1120
2	3266	53	1619	1653	915	1167
2a	3939	30	1785	1792	832	1207
3	3403	60	1709	1693	940	1197
3a	4054	32	1887	1849	842	1214
4	3076	27	1592	1620	529	988
5	2406	110	1423	1451	1328	1155
5a	2712	86	1472	1483	984	1095
6	3019	42	1594	1588	829	1172
6a	3790	25	1755	1735	677	1124
7	3019	42	1592	1593	834	1162
7a	3781	24	1743	1718	670	1110
8	3185	47	1727	1652	684	1183
8a	3410	45	1752	1751	676	1203

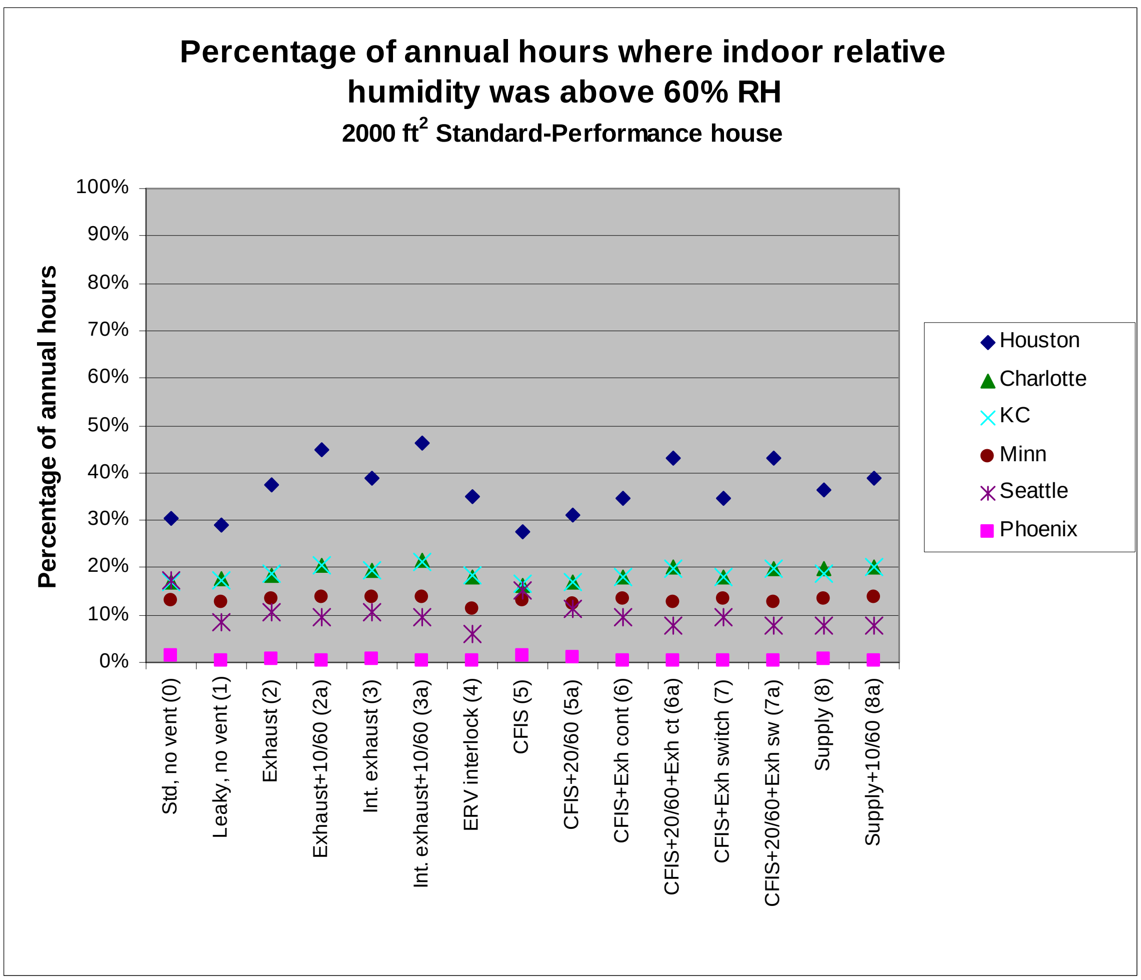


Figure 3.14(a) Percentage of annual hours where indoor relative humidity exceeded 60% RH for Standard-Performance house

Table 3.14(b) Number of annual hours where indoor relative humidity exceeded 60% RH for Higher-Performance house

Ventilation System Number	Number of annual hours where RH was above 60%					
	Houston	Phoenix	Charlotte	Kansas City	Seattle	Minneapolis
2	6089	64	3522	2795	1737	2441
2a	6160	67	3578	2818	1713	2455
3	6133	64	3613	2825	1743	2464
3a	6214	66	3669	2858	1722	2468
4	5765	44	3592	2650	1087	2132
5	4725	420	4041	2992	3865	3005
5a	5370	198	3939	2997	3242	2777
6	5771	55	3455	2708	1625	2392
6a	5906	52	3488	2700	1433	2266
7	5764	54	3463	2707	1627	2399
7a	5875	56	3472	2704	1426	2262
8	5930	59	3673	2770	1474	2377
8a	6019	64	3721	2807	1468	2368

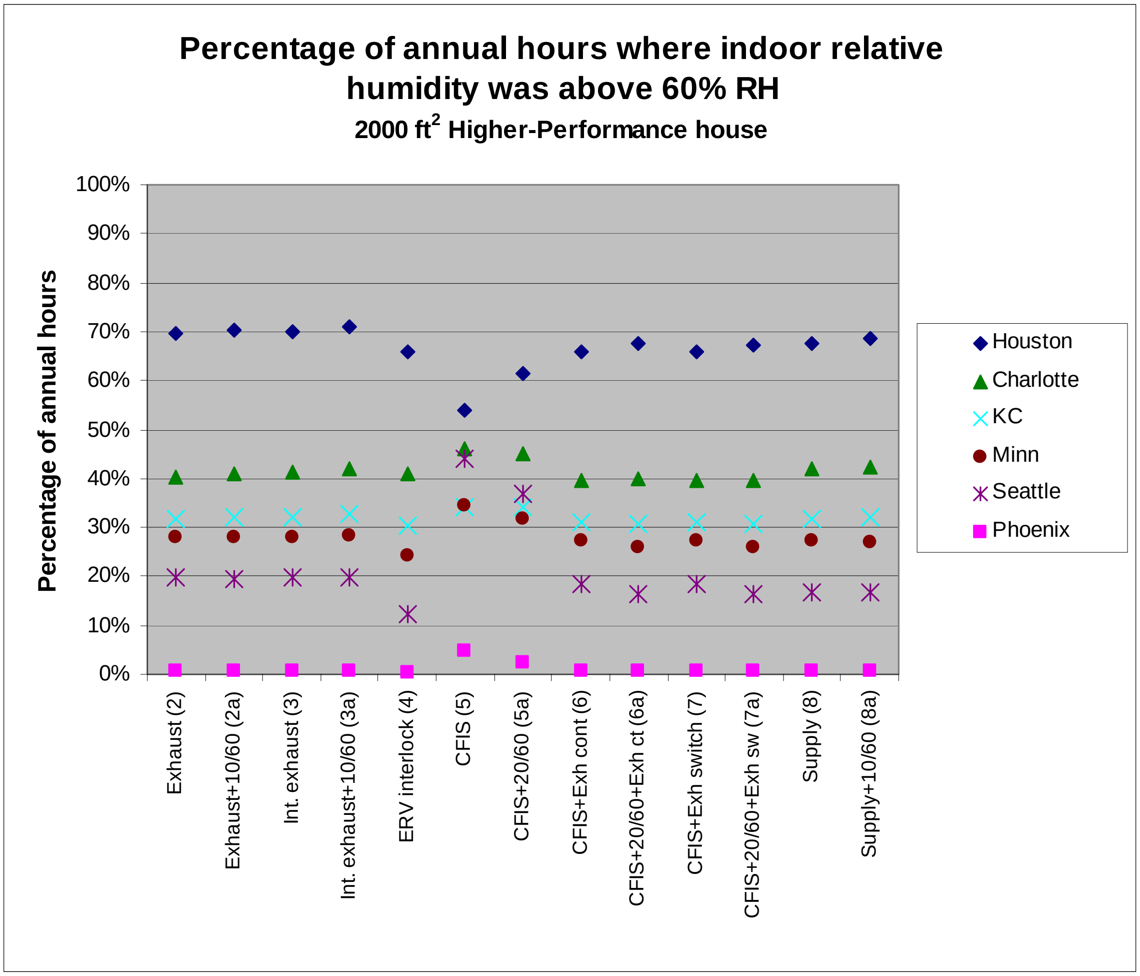


Figure 3.14(b) Percentage of annual hours where indoor relative humidity exceeded 60% RH for Higher-Performance house

Sensitivity of House Size

Simulations were completed for Houston and Minneapolis to look at small (1000 ft^2, 2 bedroom) and large (4000 ft^2, 5 bedroom) houses to see how air change and moisture conditions may differ from the medium size (2000 ft^2, 3 bedroom) house. As shown in Table 3.15, the annual average air change rate is higher for smaller houses with mechanical ventilation then it is for larger houses with mechanical ventilation. This is due to the way the ventilation rate scales with house floor area and number of bedrooms. The change in annual average ach is less for the System 0 (no mechanical ventilation) than it is for the systems with mechanical ventilation, however, on a smaller time scale (hourly or daily basis) the change in ach is more variable for the house without mechanical ventilation.

Table 3.15 Annual average air change rate for the large and small house, for ventilation Systems 0, 2, and 6a, using the Standard-Performance house characteristics

	Annual average air change rate (ach)					
	Houston			Minneapolis		
System Number	Small house	Medium house	Large house	Small house	Medium house	Large house
0	0.20	0.18	0.16	0.31	0.29	0.26
2	0.35	0.28	0.25	0.40	0.35	0.32
6a	0.40	0.33	0.29	0.46	0.40	0.36

Figures 3.15 and 3.16 show the hourly average air change rate frequency, plotted in bins of 0.02 ach, for Systems 0, 2, and 6a for the large and small house, in both Houston and Minneapolis. The data clearly show the higher air change rate for smaller houses.

Figures 3.17 and 3.18 show the relative humidity frequency, plotted in bins of 1% RH, for Systems 0, 2, and 6a for the large and small house, in both Houston and Minneapolis. The data illustrate how moisture buildup is more problematic in small houses where occupant and moisture generation density is higher. Understandably, it is more significant in the more humid climate of Houston.

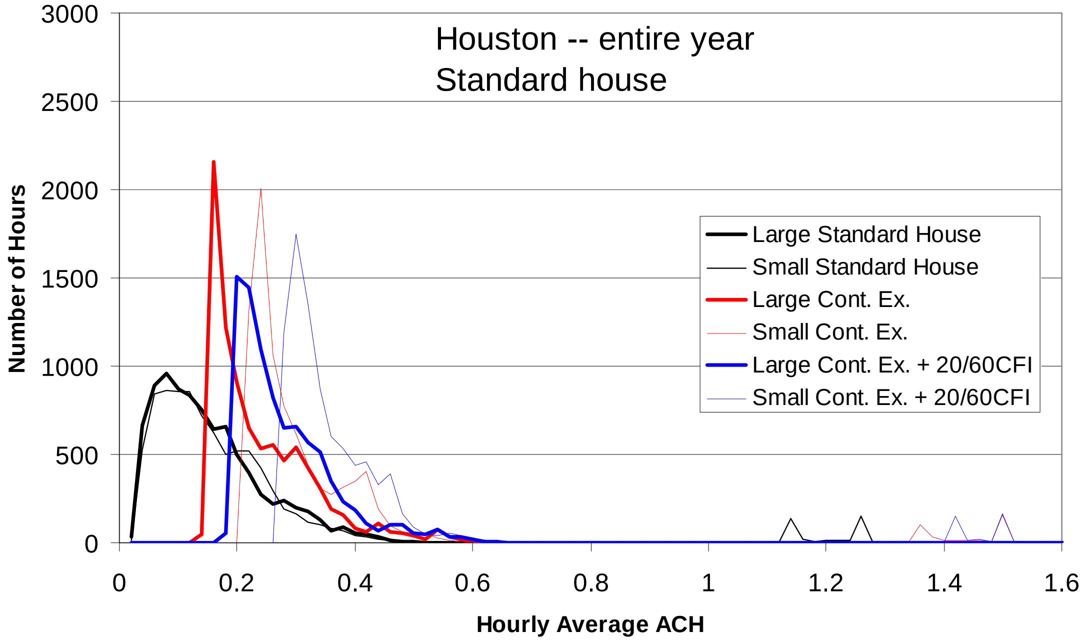


Figure 3.15 Frequency distribution of hourly average air change rate (0.02 ach bins) for large and small houses, for the Standard-Performance house in Houston

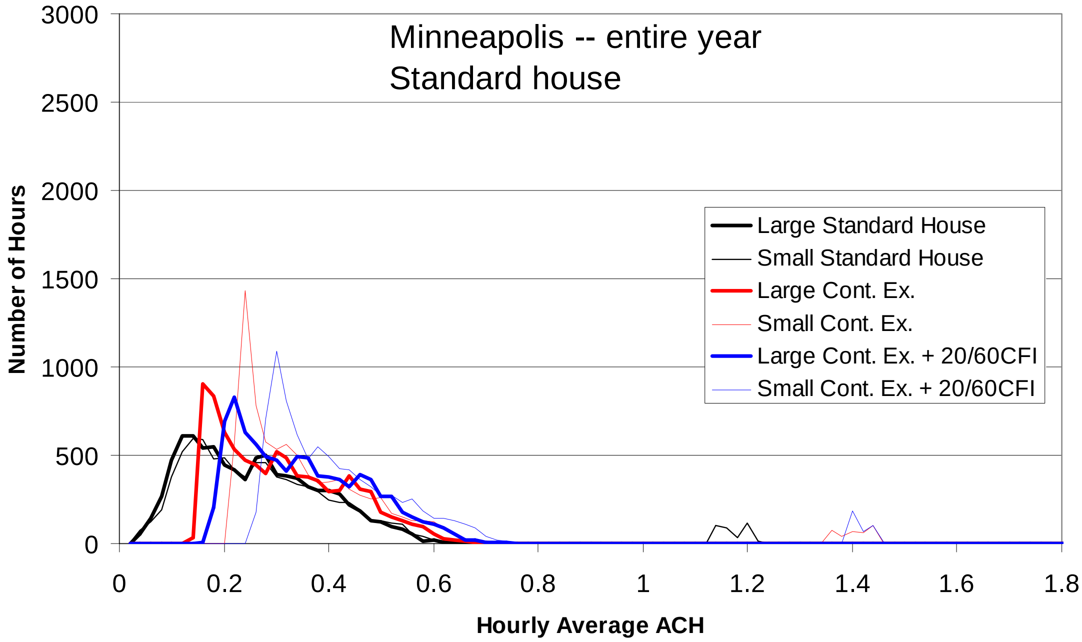


Figure 3.16 Frequency distribution of hourly average air change rate (0.02 ach bins) for large and small houses, for the Standard-Performance house in Minneapolis

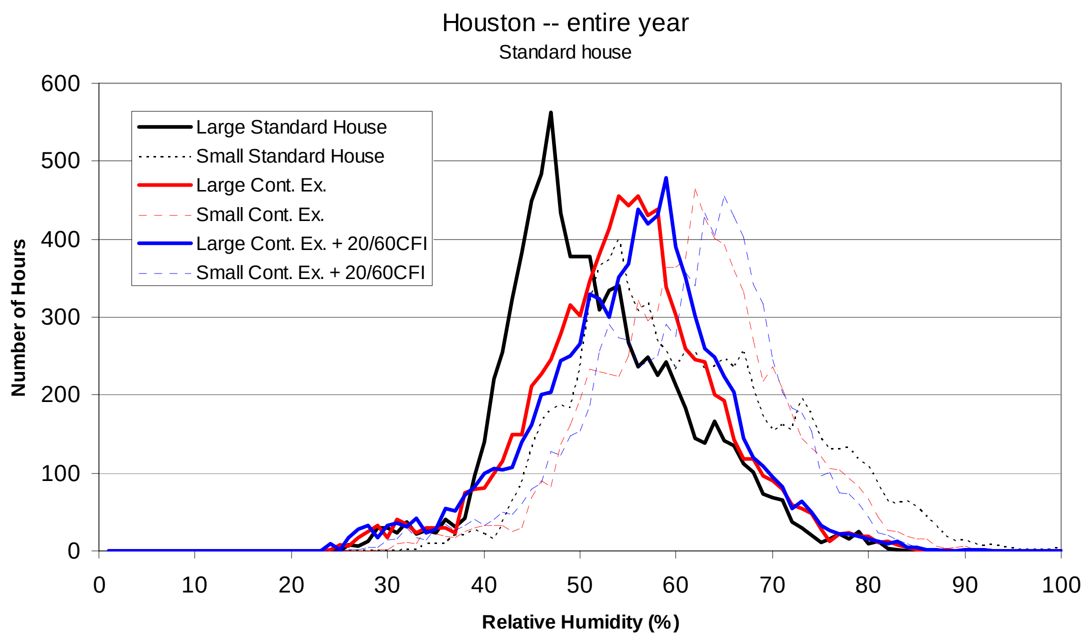


Figure 3.17 Frequency distribution of hourly average indoor relative humidity (1% bins) for large and small houses, for the Standard-Performance house in Houston

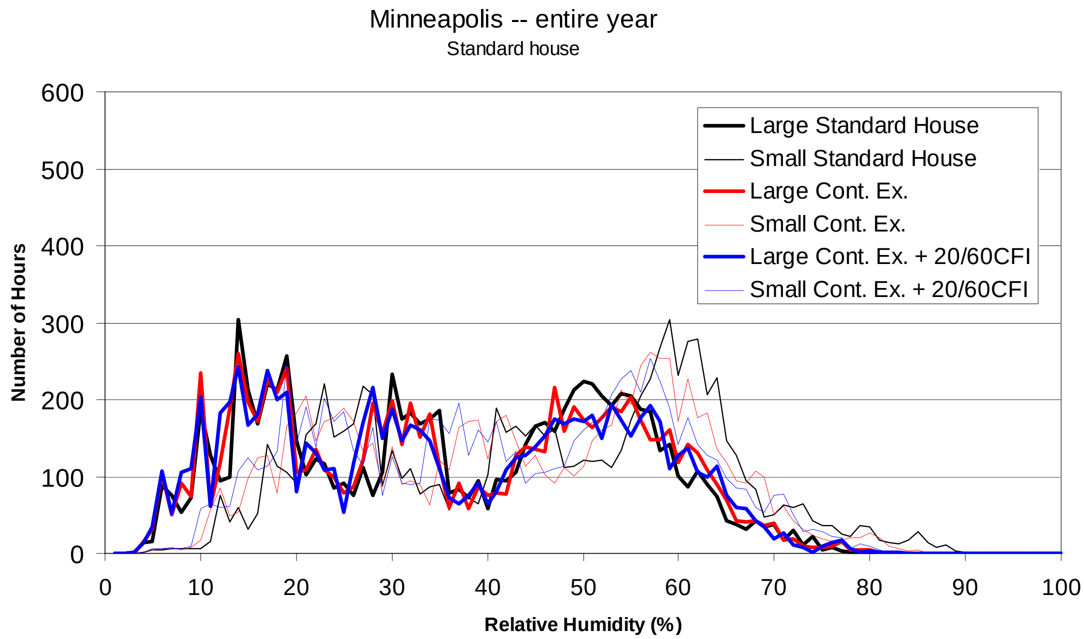


Figure 3.18 Frequency distribution of hourly average indoor relative humidity (1% bins) for large and small houses, for the Standard-Performance house in Minneapolis

Executive Summary (of Task 3)

A comprehensive set of simulations have been performed to examine the relative operating costs and air change rates of various residential ventilation methods. All but two of the systems met ASHRAE Standard 62.2. These simulations included six major U.S. climate zones, three house sizes, two house types (standard-performance IECC and higher-performance), 13 ventilation systems, and 2 different base case houses (same tightness as for the ventilation systems, and leaky enough to meet ASHRAE Standard 62.2 without mechanical ventilation). The key results were:

1. ASHRAE Standard 62.2 can be met using a simple exhaust-only system for moderate annual operating costs of $50 to $100 per year for Standard-Performance houses, and from a savings of $225 to a cost of $150 for Higher-Performance houses compared to the Standard-Performance house without mechanical ventilation.
2. Depending on the climate, minimally ASHRAE Standard 62.2 compliant exhaust-only systems provided between 0.06 to 0.11 more annual average air change in Standard-Performance houses, and between 0.05 less to 0.05 more in Higher-Performance houses compared to the Standard-Performance house without mechanical ventilation. The higher increase in ACH was in the milder climates.
3. More sophisticated systems can provide additional air change and ventilation air distribution, but at different cost, between $70 and $250 more for Standard-Performance houses, and between $200 less to $250 more for Higher-Performance houses compared to the Standard-Performance house without mechanical ventilation.
4. For the same rated fan flow, the greatest increase in air change rate is for balanced systems (HRV/ERV), the least change is for exhaust–only systems, and supply–only systems are between those two (note that ASHRAE Standard 62.2 does not currently distinguish between exhaust, supply, or balanced systems in its mechanical ventilation requirements).
5. HRV/ERV's sized to meet ASHRAE Standard 62.2, and as commonly installed to require coincident operation of the central air handler fan, tend to require the greatest operating cost due to higher air change and fan energy consumption.
6. Depending on climate and electricity cost, the cost to provide ventilation air distribution and thermal comfort mixing using a central fan cycling system at a 20 minute per hour minimum was between $10 and $120 for Standard Performance houses, and between -$20 to +$90 for Higher-Performance houses. At a 10 minute minimum, the cost was between $15 and $60 for Standard-Performance houses and between $10 and $40 for Higher-Performance houses. The higher costs were in hotter climates with more expensive electricity.
7. In humid climates, the ASHRAE Standard 62.2 compliant ventilation systems increased the median relative humidity by about 10% compared to the Standard-Performance house without ventilation, but only about half as much compared to the Leaky house. For Higher-Performance houses, the median relative humidity increased about 15% compared to the Standard-Performance house without mechanical ventilation. It was clear that, with or without mechanical ventilation, supplemental dehumidification would be required to control elevated indoor relative humidity year-around.

8. Higher-Performance houses had better air change control (less variability) than Standard-Performance houses, and had lower operating costs except for Seattle and Kansas City where the lower glazing SHGC increased heating more than it reduced cooling.
9. Using intermittent exhaust ventilation to avoid ventilating for 4 hours during peak load conditions saved less than 0.5% in annual HVAC costs for Standard-Performance houses and showed increased costs of up to 1% for Higher-Performance houses.
10. When using central-fan-integrated supply (CFIS) ventilation in conjunction with exhaust ventilation, switching off the exhaust fan whenever CFIS was already occurring saved only between 0.5% and 1% in annual HVAC costs for Standard- and Higher-Performance houses.

APPENDIX A

Additional plots from simulation results for the Standard-Performance house

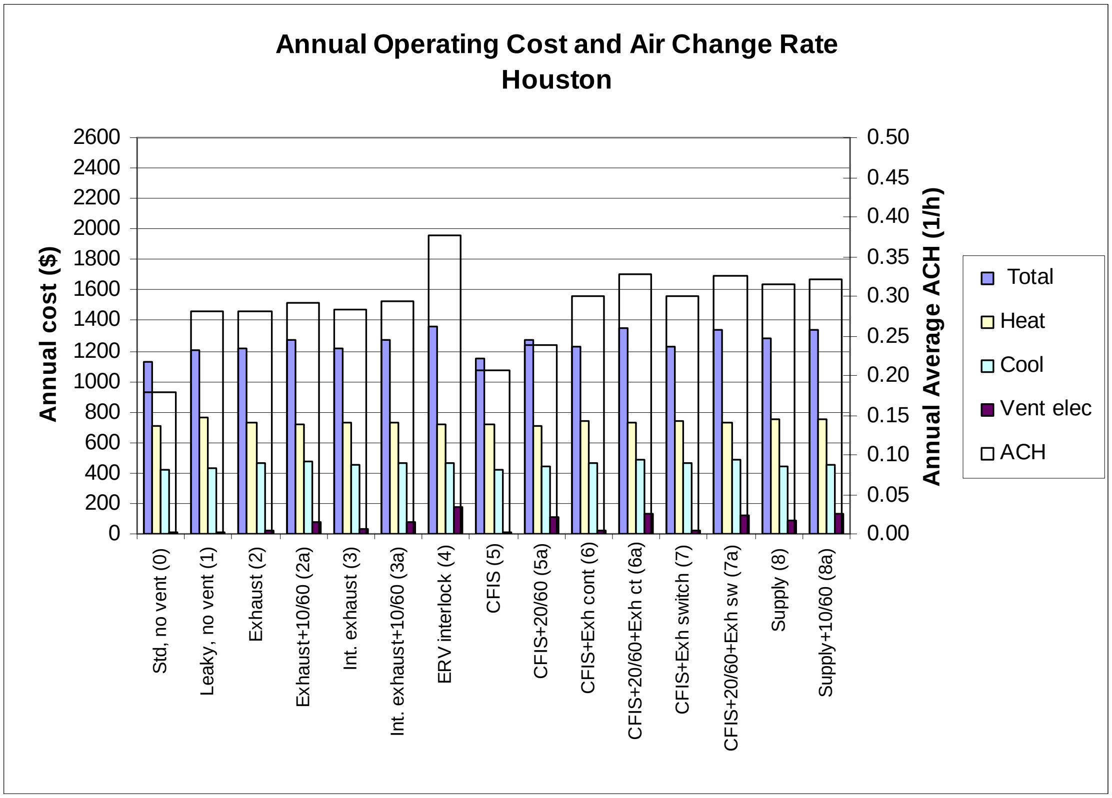


Figure A.1 Annual operating cost and annual average air change rate for Houston

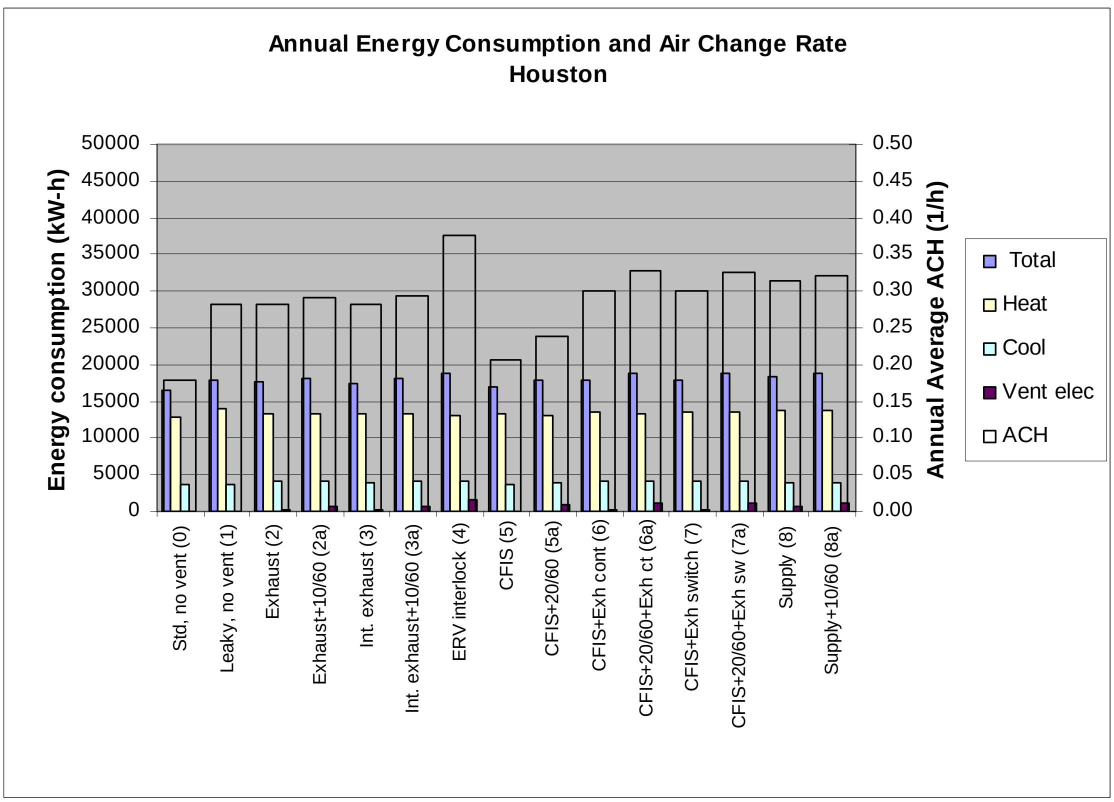


Figure A.2 Annual energy consumption and annual average air change rate for Houston

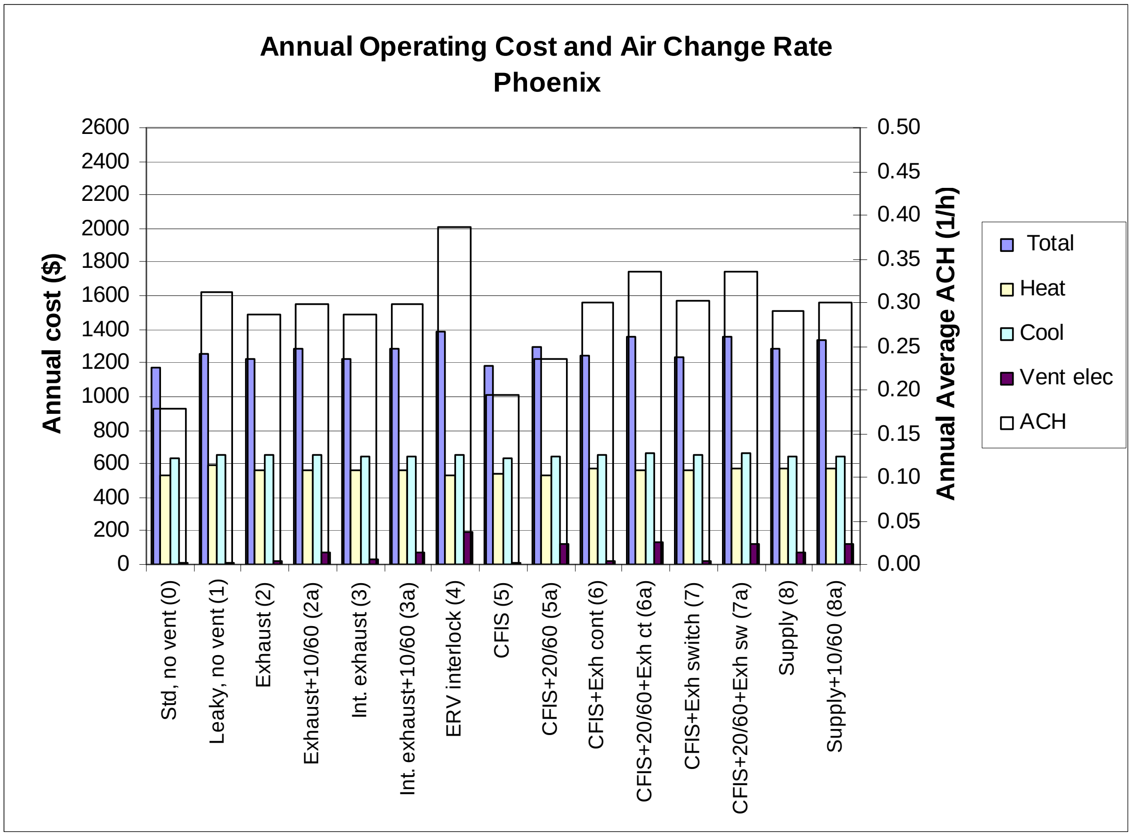


Figure A.3 Annual operating cost and annual average air change rate for Phoenix

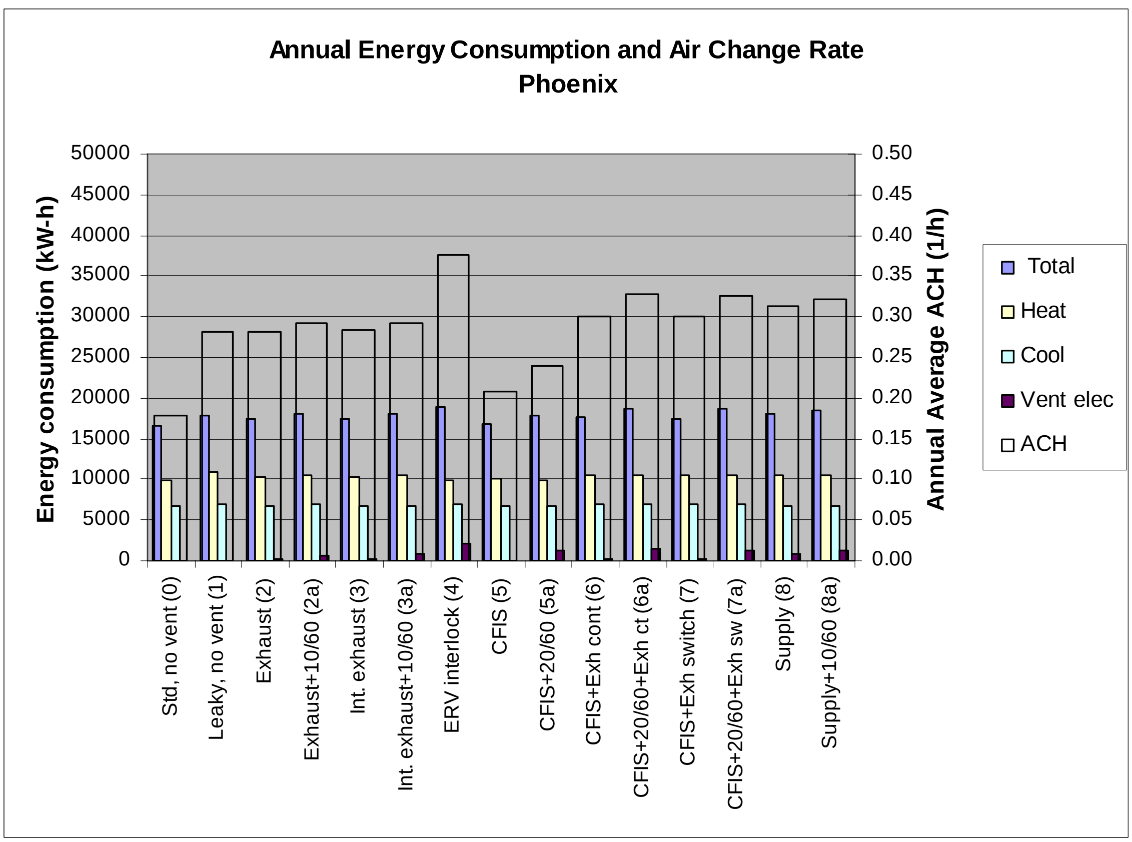


Figure A.4 Annual energy consumption and annual average air change rate for Phoenix

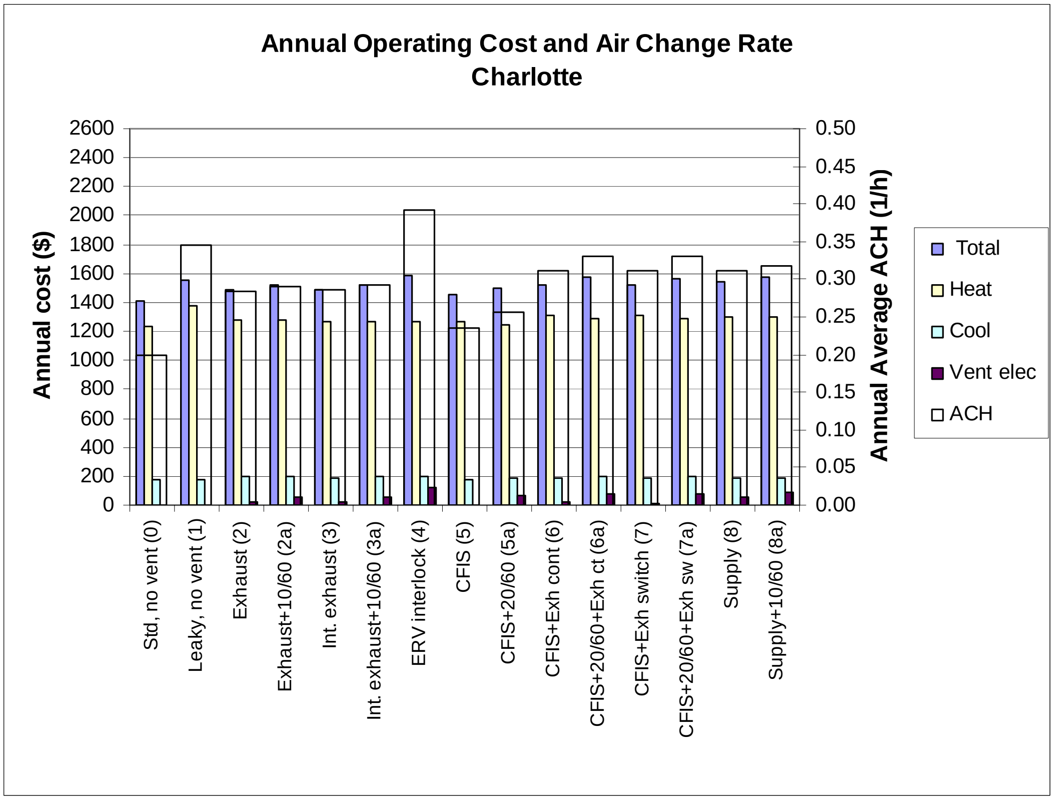


Figure A.5 Annual operating cost and annual average air change rate for Charlotte

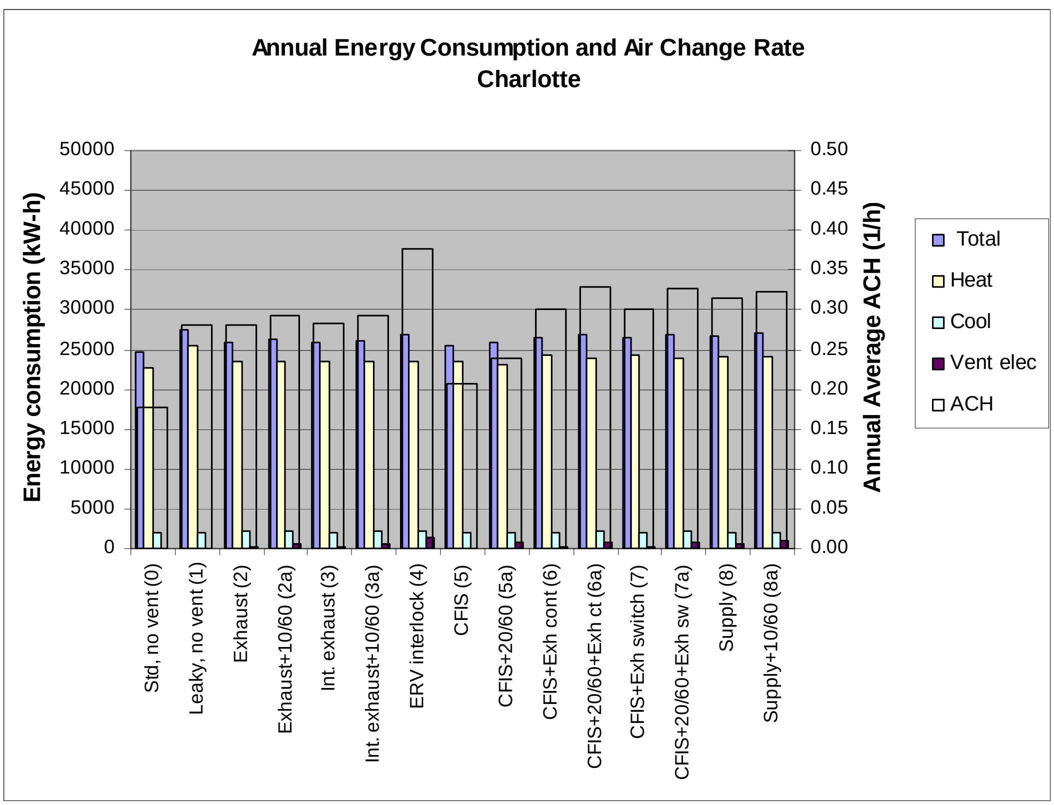


Figure A.6 Annual energy consumption and annual average air change rate for Charlotte

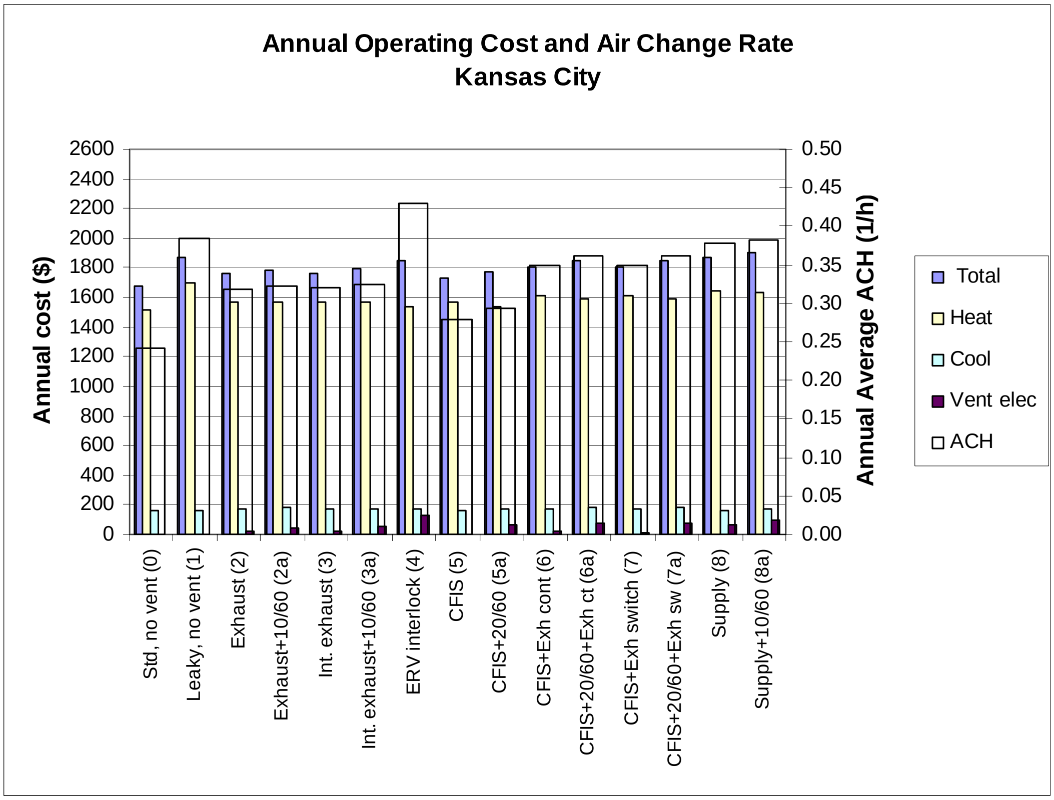


Figure A.7 Annual operating cost and annual average air change rate for Kansas City

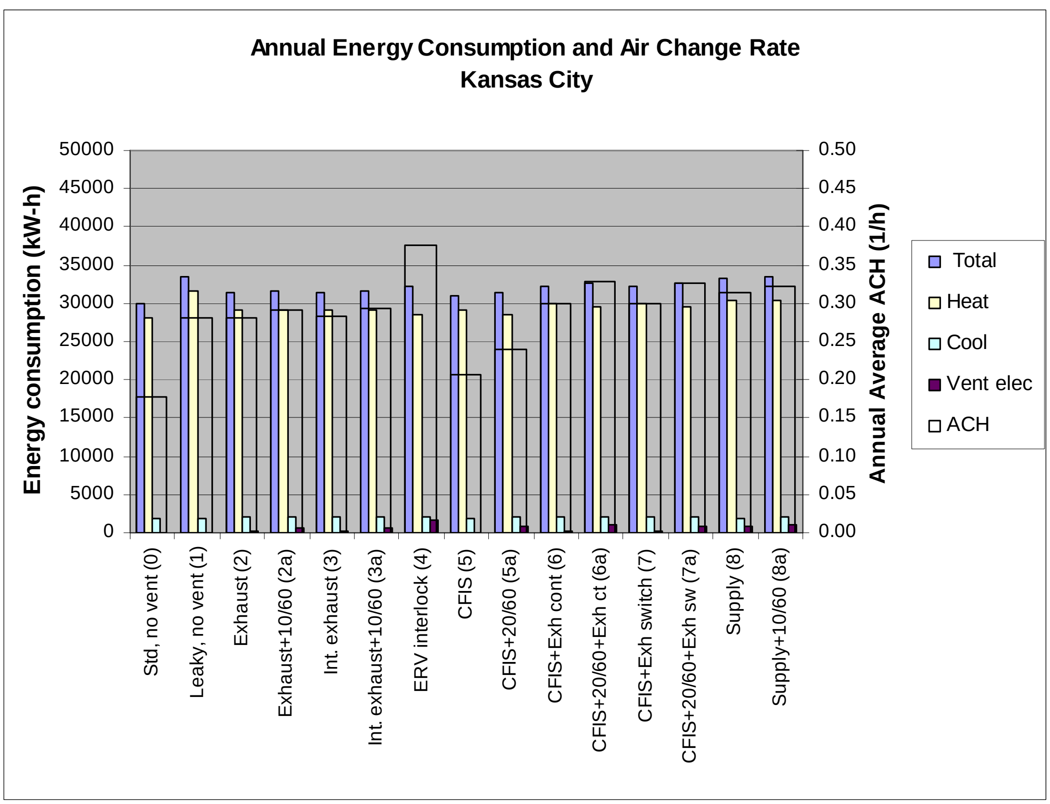


Figure A.8 Annual energy consumption and annual average air change rate for Kansas City

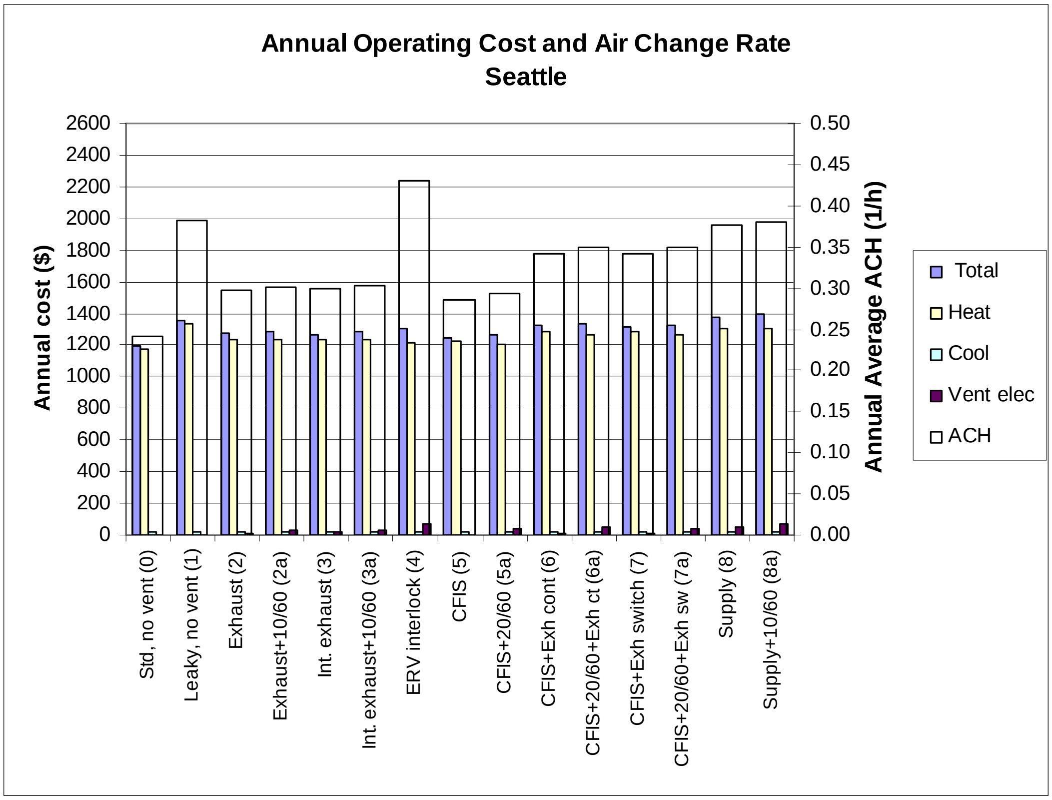


Figure A.9 Annual operating cost and annual average air change rate for Seattle

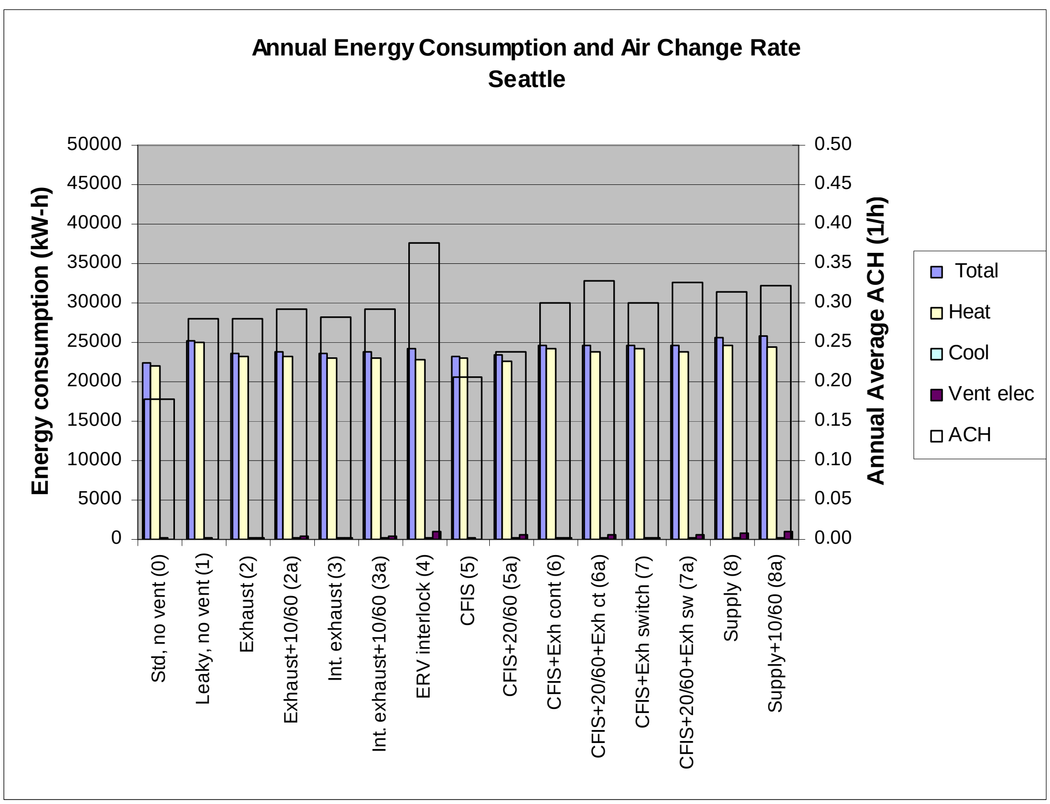


Figure A.10 Annual energy consumption and annual average air change rate for Seattle

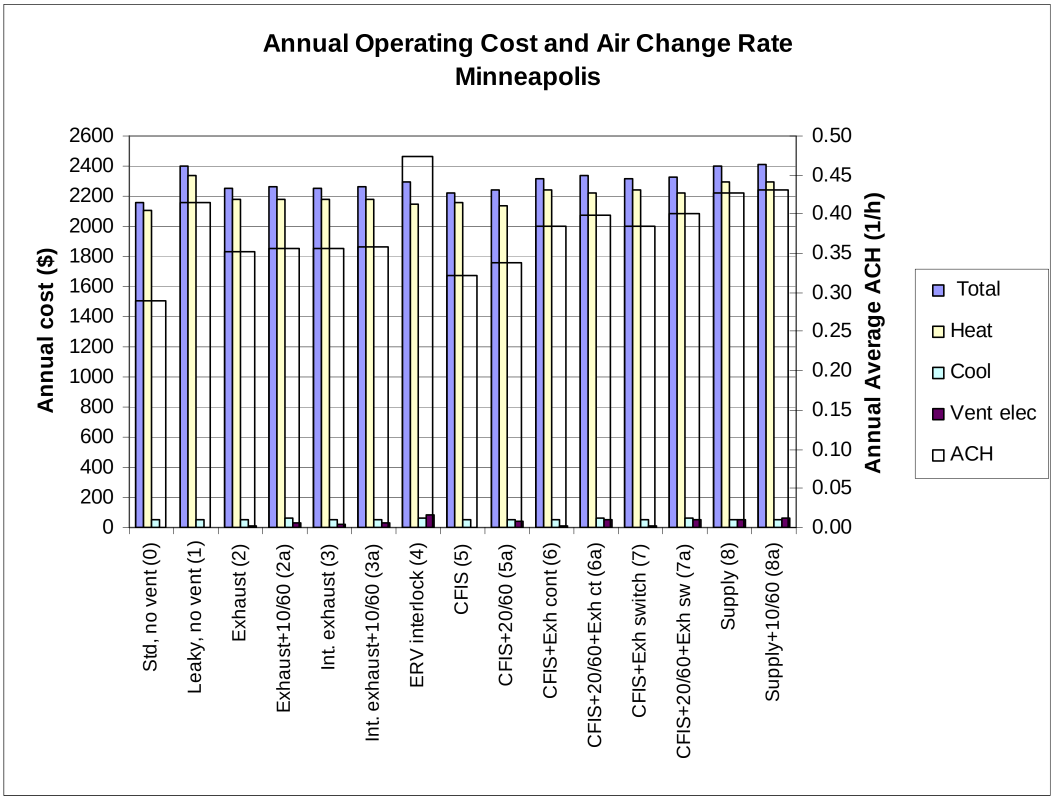


Figure A.11 Annual operating cost and annual average air change rate for Minneapolis

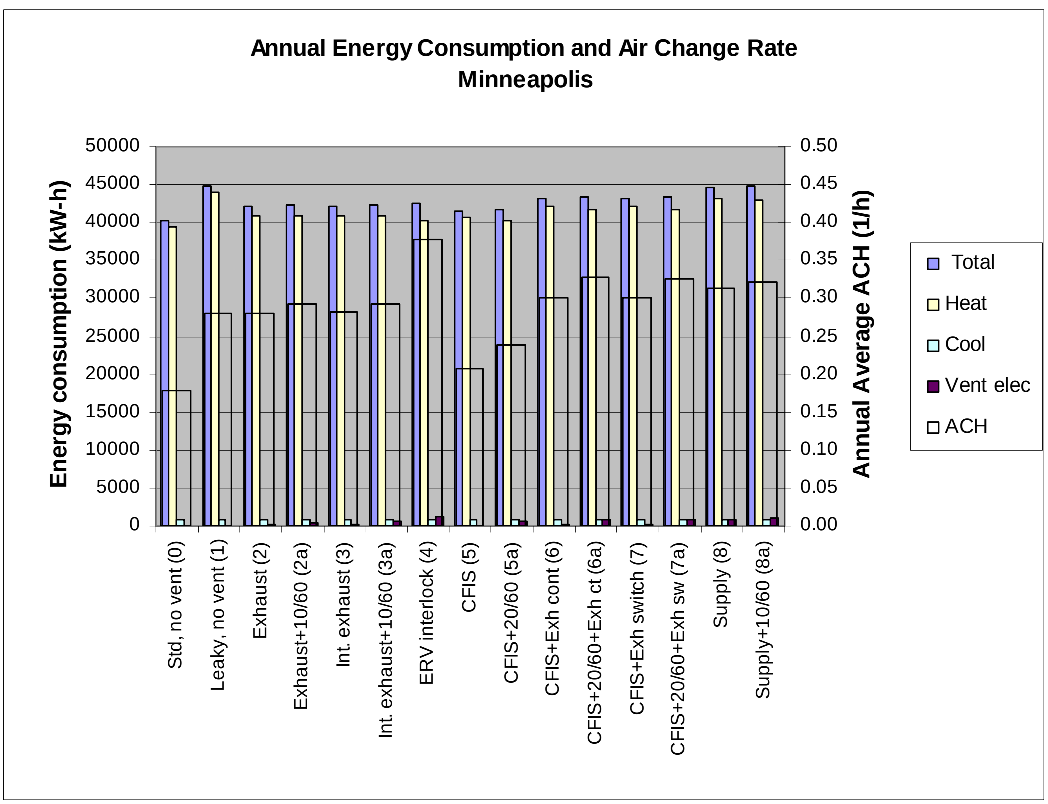


Figure A.12 Annual energy consumption and annual average air change rate for Minneapolis

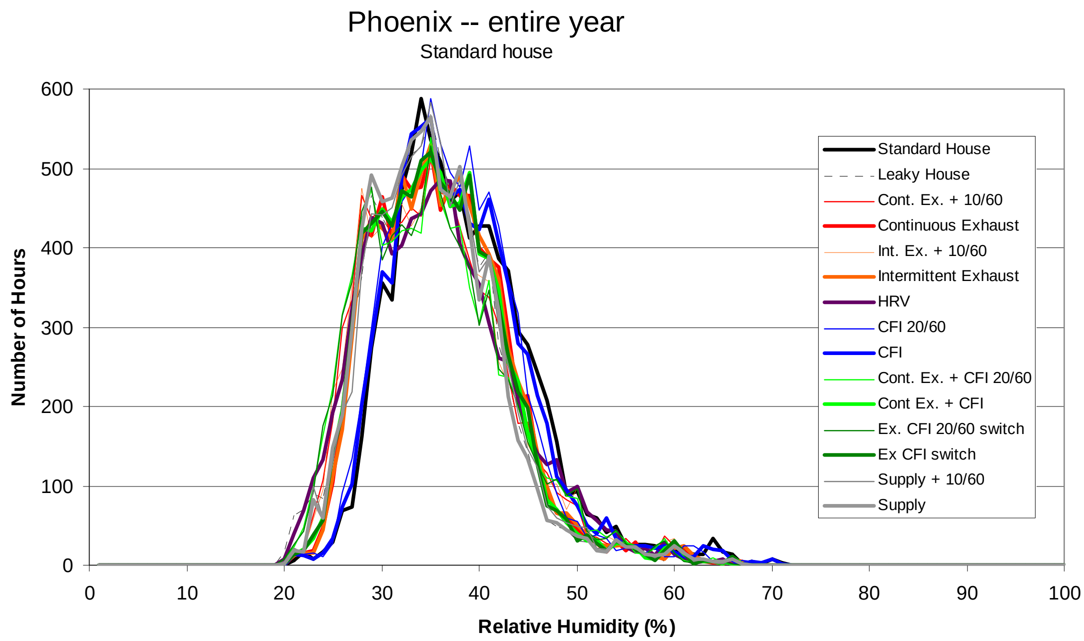


Figure A.13 Frequency distribution of hourly average indoor relative humidity (1% bins) for the entire year in Phoenix

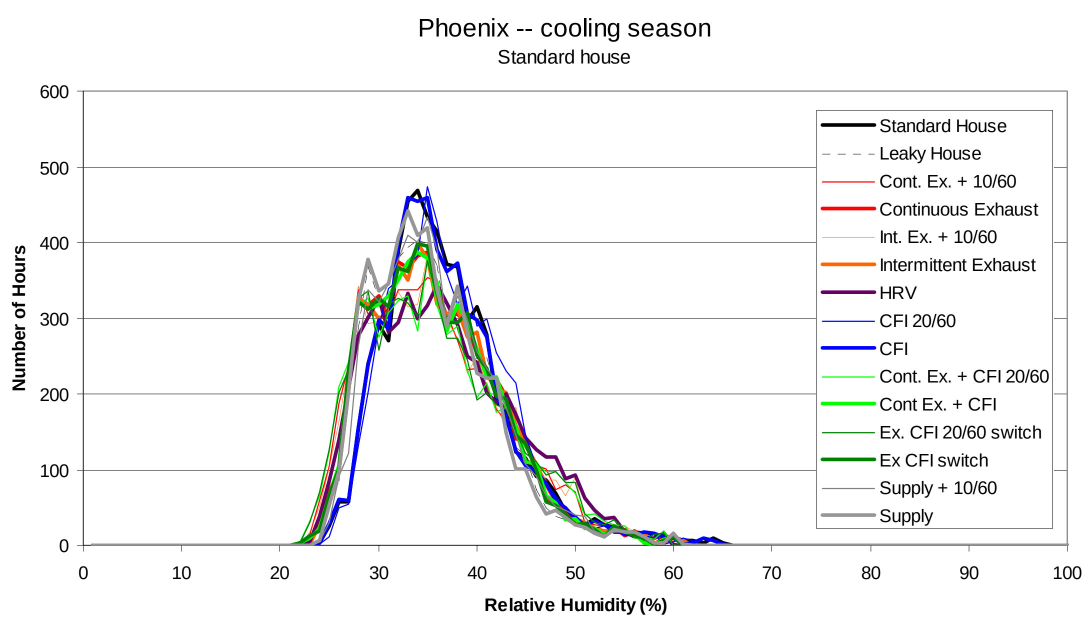


Figure A.14 Frequency distribution of hourly average indoor relative humidity (1% bins) for the cooling season in Phoenix

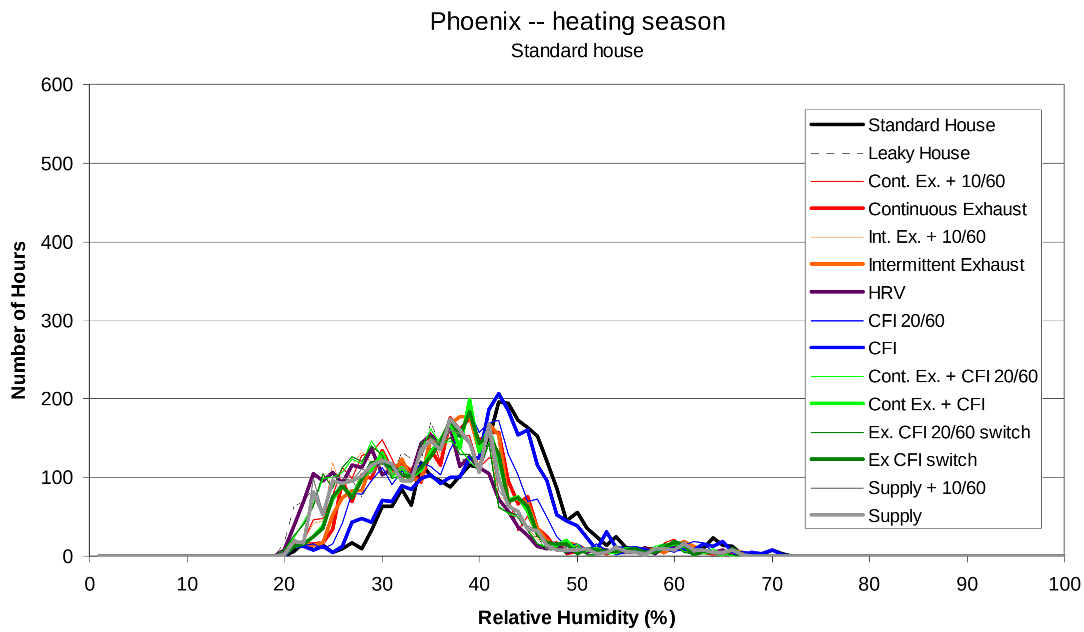


Figure A.15 Frequency distribution of hourly average indoor relative humidity (1% bins) for the heating season in Phoenix

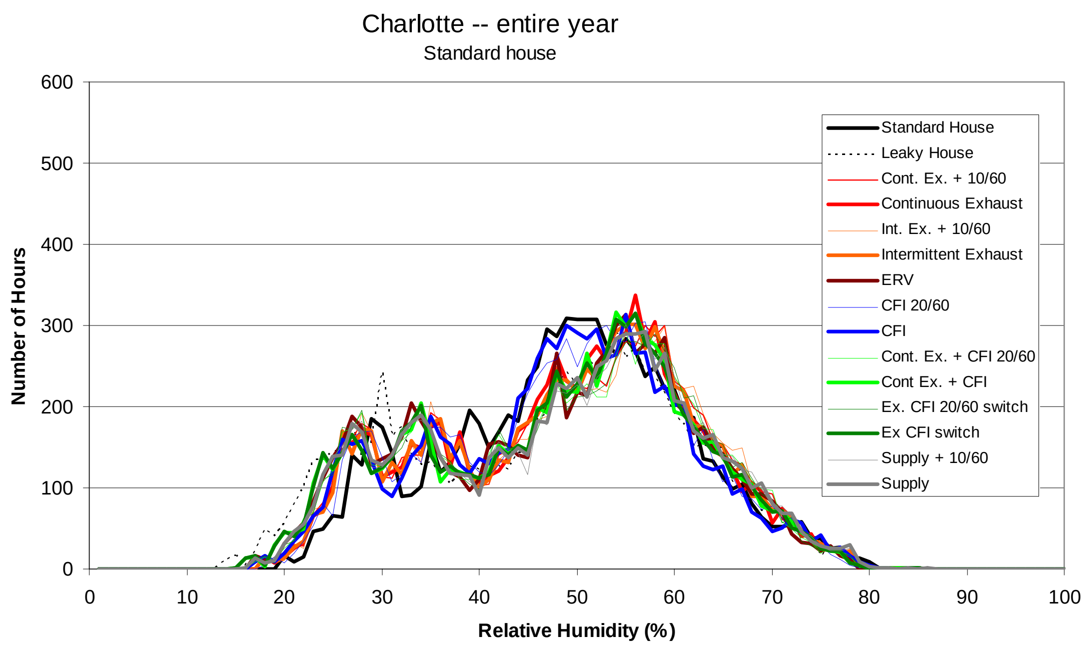


Figure A.16 Frequency distribution of hourly average indoor relative humidity (1% bins) for the entire year in Charlotte

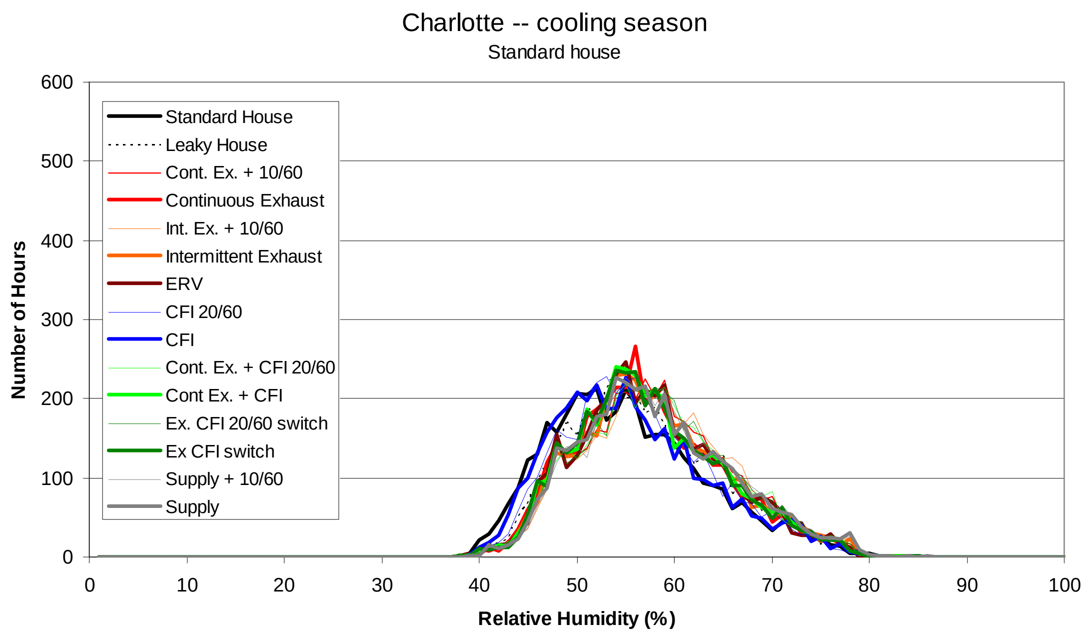


Figure A.17 Frequency distribution of hourly average indoor relative humidity (1% bins) for the cooling season in Charlotte

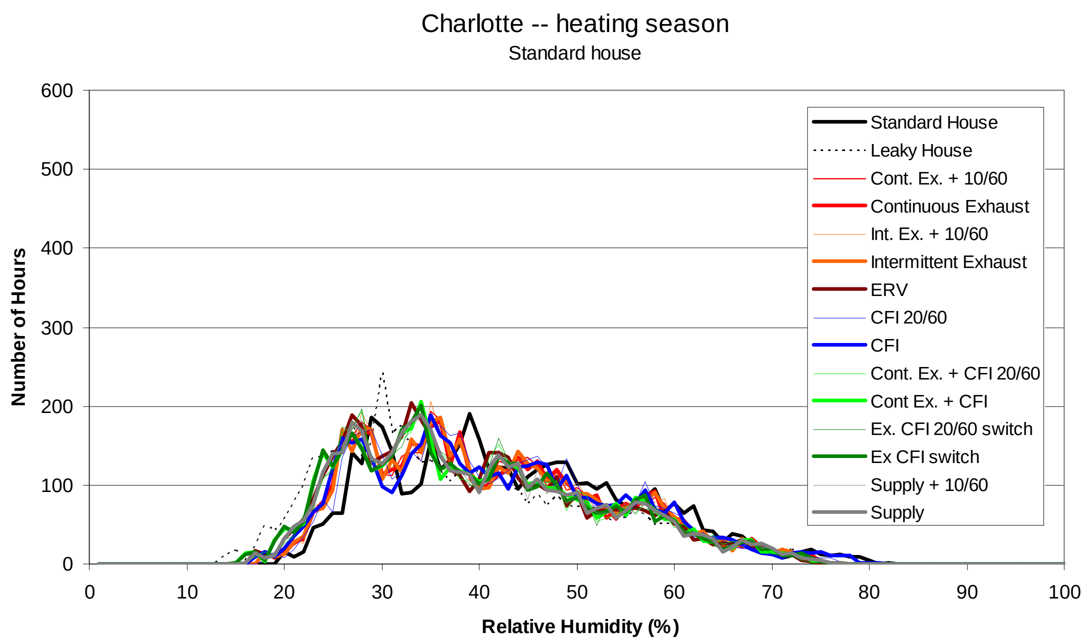


Figure A.18 Frequency distribution of hourly average indoor relative humidity (1% bins) for the heating season in Charlotte

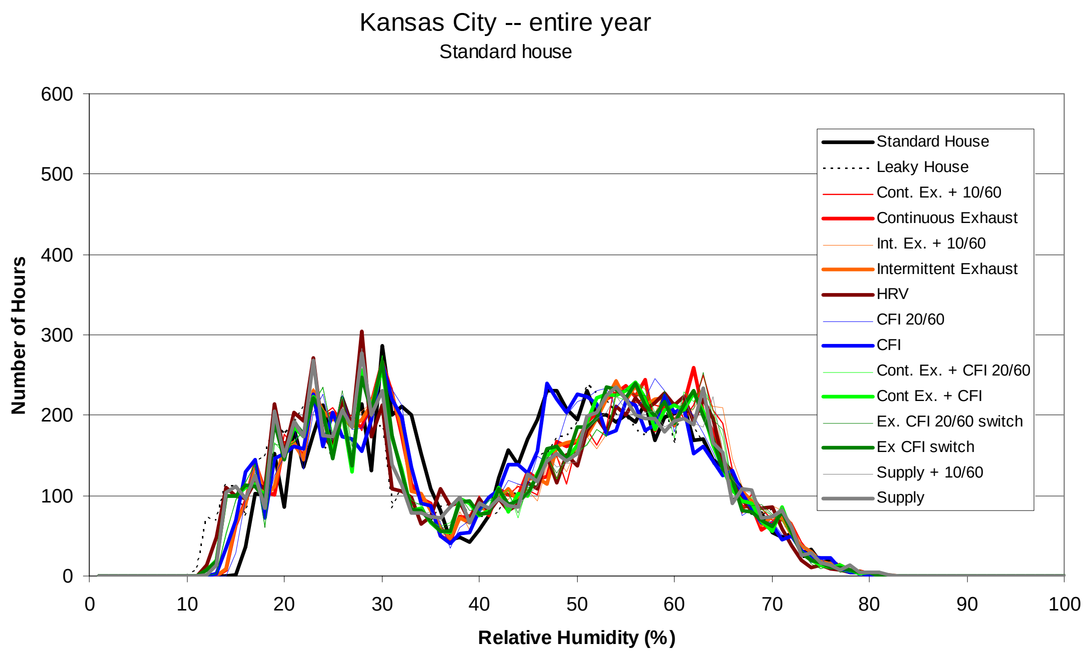


Figure A.19 Frequency distribution of hourly average indoor relative humidity (1% bins) for the entire year in Kansas City

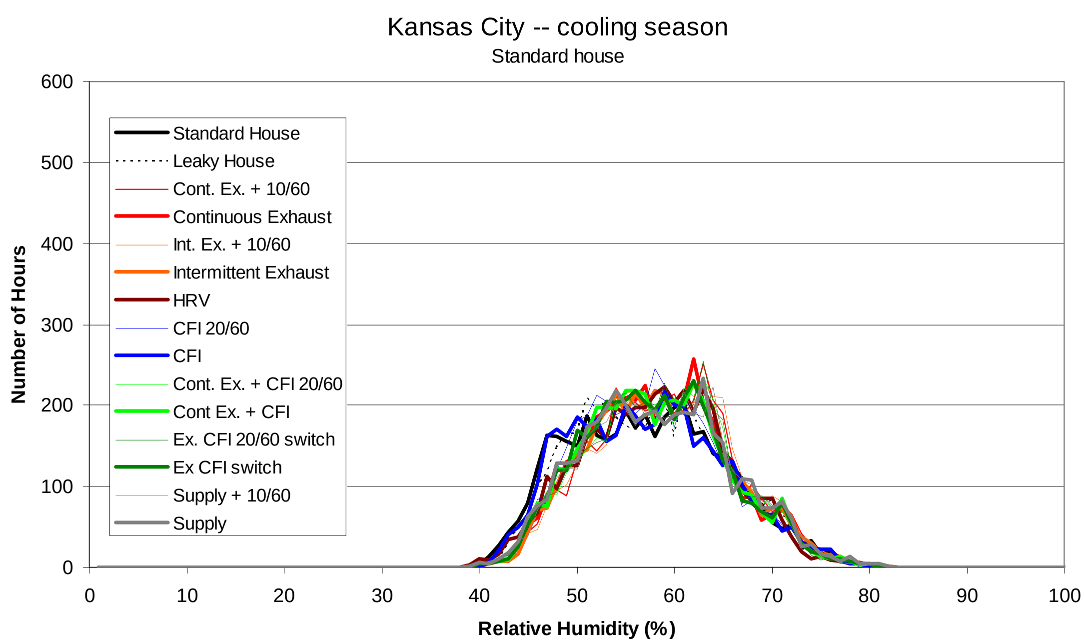


Figure A.20 Frequency distribution of hourly average indoor relative humidity (1% bins) for the cooling season in Kansas City

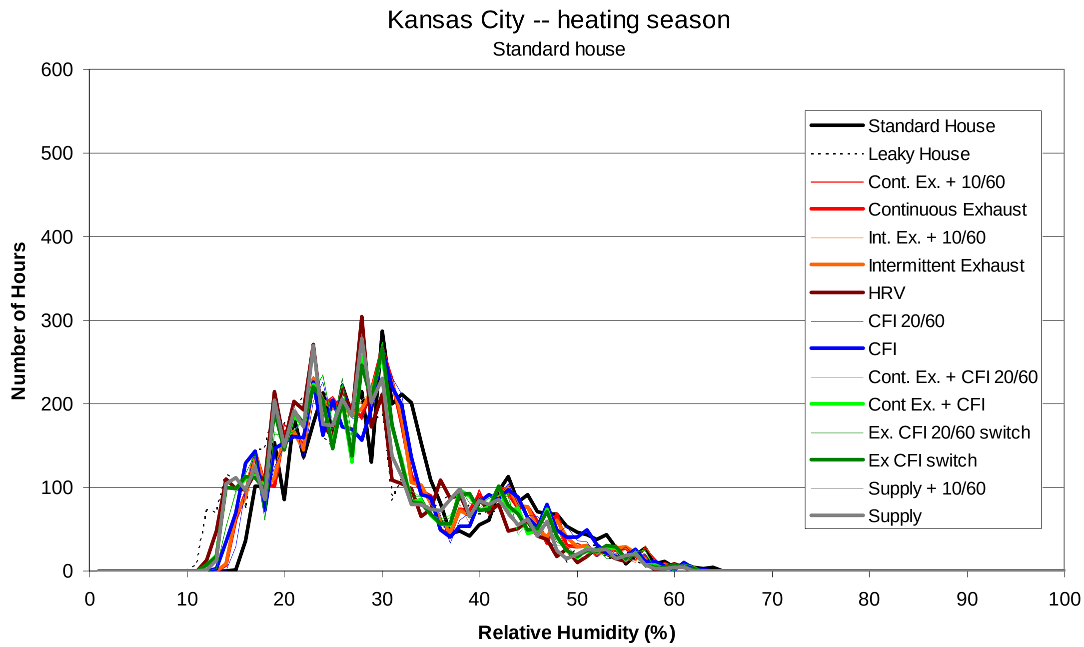


Figure A.21 Frequency distribution of hourly average indoor relative humidity (1% bins) for the heating season in Kansas City

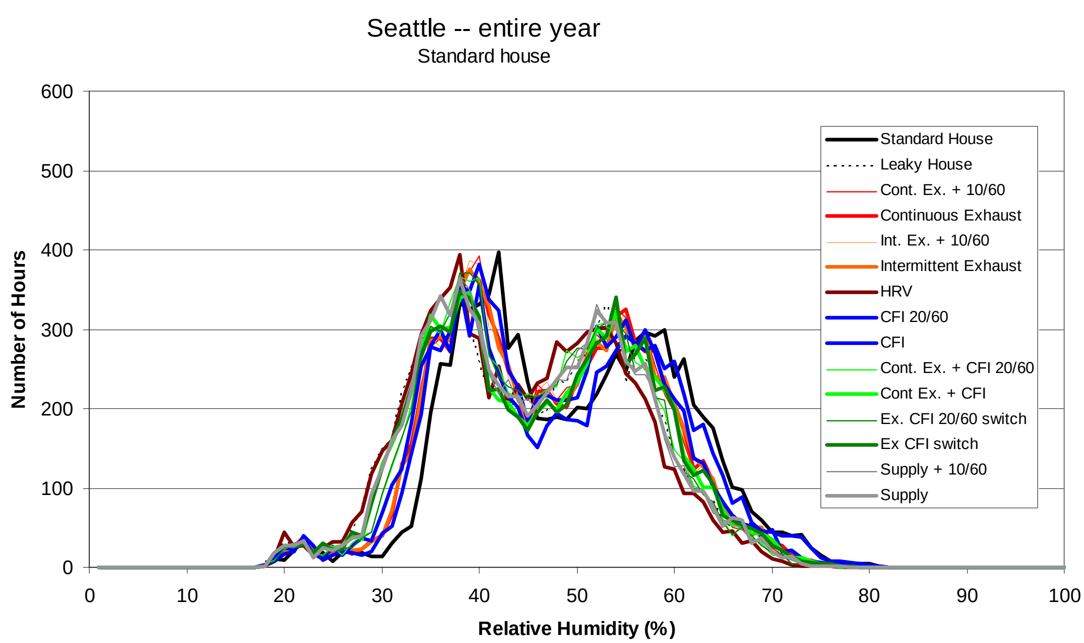


Figure A.22 Frequency distribution of hourly average indoor relative humidity (1% bins) for the entire year in Seattle

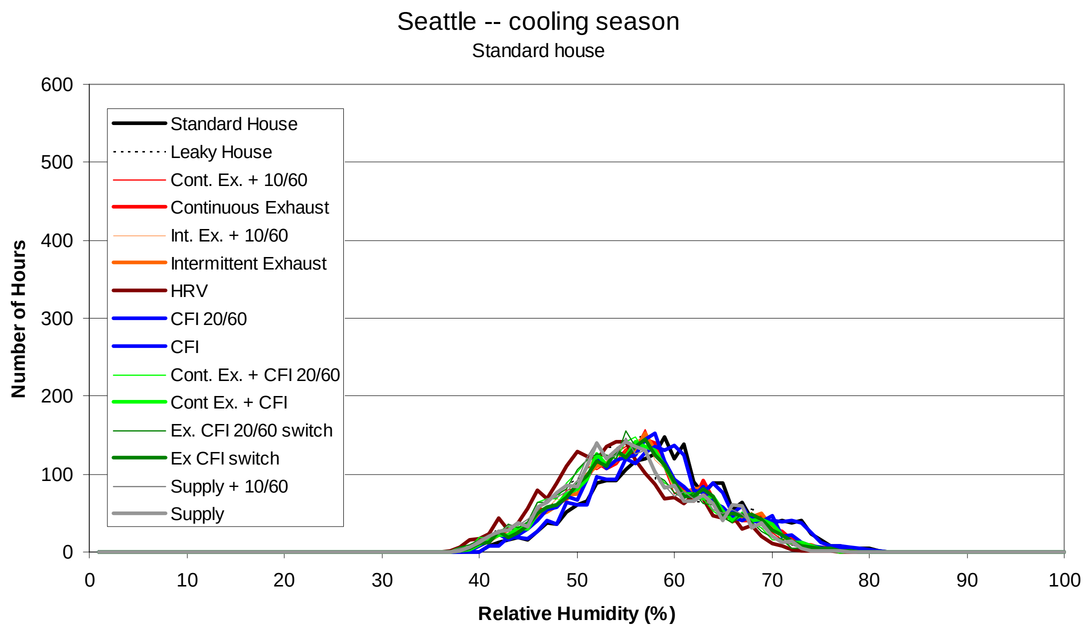


Figure A.23 Frequency distribution of hourly average indoor relative humidity (1% bins) for the cooling season in Seattle

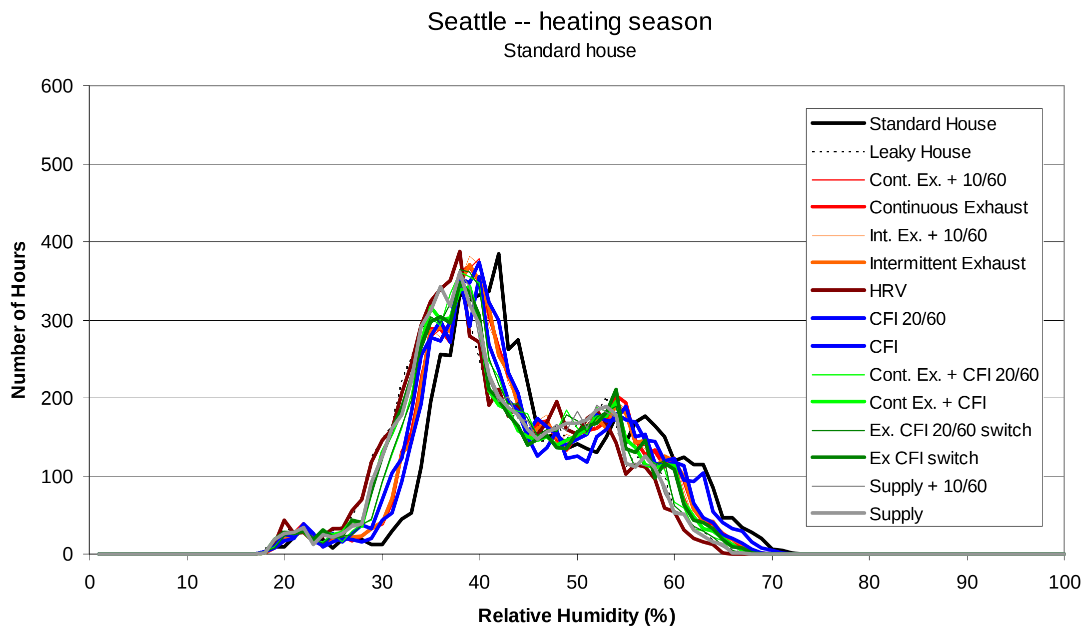


Figure A.24 Frequency distribution of hourly average indoor relative humidity (1% bins) for the heating season in Seattle

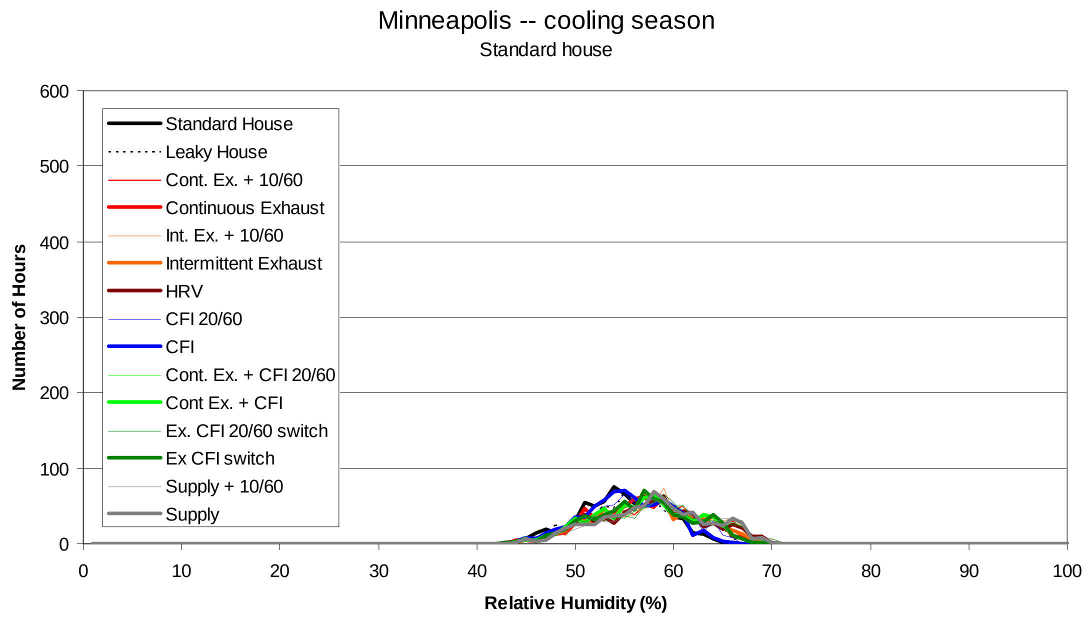


Figure A.25 Frequency distribution of hourly average indoor relative humidity (1% bins) for the cooling season in Minneapolis

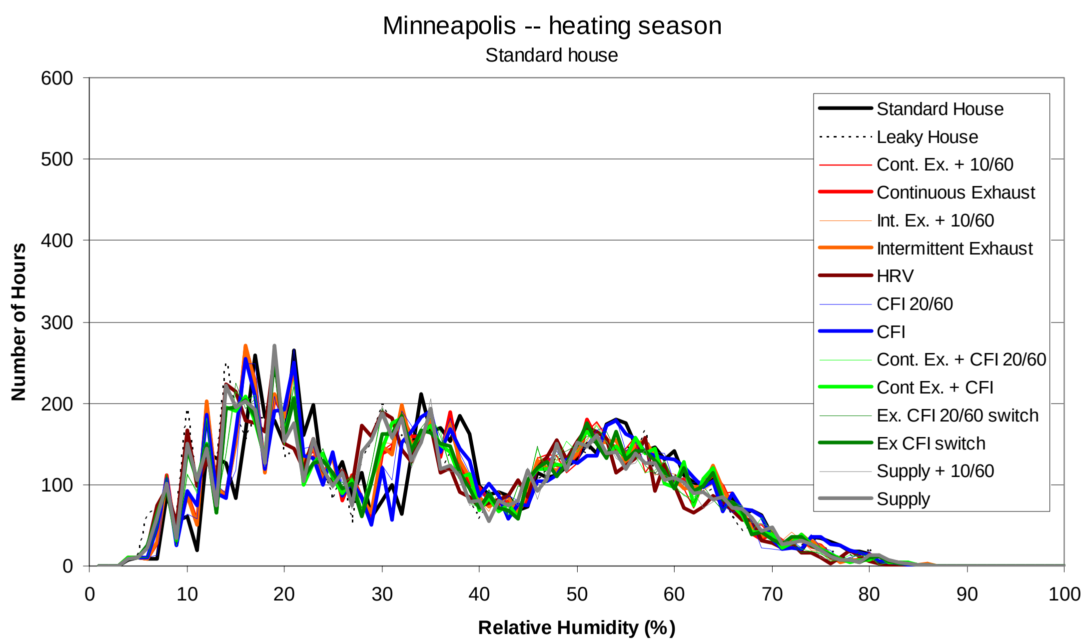


Figure A.26 Frequency distribution of hourly average indoor relative humidity (1% bins) for the heating season in Minneapolis

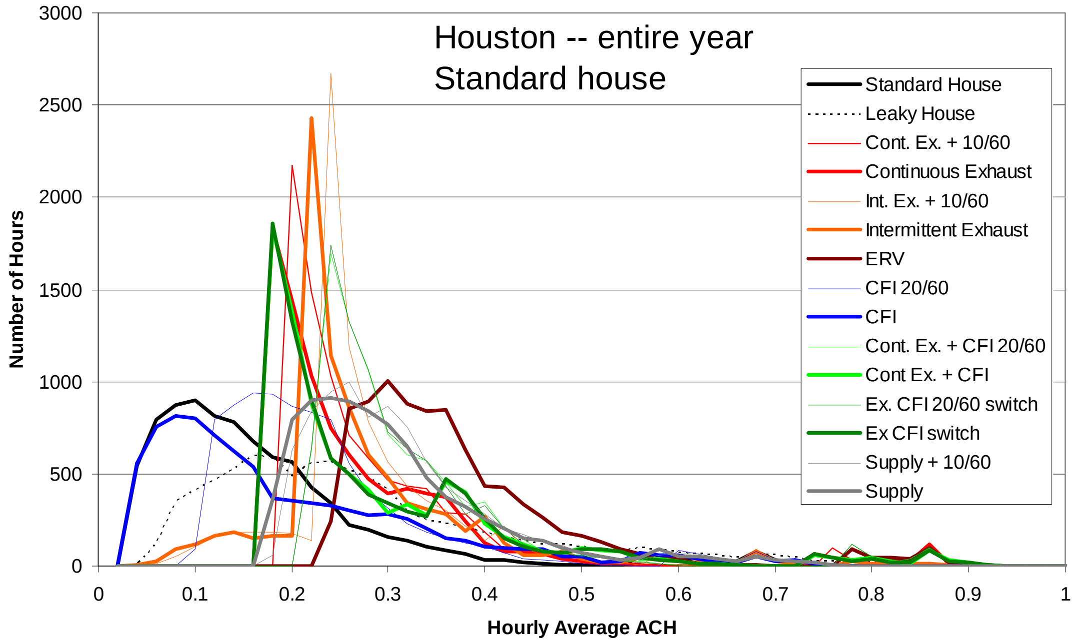


Figure A.27 Frequency distribution of hourly average air change rate (0.02 ach bins) for the entire year in Houston

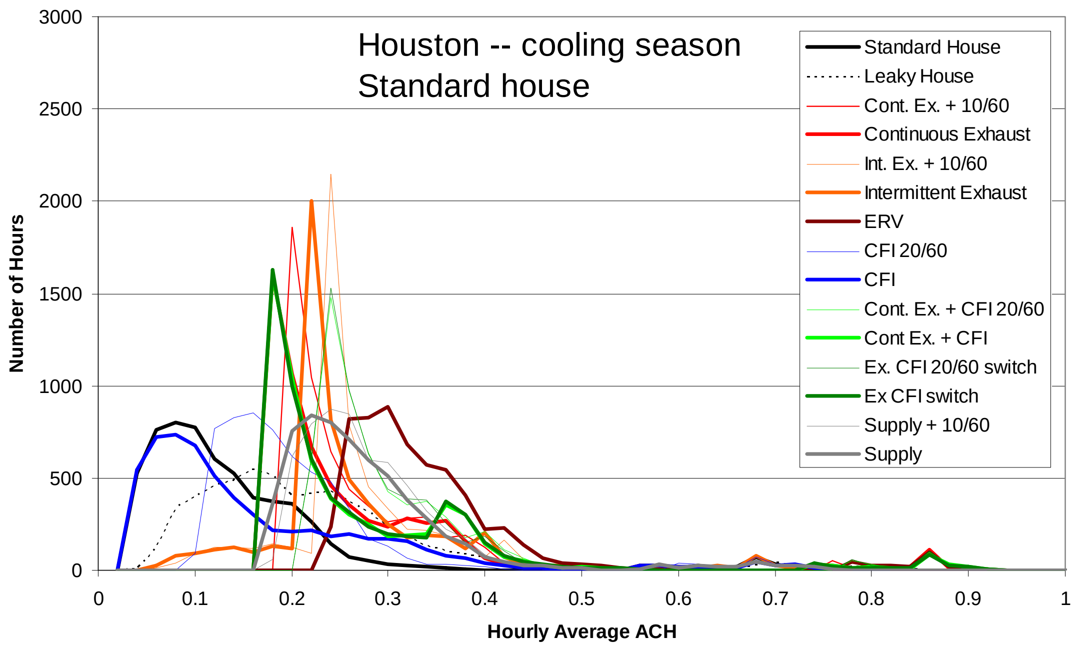


Figure A.28 Frequency distribution of hourly average air change rate (0.02 ach bins) for the cooling season in Houston

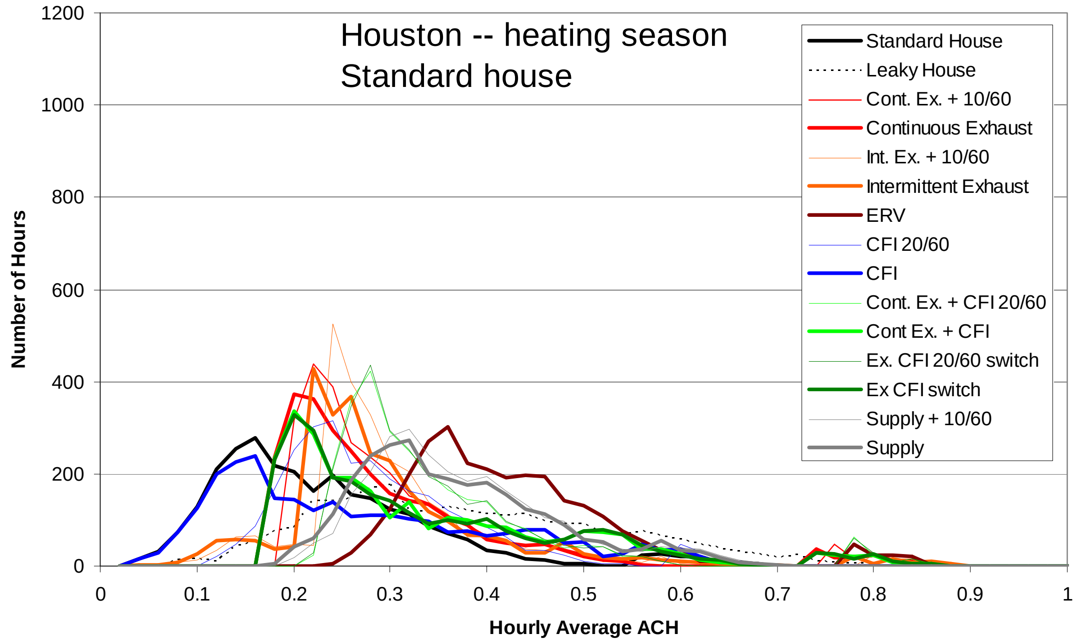


Figure A.29 Frequency distribution of hourly average air change rate (0.02 ach bins) for the heating season in Houston

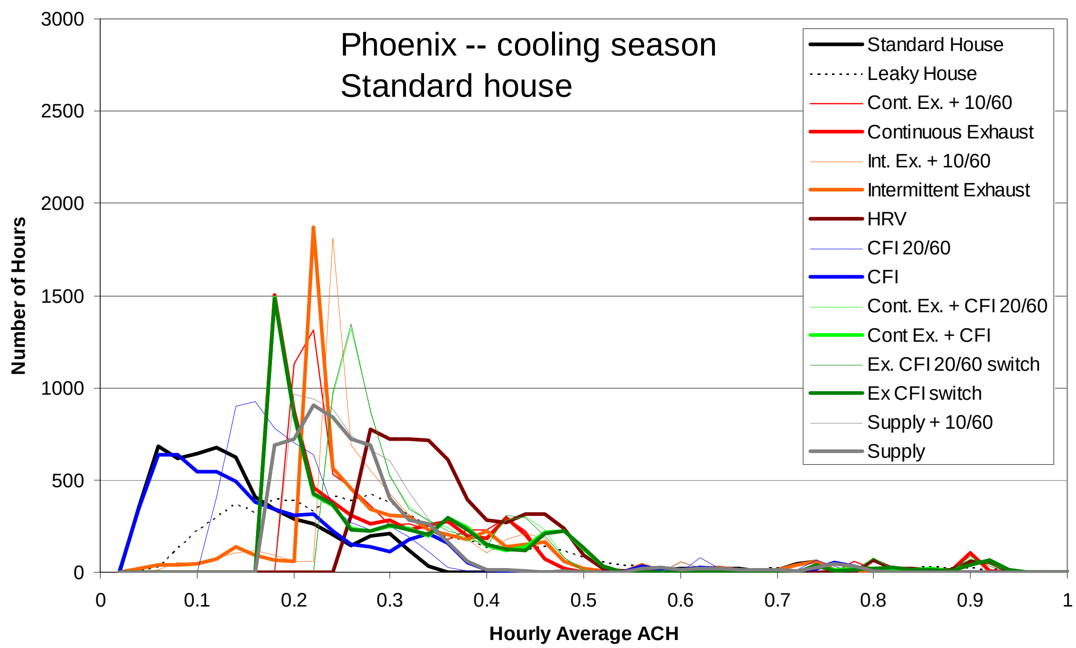


Figure A.30 Frequency distribution of hourly average air change rate (0.02 ach bins) for the cooling season in Phoenix

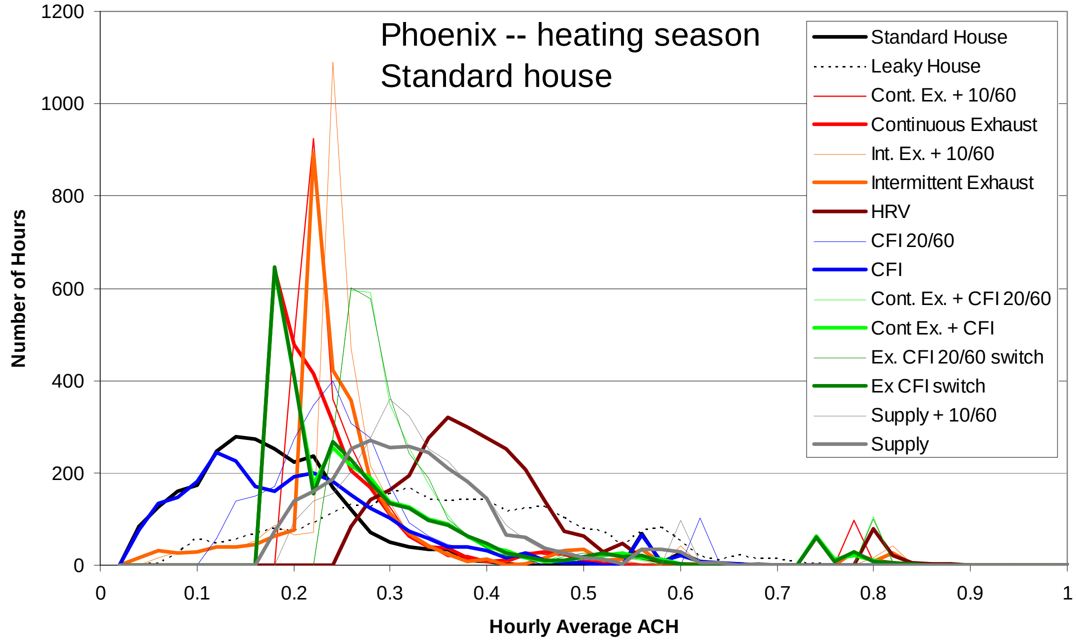


Figure A.31 Frequency distribution of hourly average air change rate (0.02 ach bins) for the heating season in Phoenix

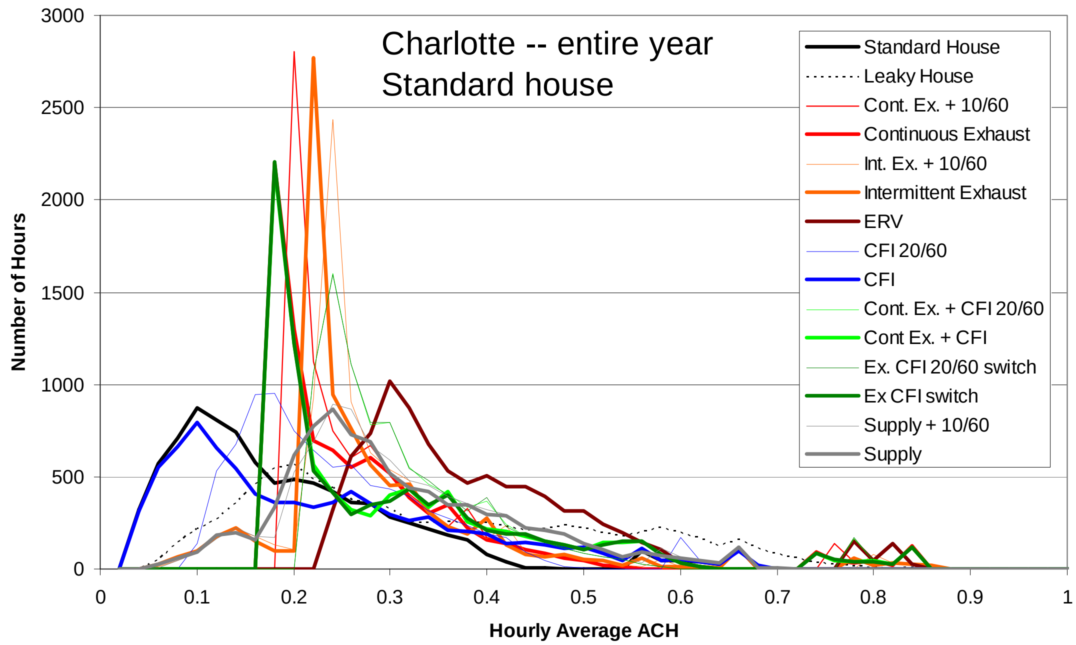


Figure A.32 Frequency distribution of hourly average air change rate (0.02 ach bins) for the entire year in Charlotte

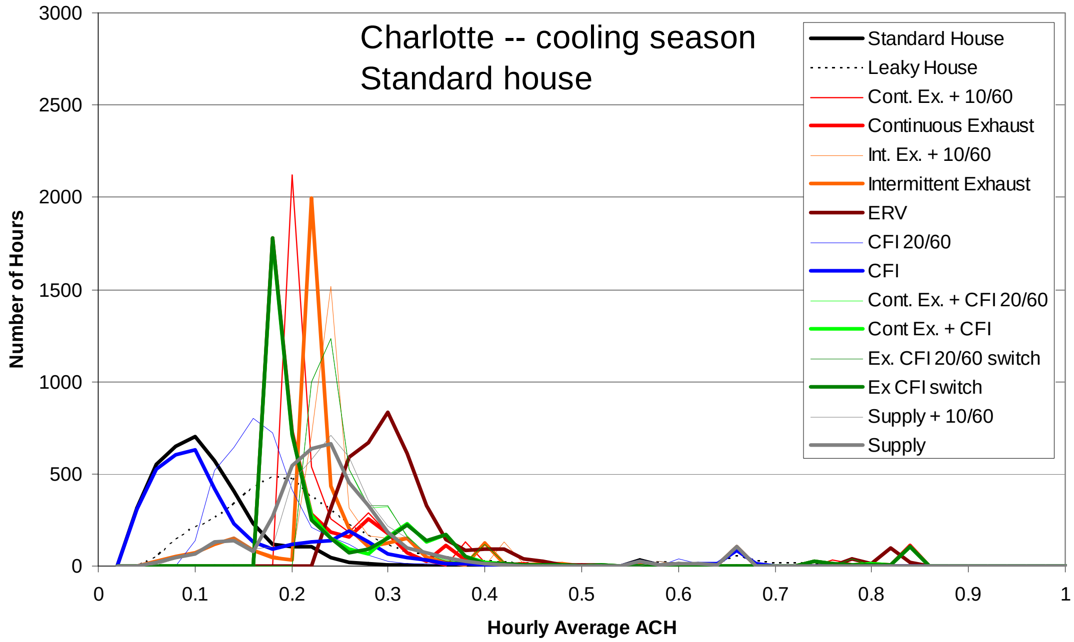


Figure A.33 Frequency distribution of hourly average air change rate (0.02 ach bins) for the cooling season in Charlotte

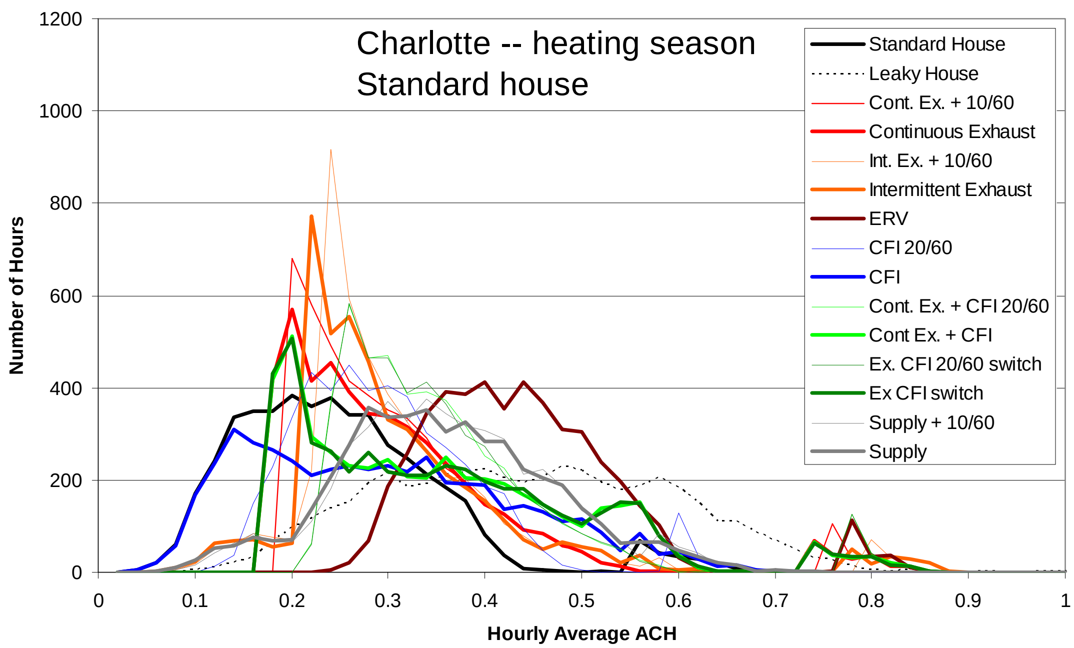


Figure A.34 Frequency distribution of hourly average air change rate (0.02 ach bins) for the heating season in Charlotte

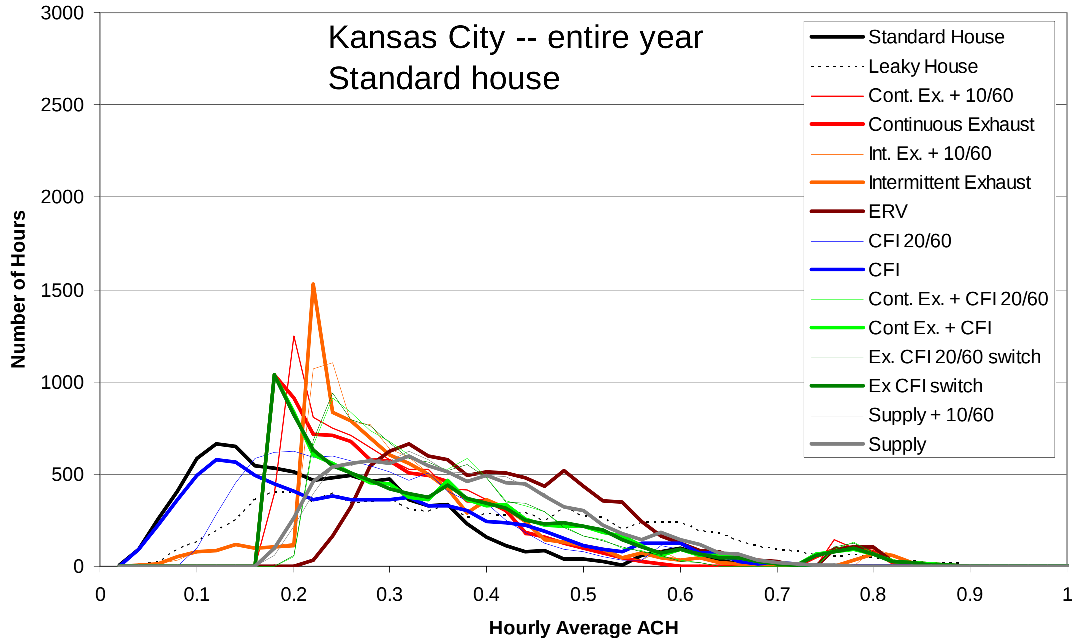


Figure A.35 Frequency distribution of hourly average air change rate (0.02 ach bins) for the entire year in Kansas City

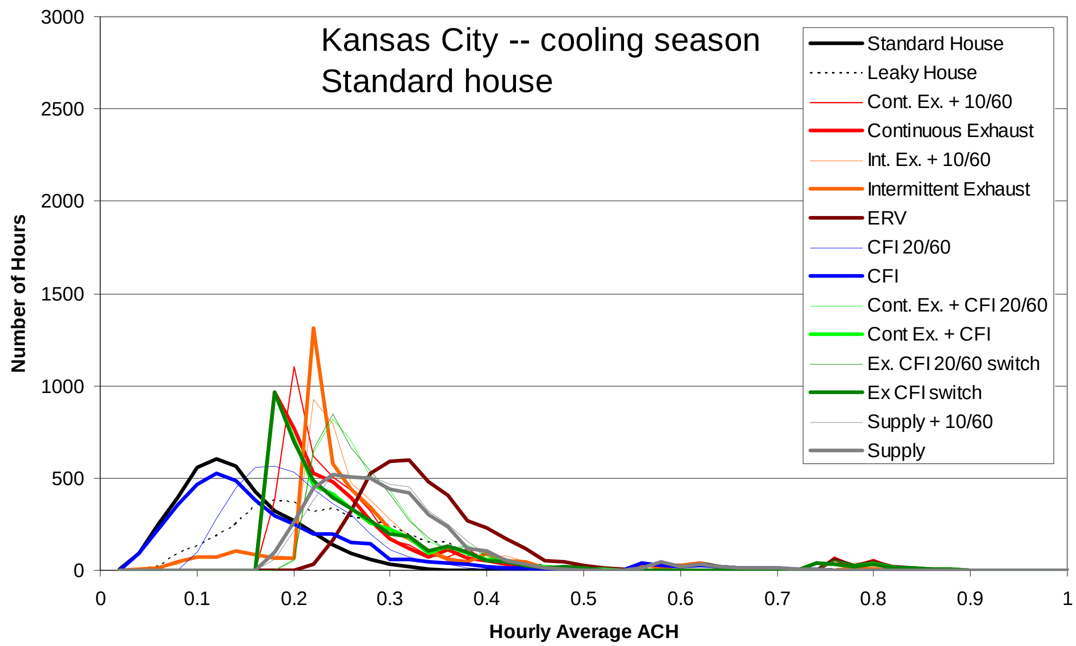


Figure A.36 Frequency distribution of hourly average air change rate (0.02 ach bins) for the cooling season in Kansas City

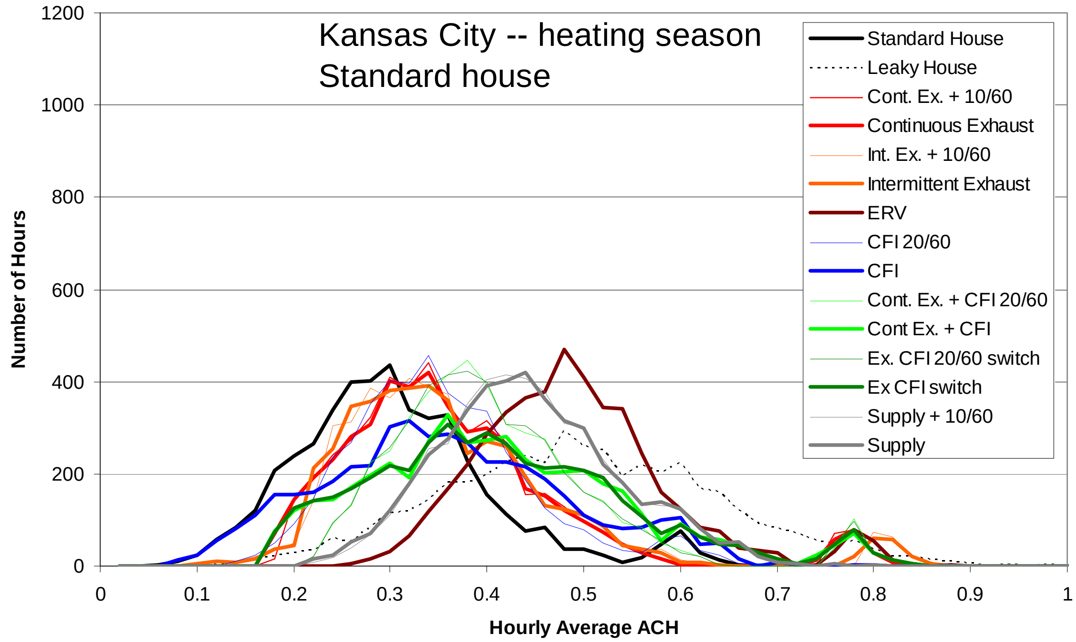


Figure A.37 Frequency distribution of hourly average air change rate (0.02 ach bins) for the heating season in Kansas City

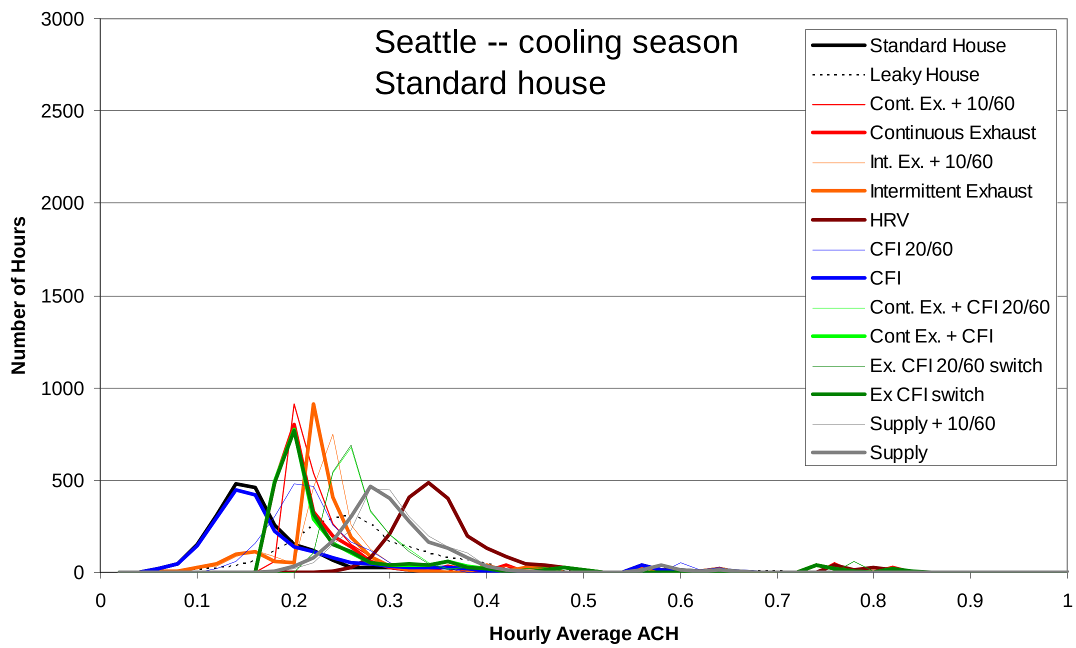


Figure A.38 Frequency distribution of hourly average air change rate (0.02 ach bins) for the cooling season in Seattle

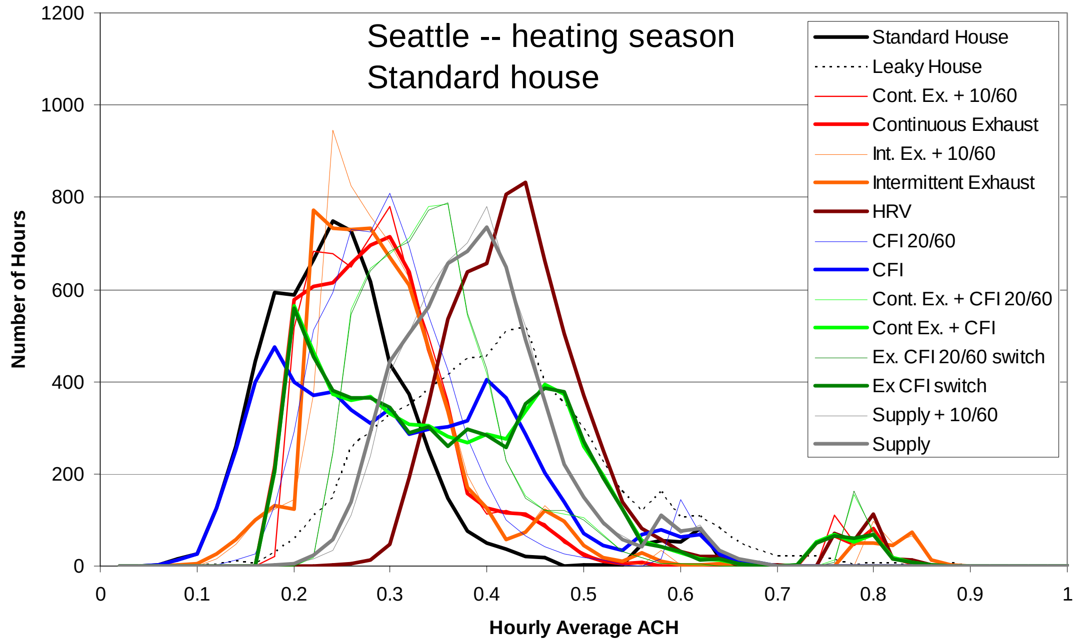


Figure A.39 Frequency distribution of hourly average air change rate (0.02 ach bins) for the heating season in Seattle

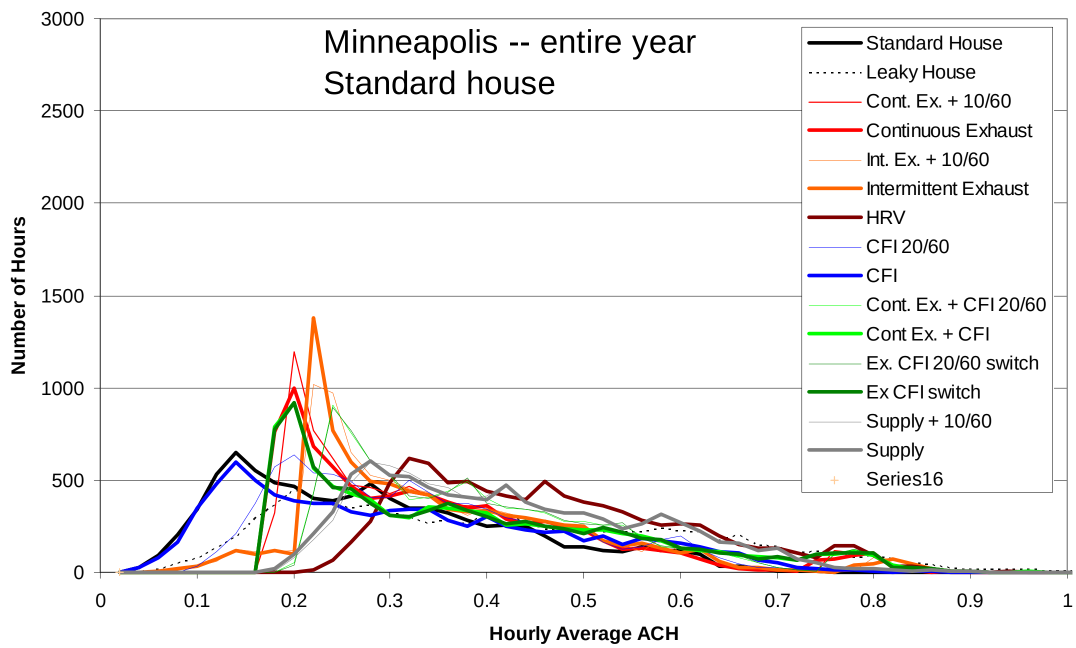


Figure A.40 Frequency distribution of hourly average air change rate (0.02 ach bins) for the entire year in Minneapolis

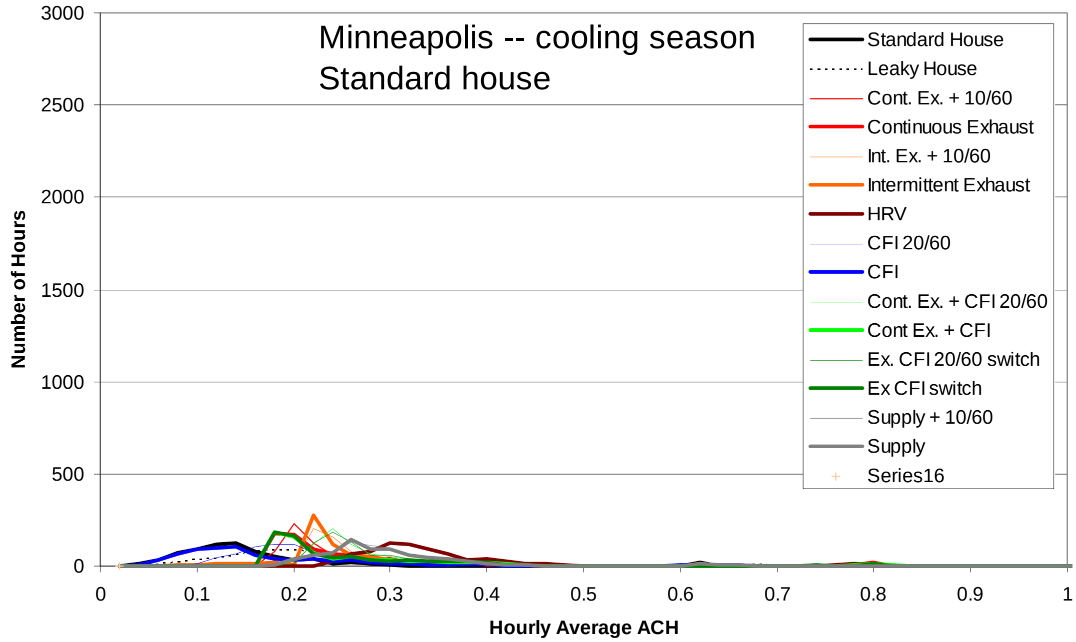


Figure A.41 Frequency distribution of hourly average air change rate (0.02 ach bins) for the cooling season in Minneapolis

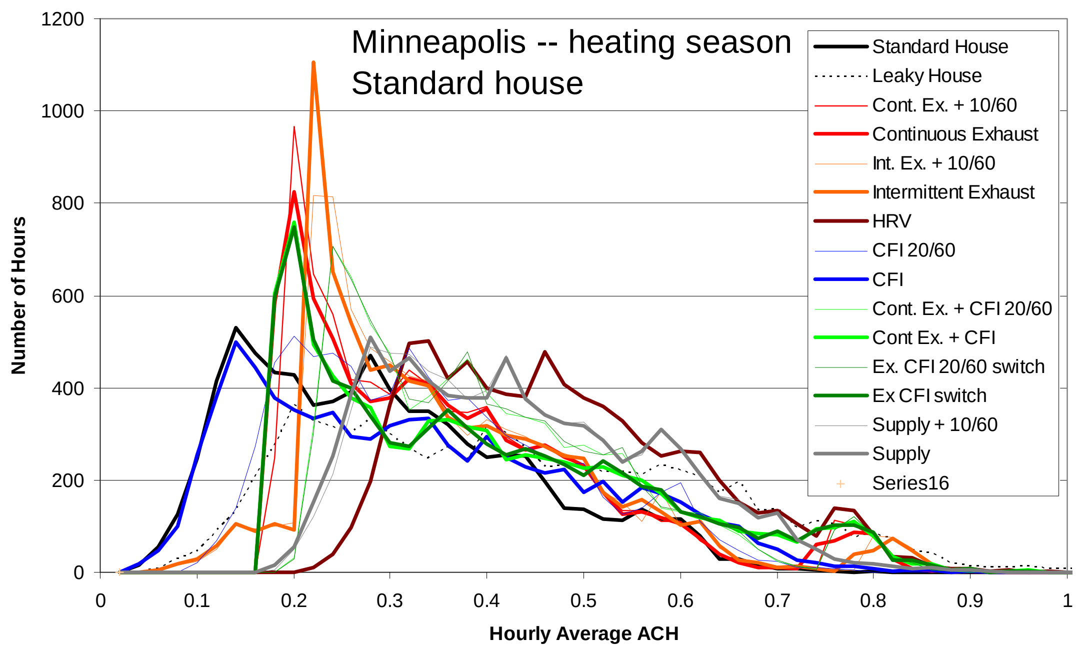


Figure A.42 Frequency distribution of hourly average air change rate (0.02 ach bins) for the heating season in Minneapolis

APPENDIX B

Additional plots from simulation results for the Higher-Performance house

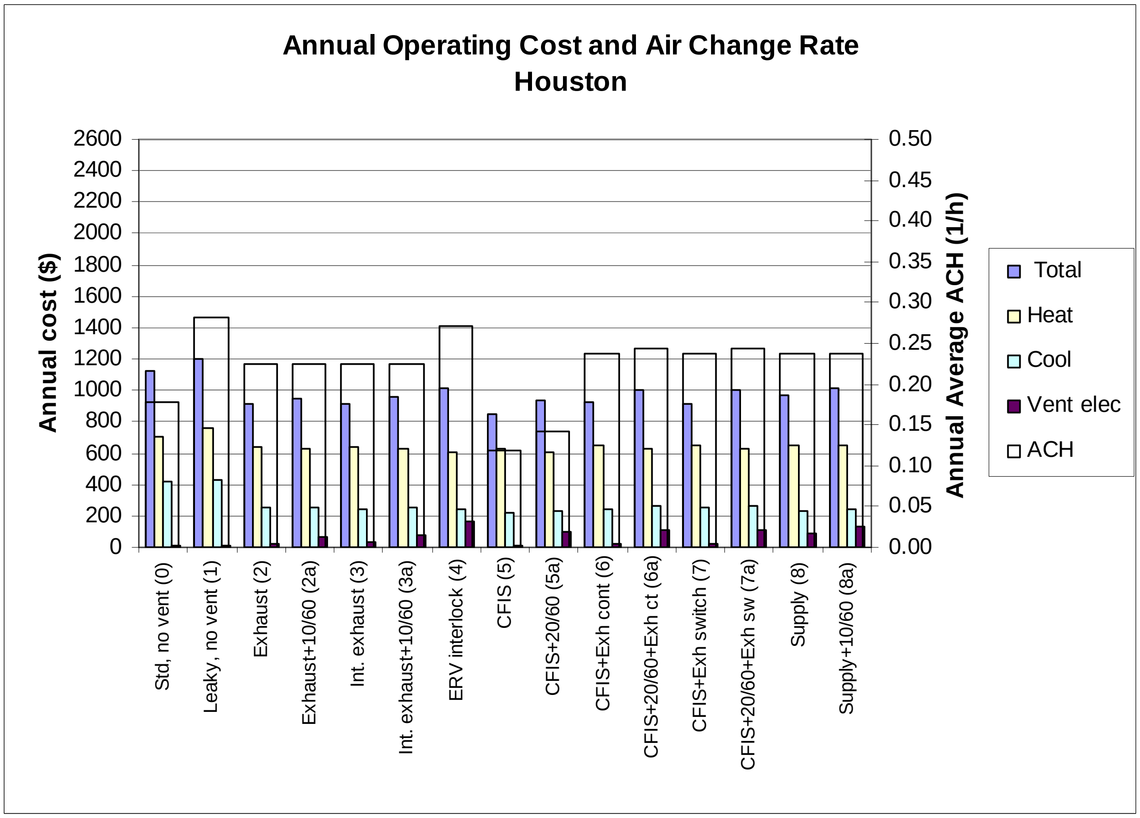


Figure B.1 Annual operating cost and annual average air change rate for Houston

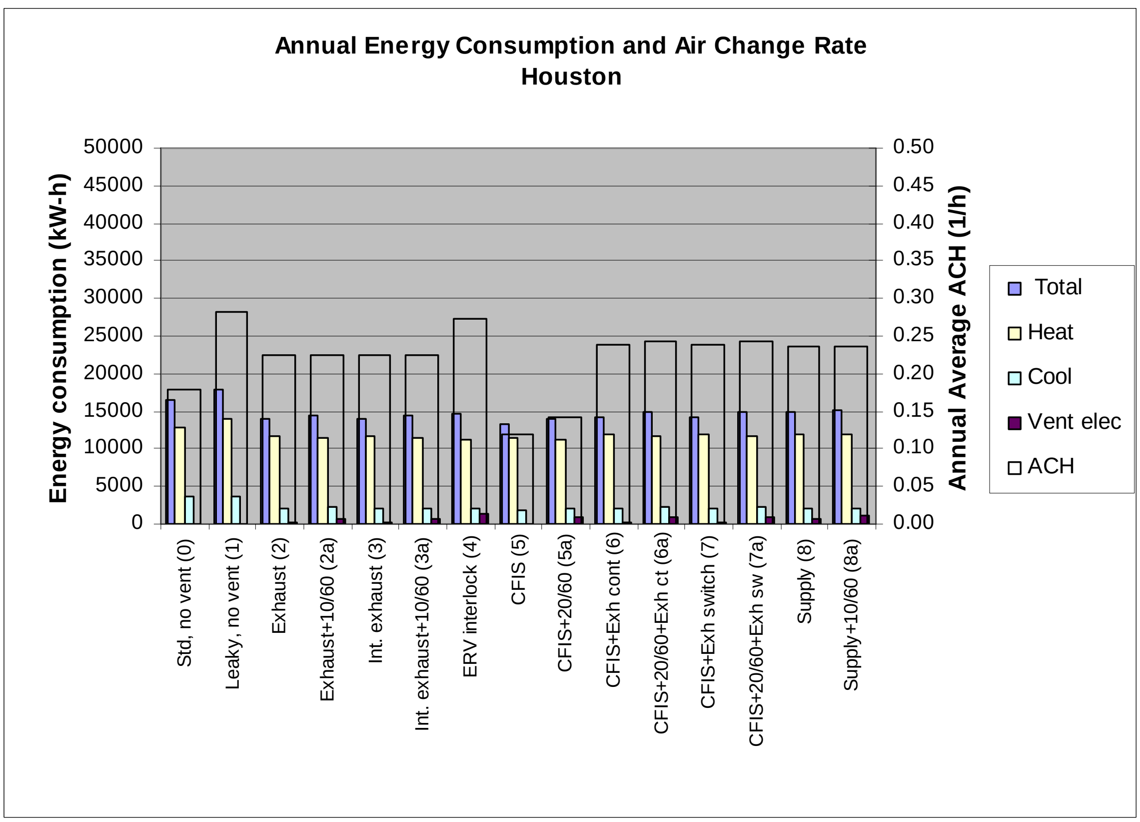


Figure B.2 Annual energy consumption and annual average air change rate for Houston

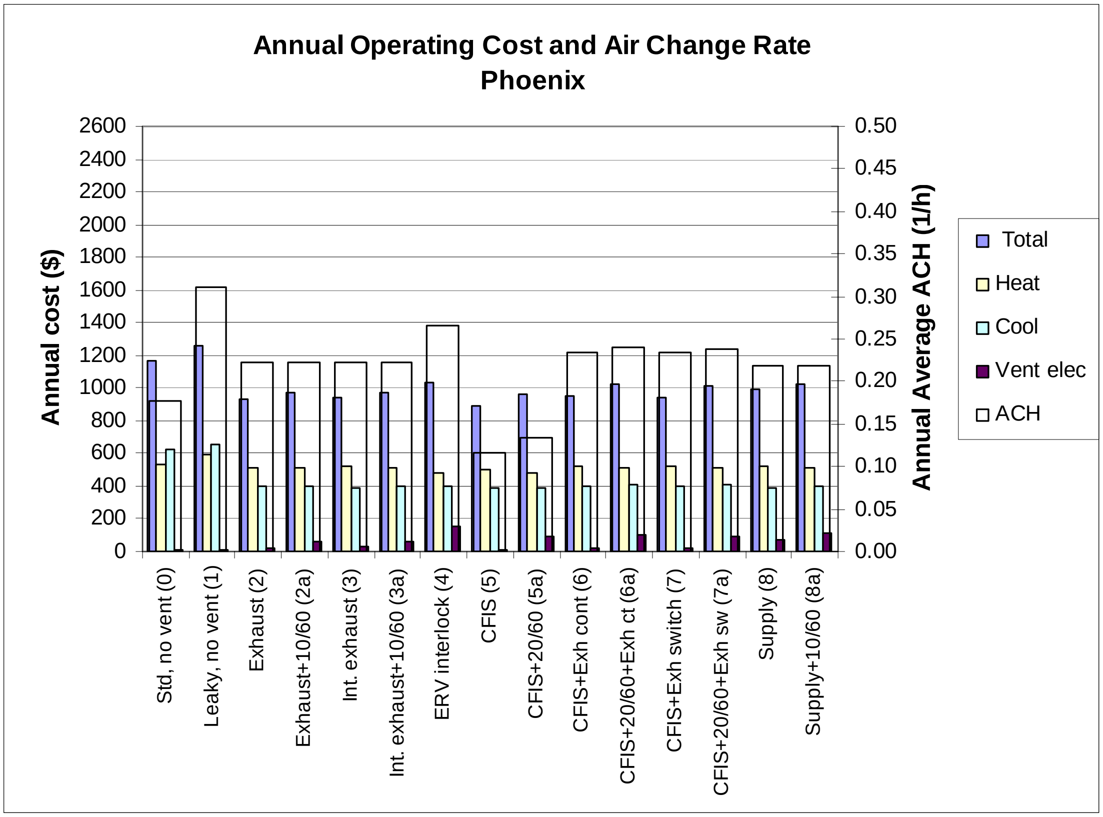


Figure B.3 Annual operating cost and annual average air change rate for Phoenix

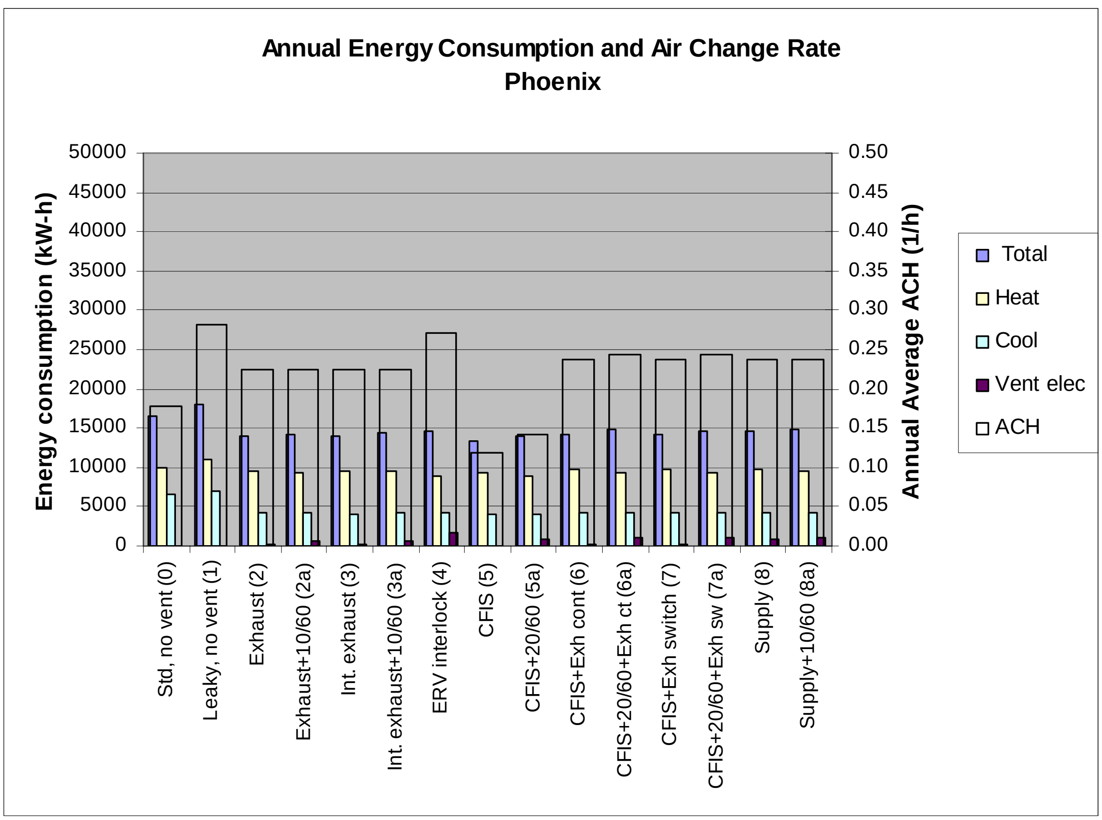


Figure B.4 Annual energy consumption and annual average air change rate for Phoenix

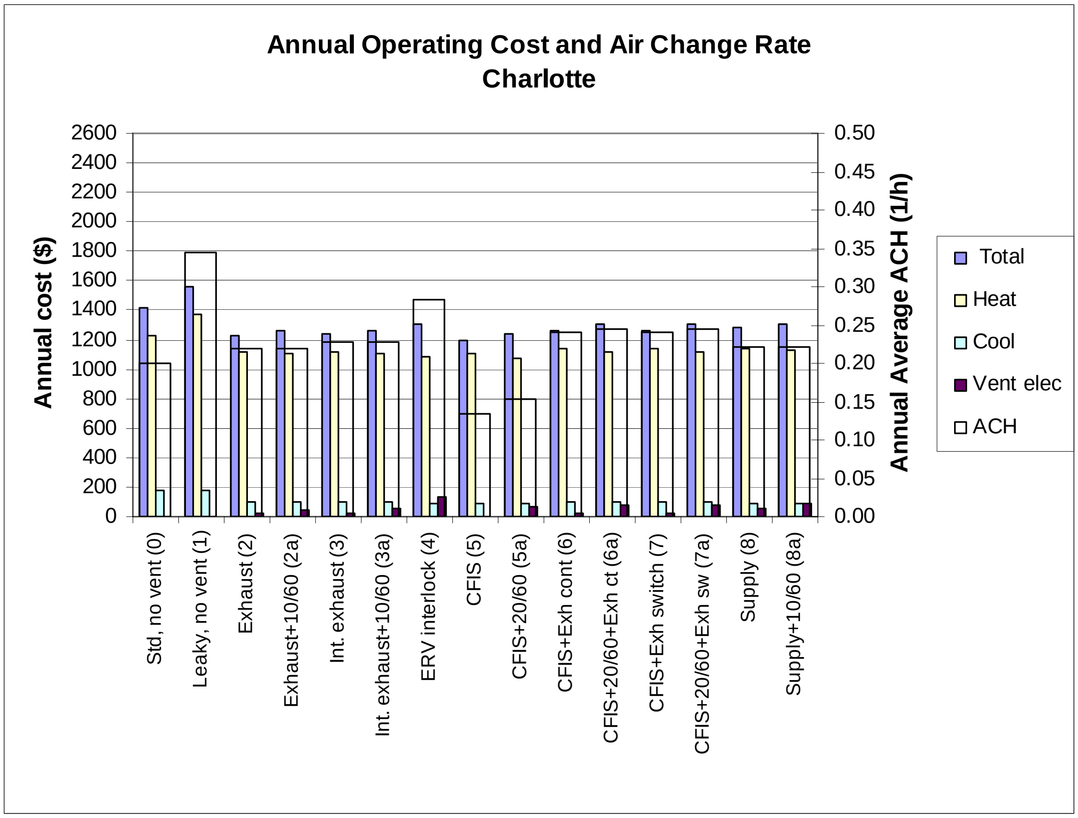


Figure B.5 Annual operating cost and annual average air change rate for Charlotte

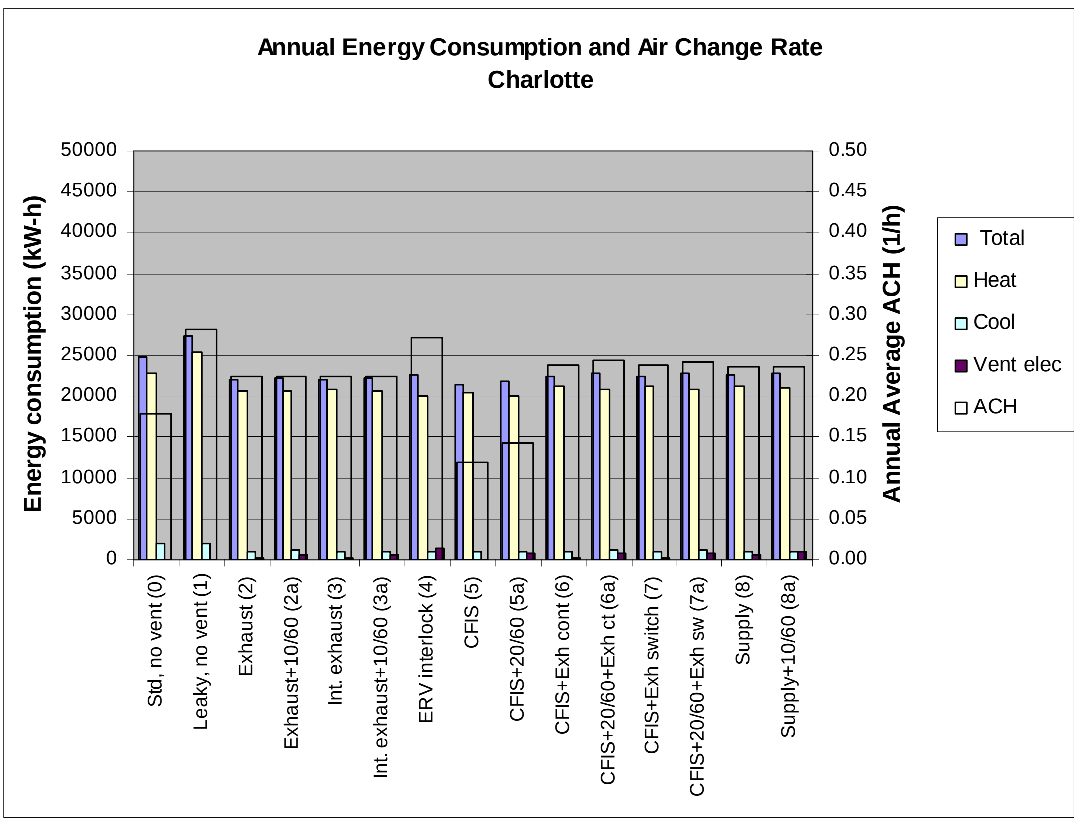


Figure B.6 Annual energy consumption and annual average air change rate for Charlotte

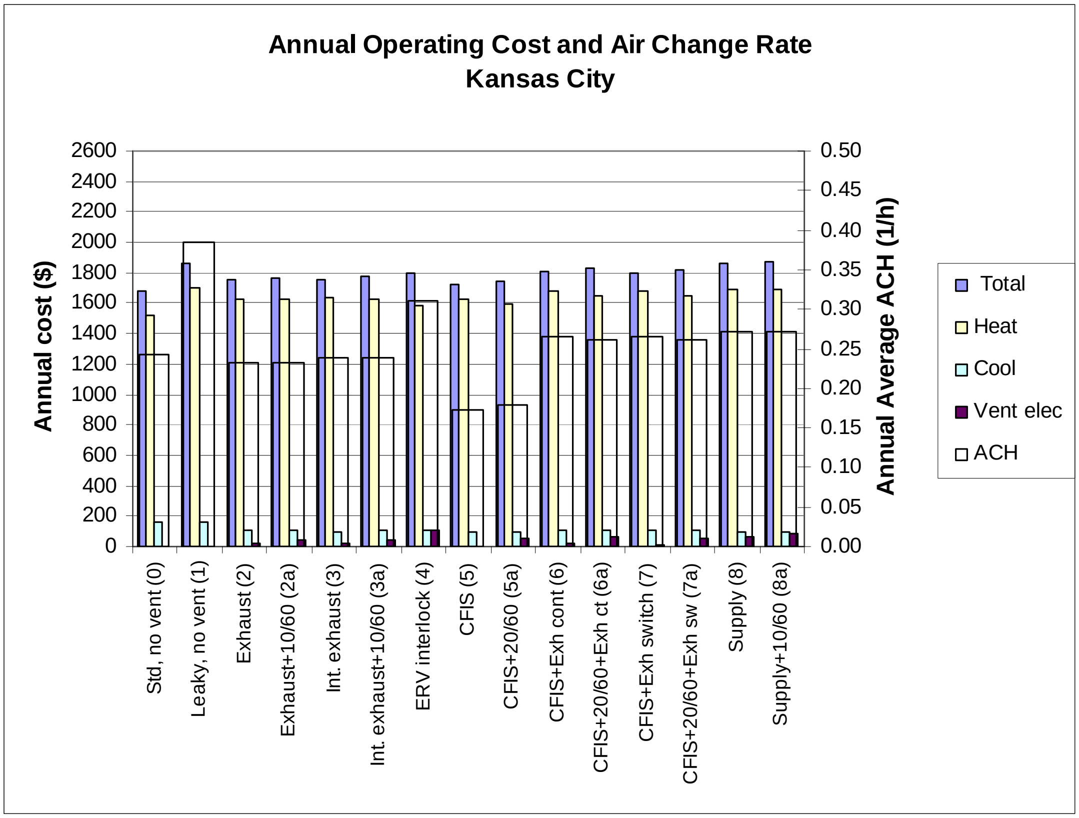


Figure B.7 Annual operating cost and annual average air change rate for Kansas City

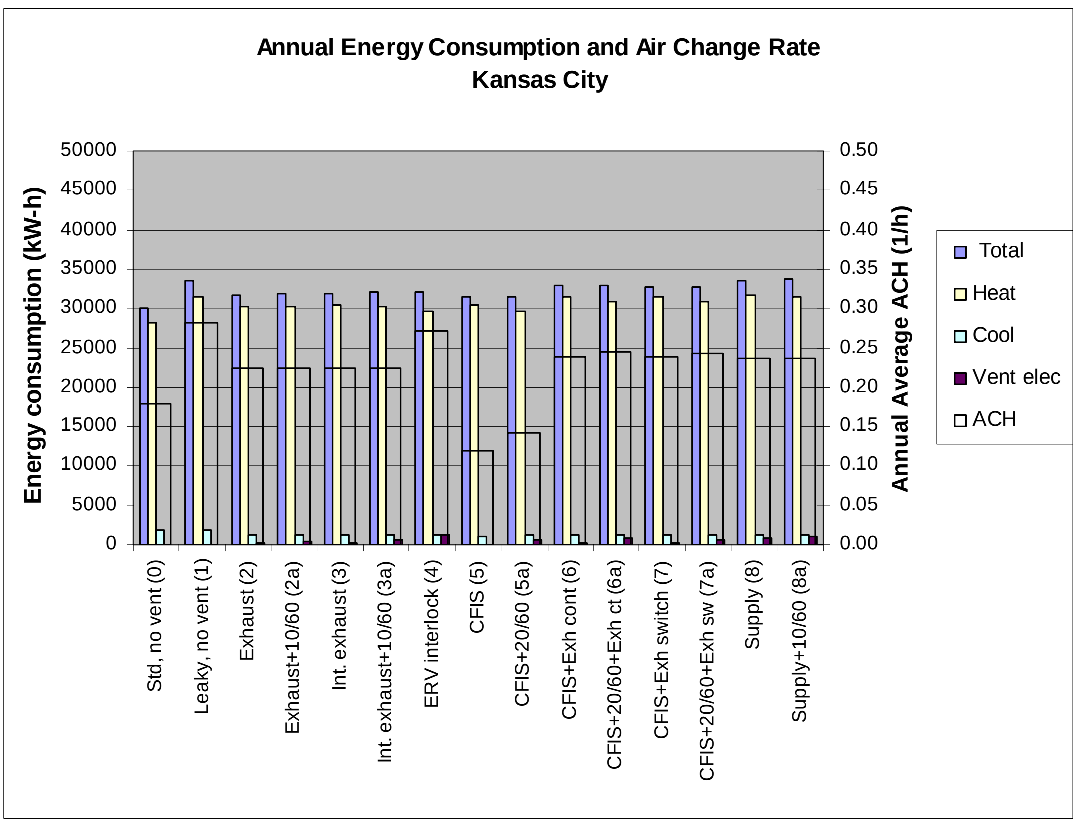


Figure B.8 Annual energy consumption and annual average air change rate for Kansas City

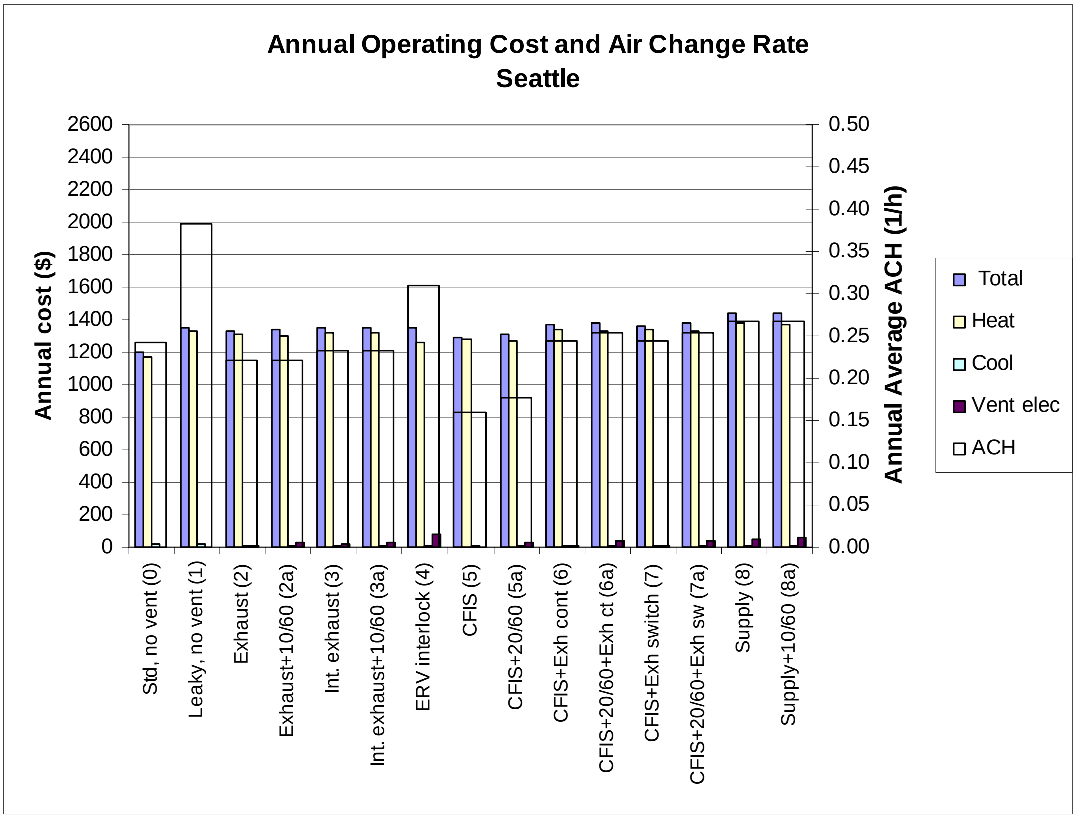


Figure B.9 Annual operating cost and annual average air change rate for Seattle

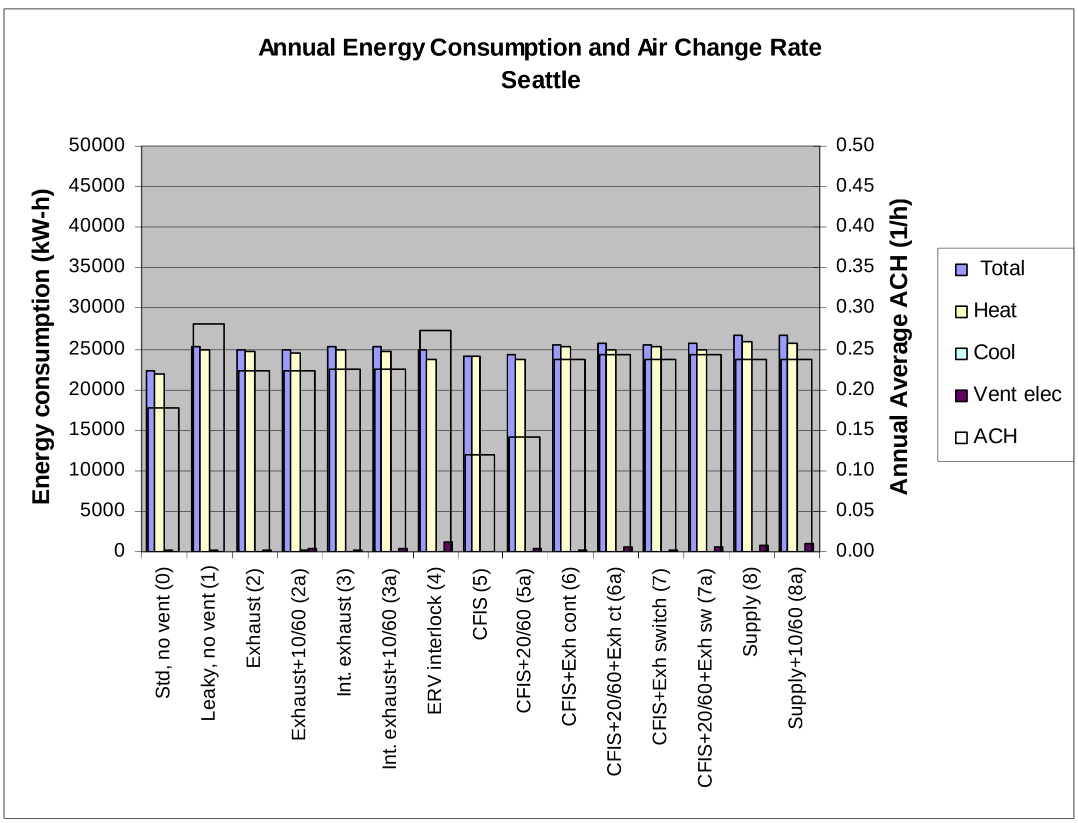


Figure B.10 Annual energy consumption and annual average air change rate for Seattle

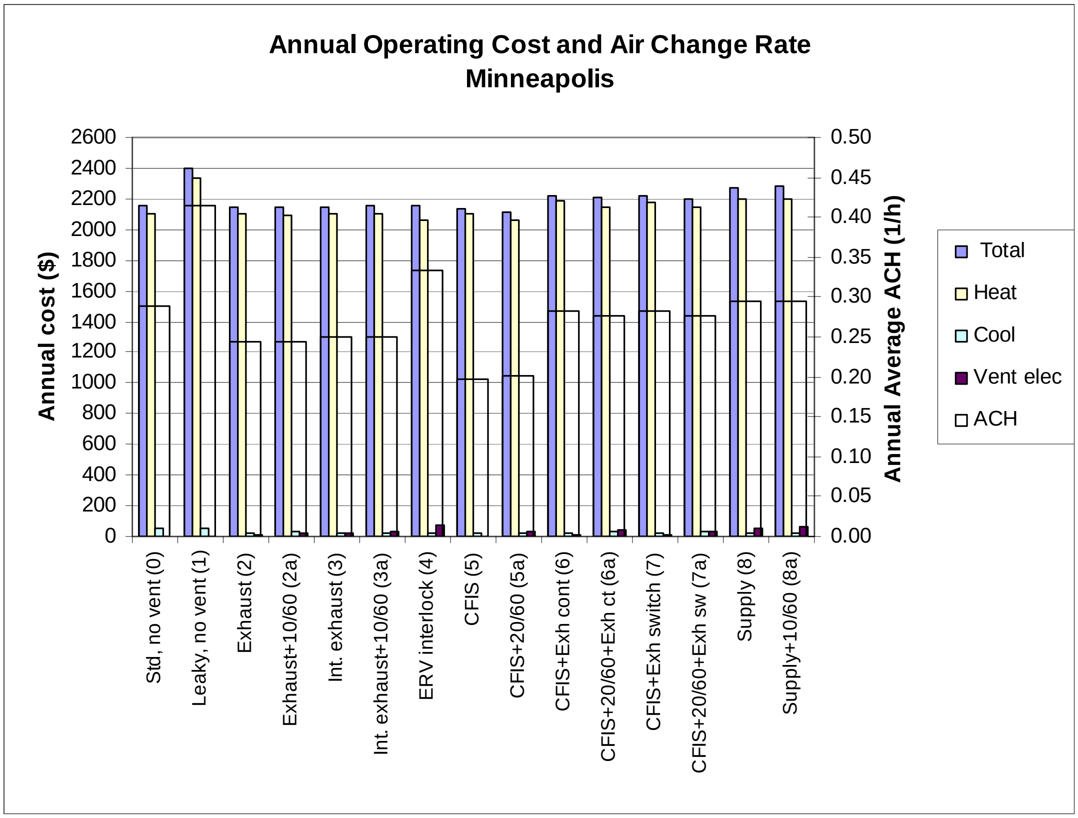


Figure B.11 Annual operating cost and annual average air change rate for Minneapolis

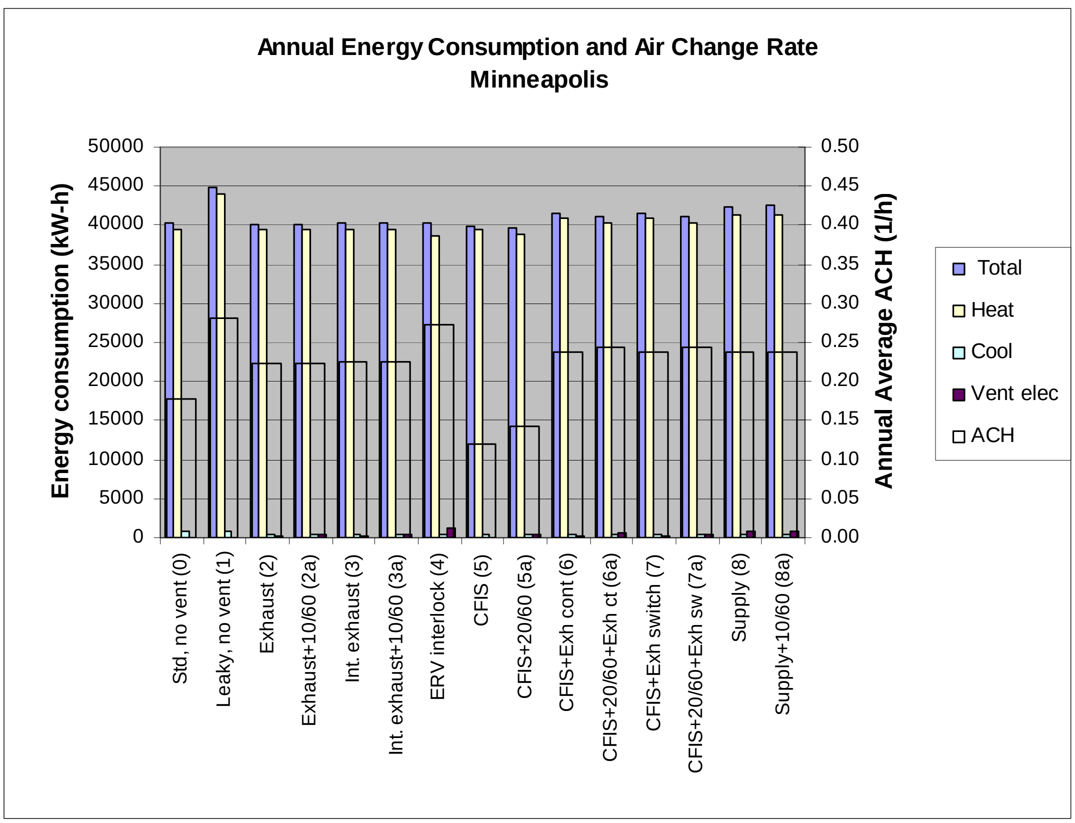


Figure B.12 Annual energy consumption and annual average air change rate for Minneapolis

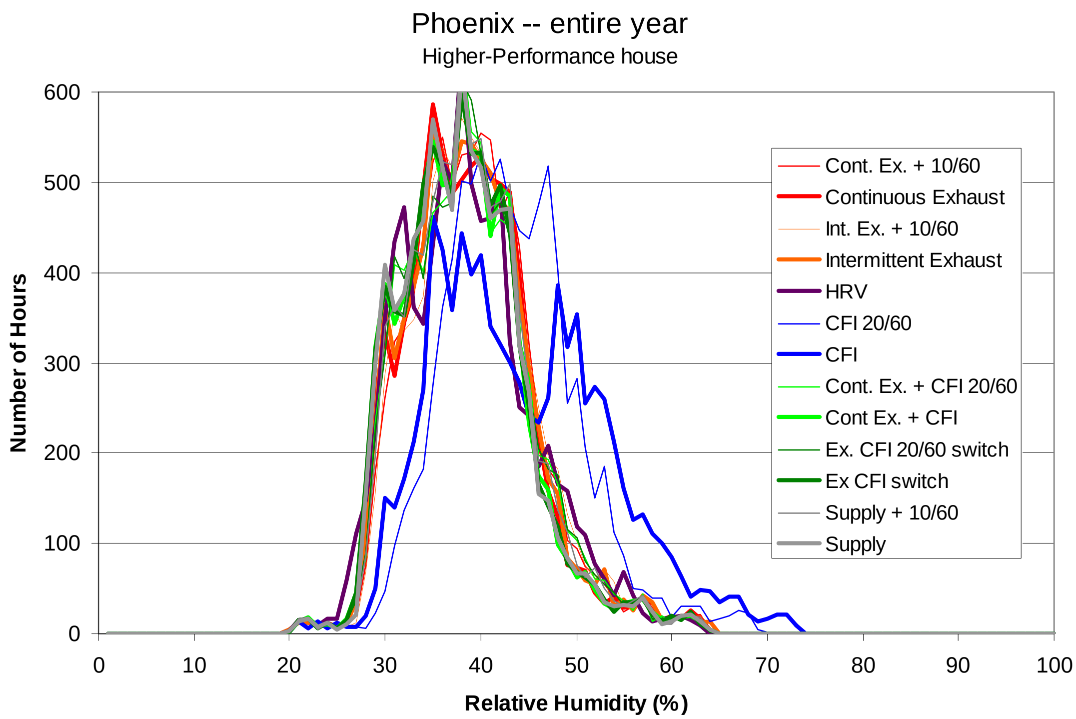


Figure B.13 Frequency distribution of hourly average indoor relative humidity (1% bins) for the entire year in Phoenix

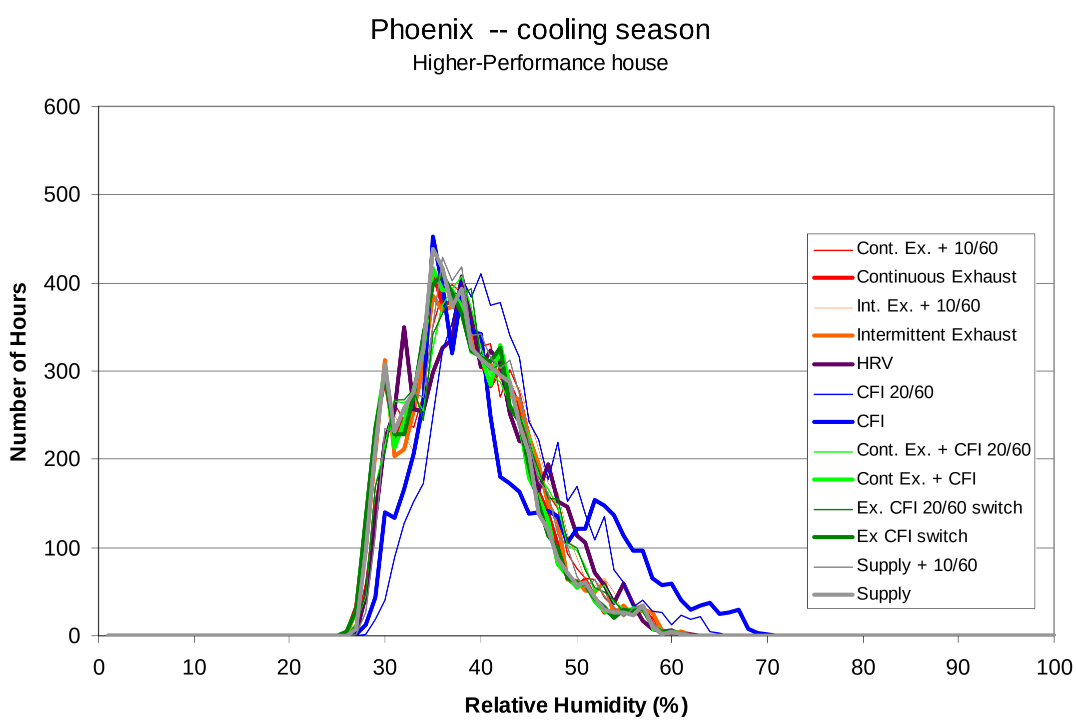


Figure B.14 Frequency distribution of hourly average indoor relative humidity (1% bins) for the cooling season in Phoenix

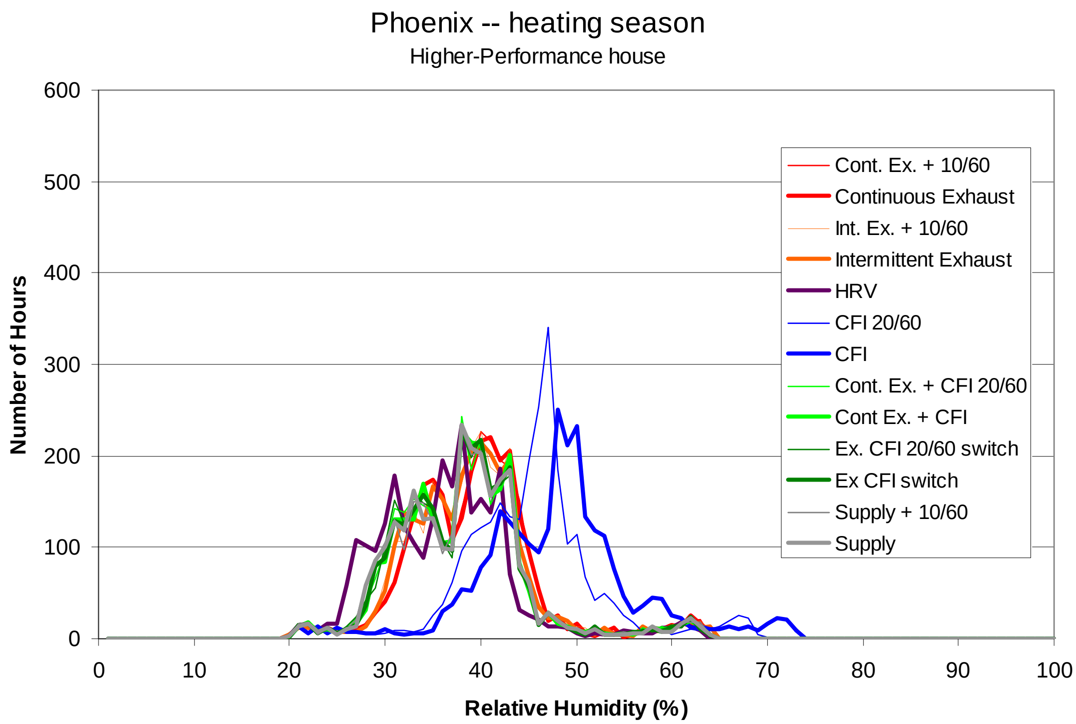


Figure B.15 Frequency distribution of hourly average indoor relative humidity (1% bins) for the heating season in Phoenix

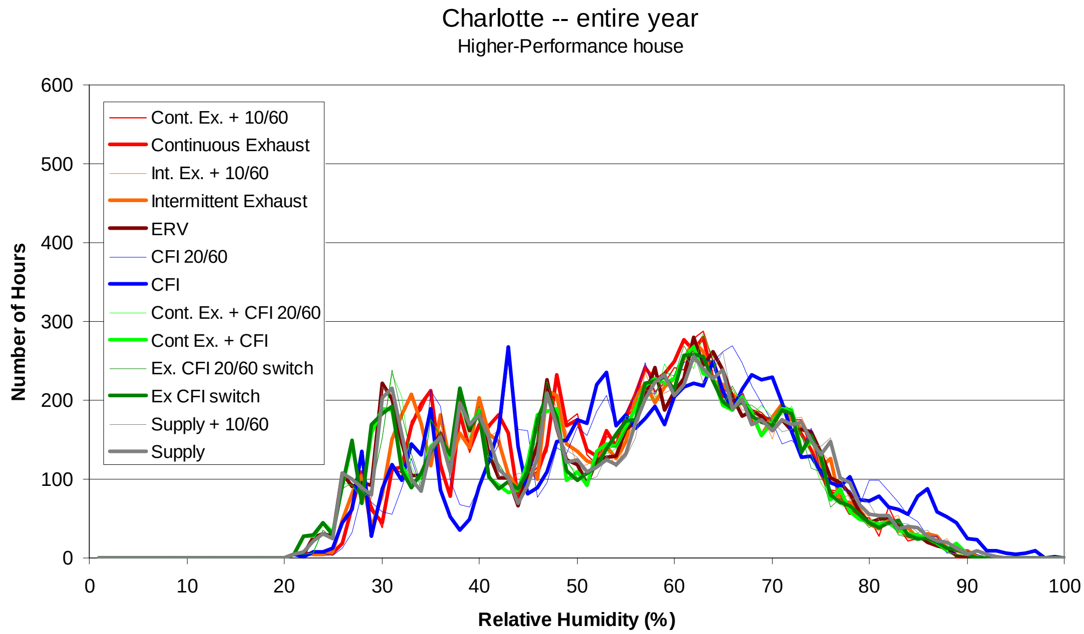


Figure B.16 Frequency distribution of hourly average indoor relative humidity (1% bins) for the entire year in Charlotte

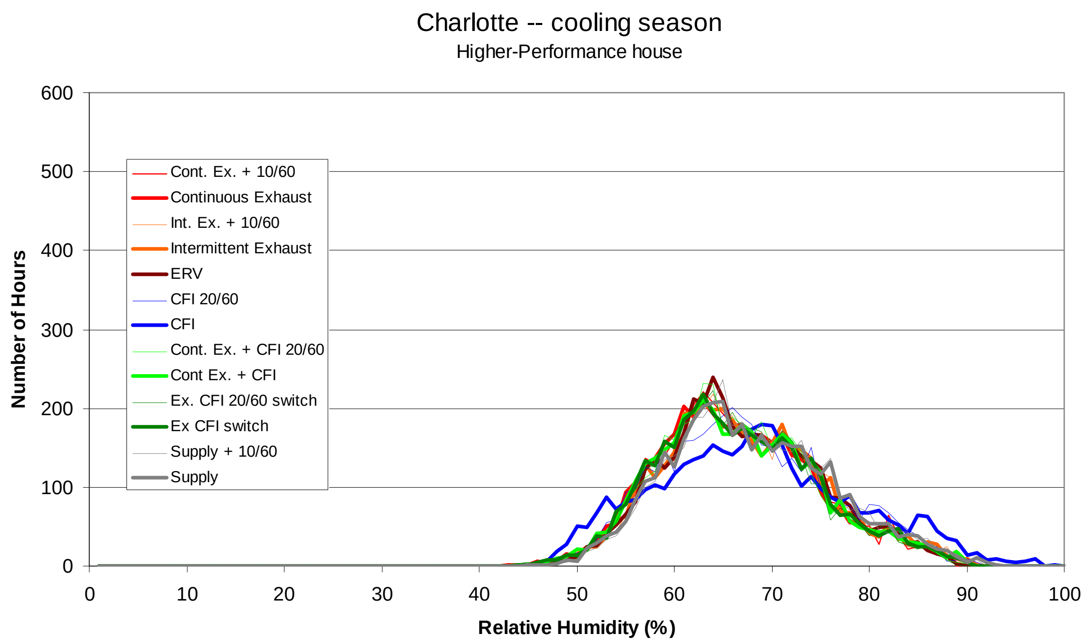


Figure B.17 Frequency distribution of hourly average indoor relative humidity (1% bins) for the cooling season in Charlotte

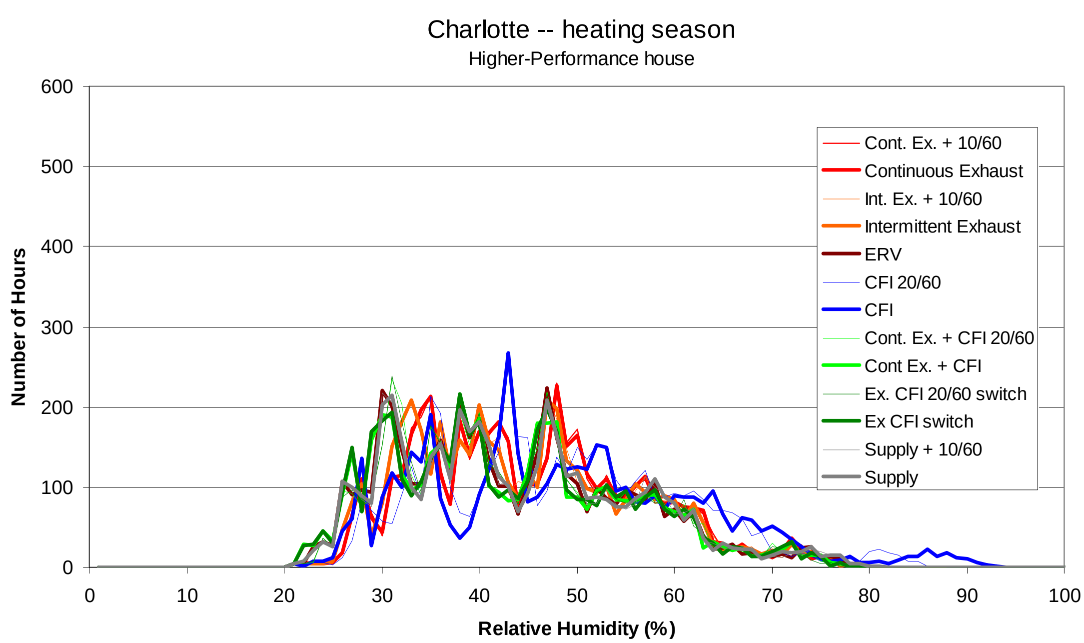


Figure B.18 Frequency distribution of hourly average indoor relative humidity (1% bins) for the heating season in Charlotte

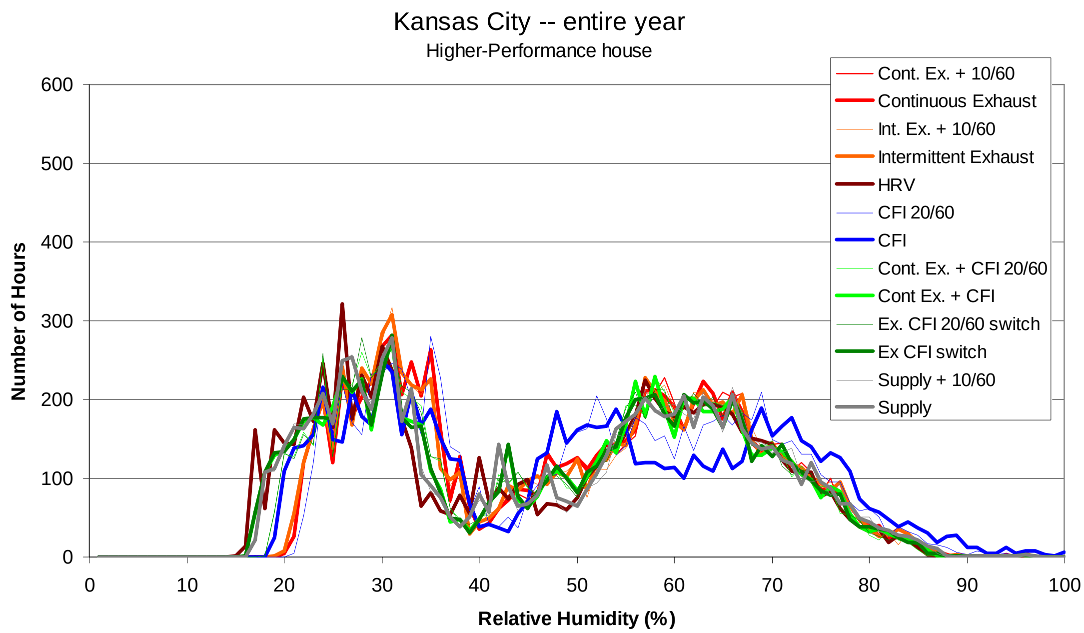


Figure B.19 Frequency distribution of hourly average indoor relative humidity (1% bins) for the entire year in Kansas City

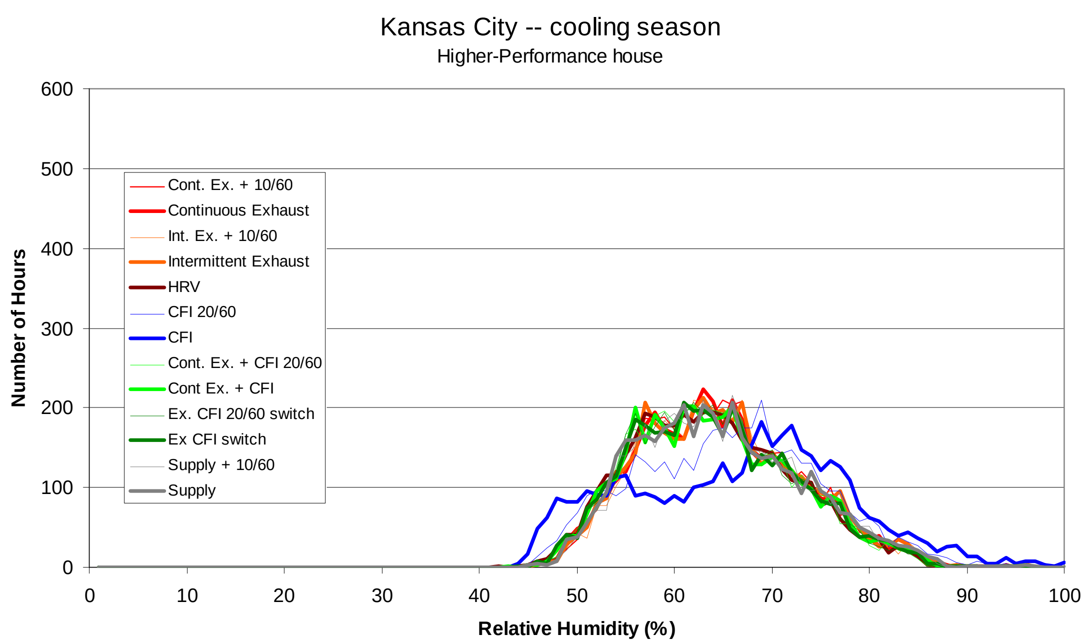


Figure B.20 Frequency distribution of hourly average indoor relative humidity (1% bins) for the cooling season in Kansas City

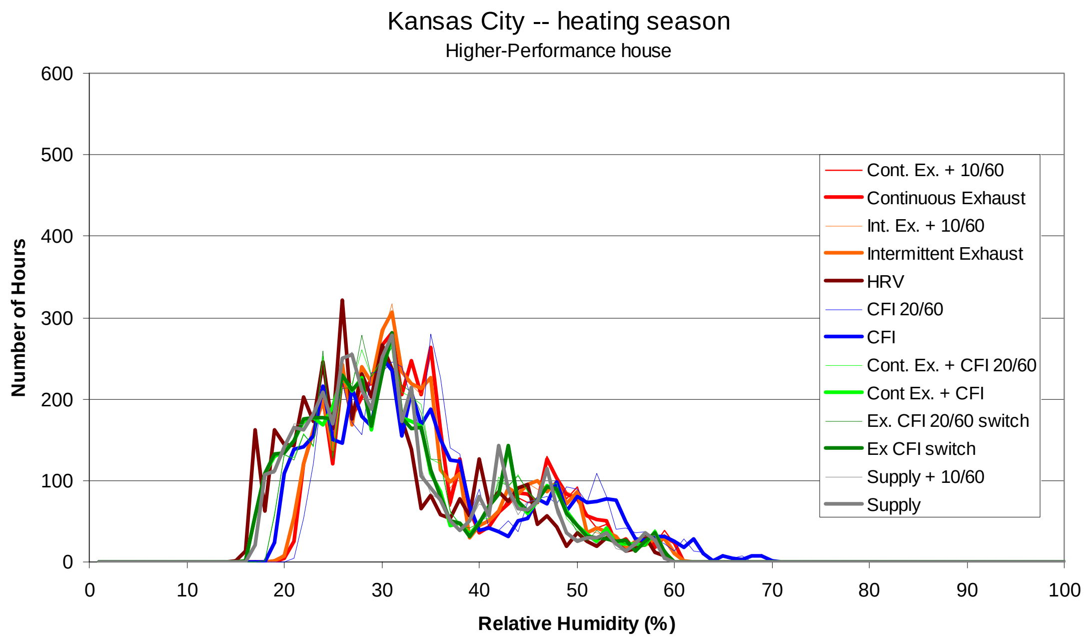


Figure B.21 Frequency distribution of hourly average indoor relative humidity (1% bins) for the heating season in Kansas City

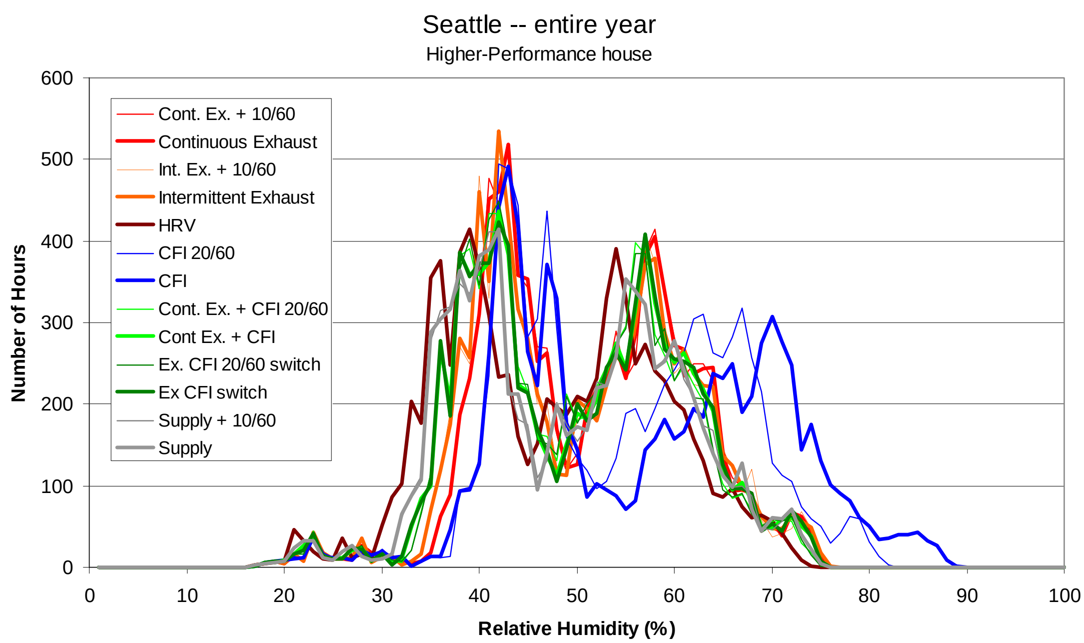


Figure B.22 Frequency distribution of hourly average indoor relative humidity (1% bins) for the entire year in Seattle

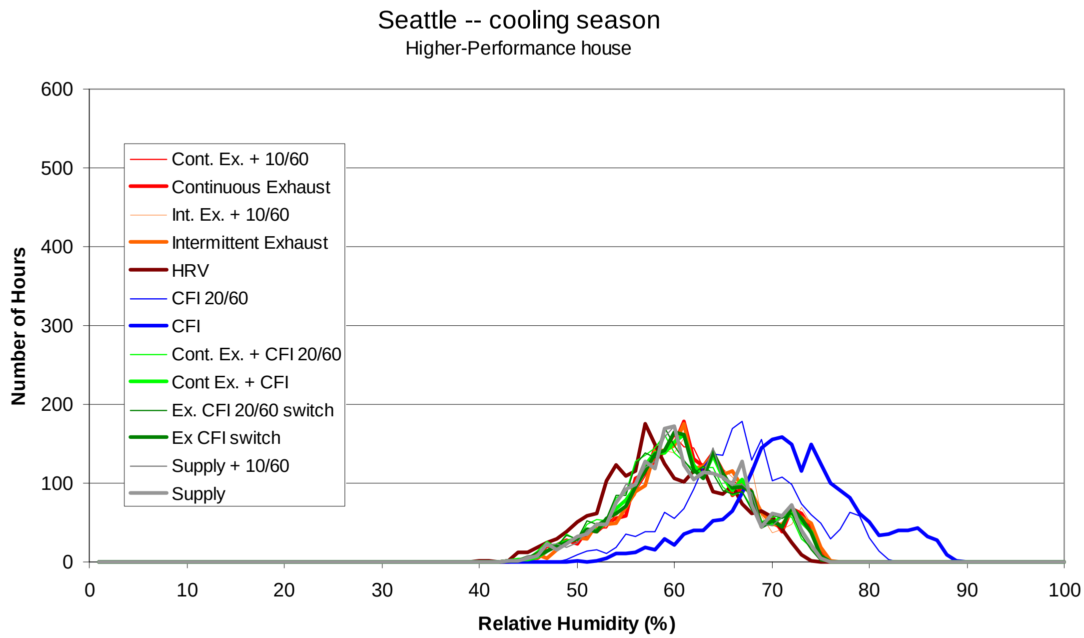


Figure B.23 Frequency distribution of hourly average indoor relative humidity (1% bins) for the cooling season in Seattle

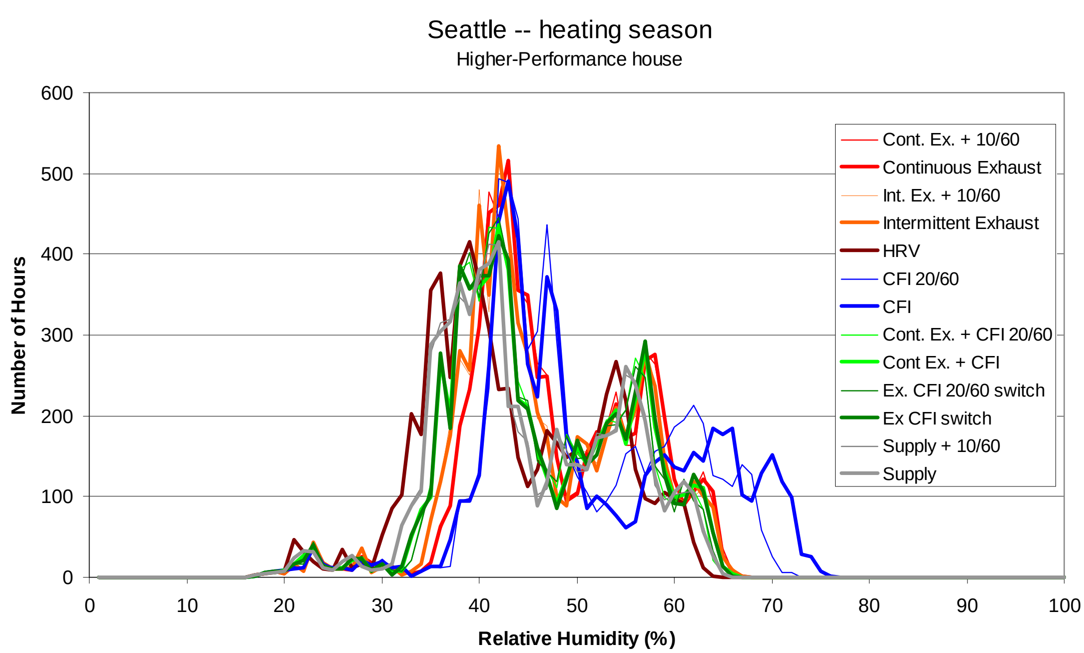


Figure B.24 Frequency distribution of hourly average indoor relative humidity (1% bins) for the heating season in Seattle

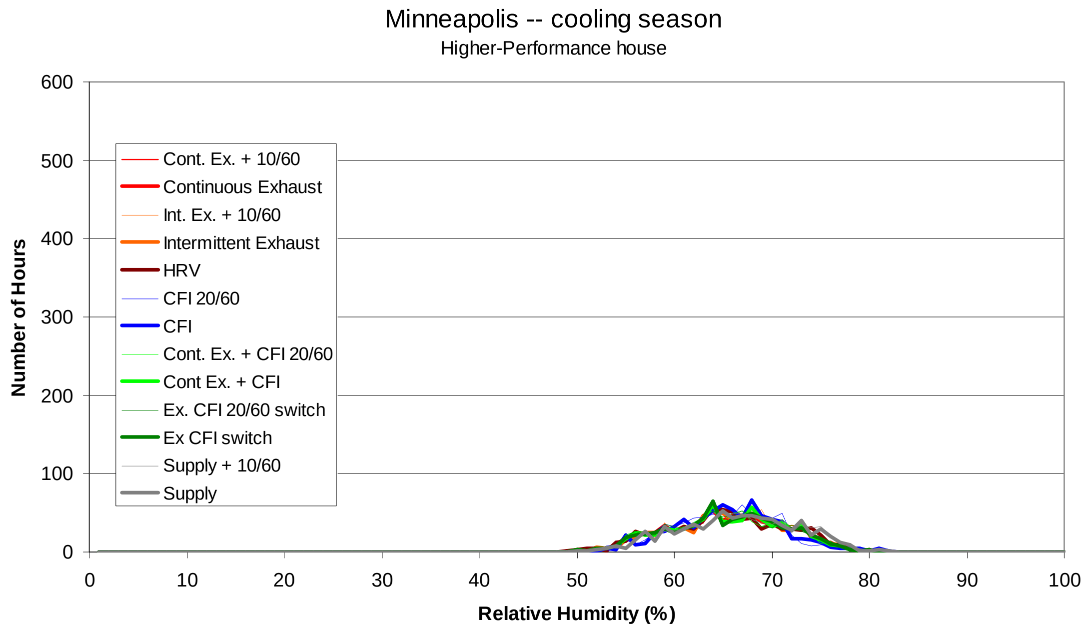


Figure B.25 Frequency distribution of hourly average indoor relative humidity (1% bins) for the cooling season in Minneapolis

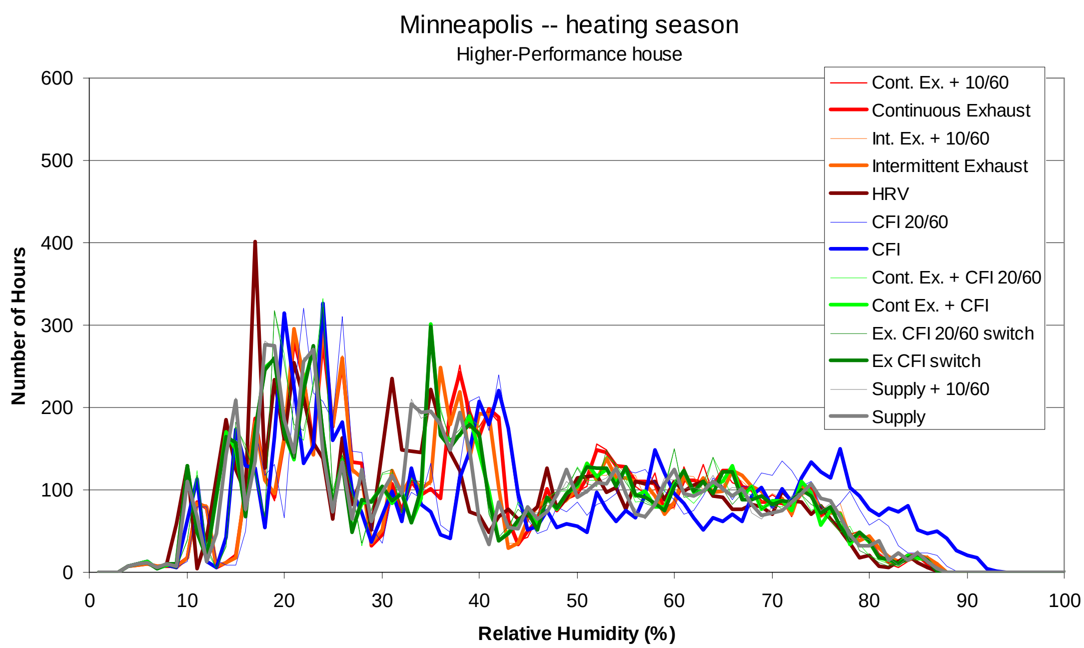


Figure B.26 Frequency distribution of hourly average indoor relative humidity (1% bins) for the heating season in Minneapolis

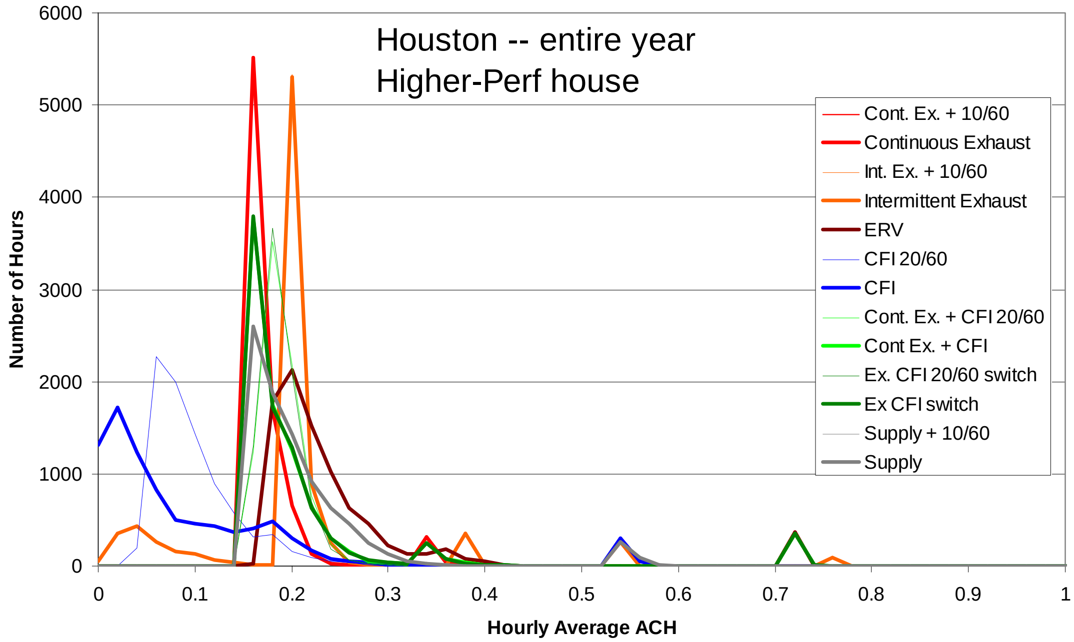


Figure B.27 Frequency distribution of hourly average air change rate (0.02 ach bins) for the entire year in Houston

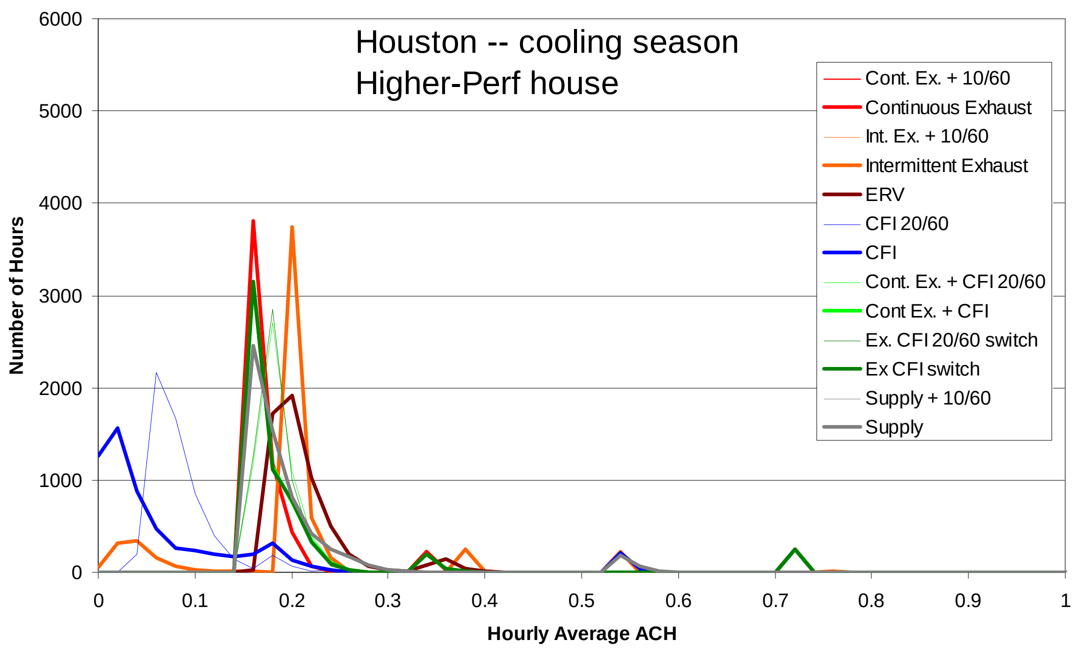


Figure B.28 Frequency distribution of hourly average air change rate (0.02 ach bins) for the cooling season in Houston

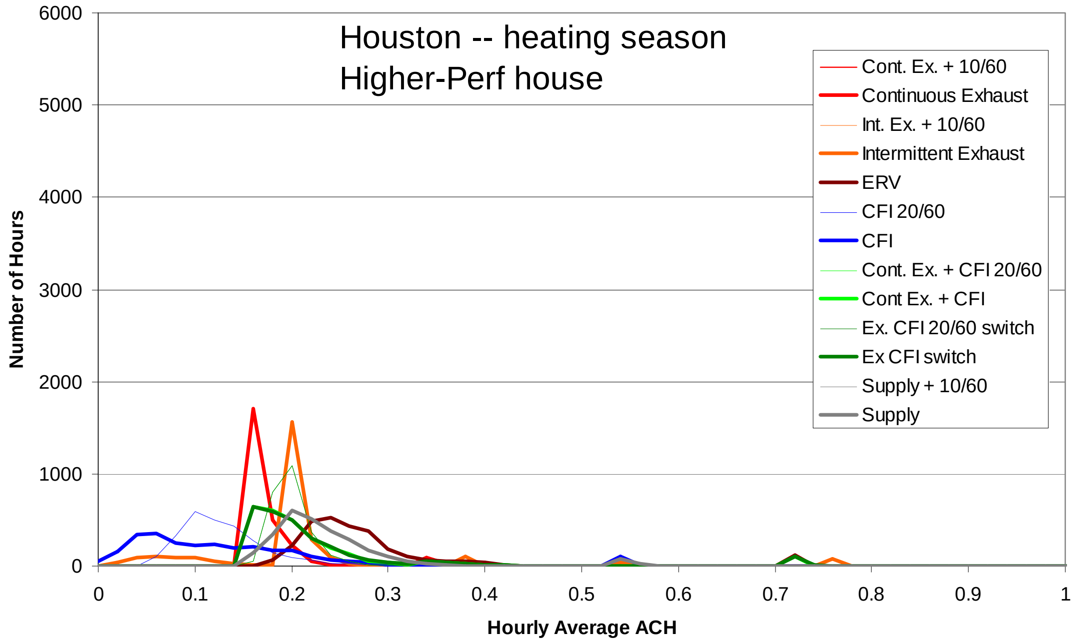


Figure B.29 Frequency distribution of hourly average air change rate (0.02 ach bins) for the heating season in Houston

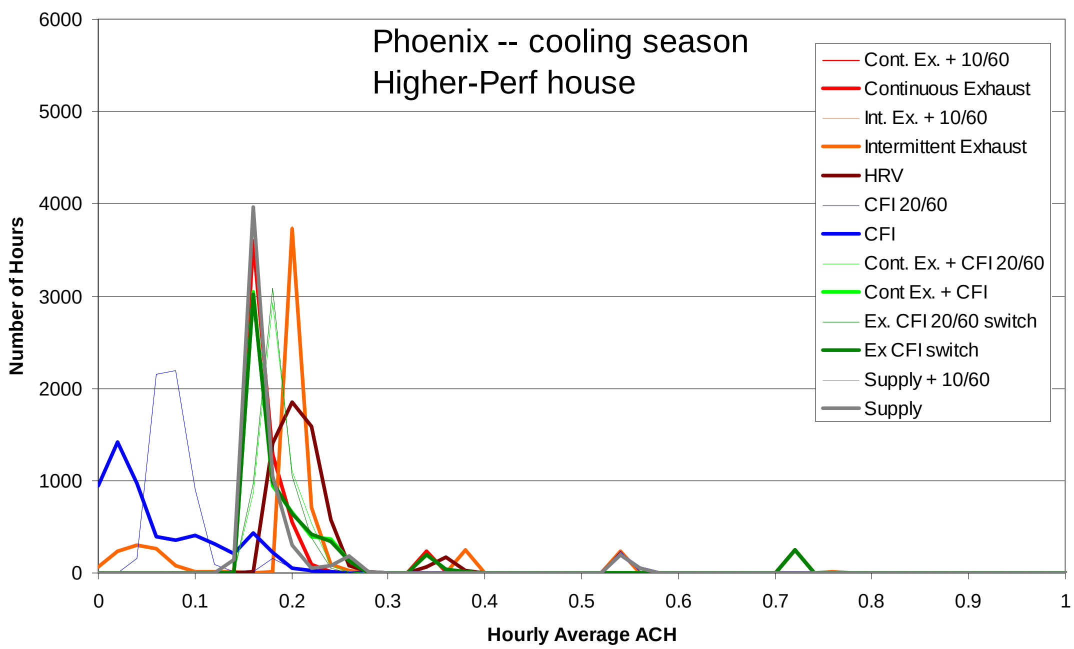


Figure B.30 Frequency distribution of hourly average air change rate (0.02 ach bins) for the cooling season in Phoenix

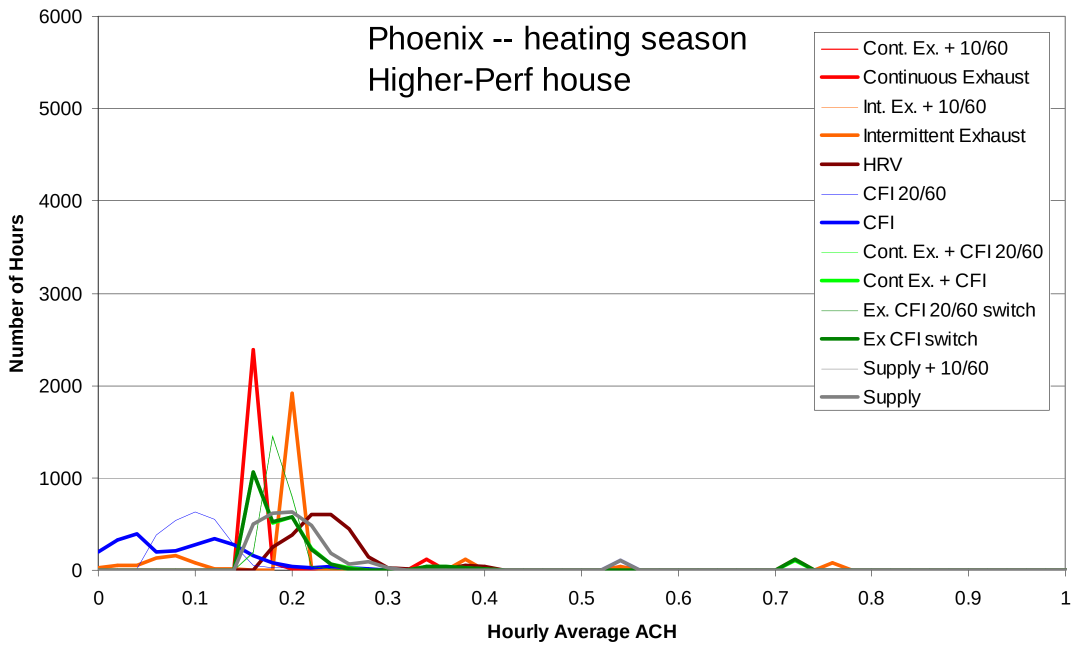


Figure B.31 Frequency distribution of hourly average air change rate (0.02 ach bins) for the heating season in Phoenix

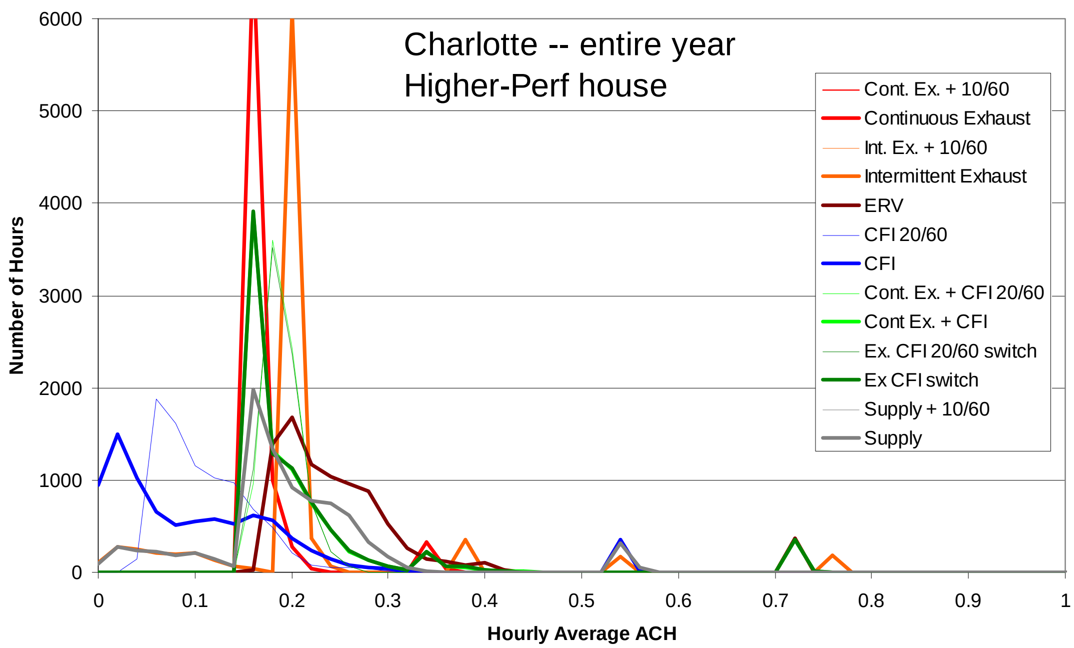


Figure B.32 Frequency distribution of hourly average air change rate (0.02 ach bins) for the entire year in Charlotte

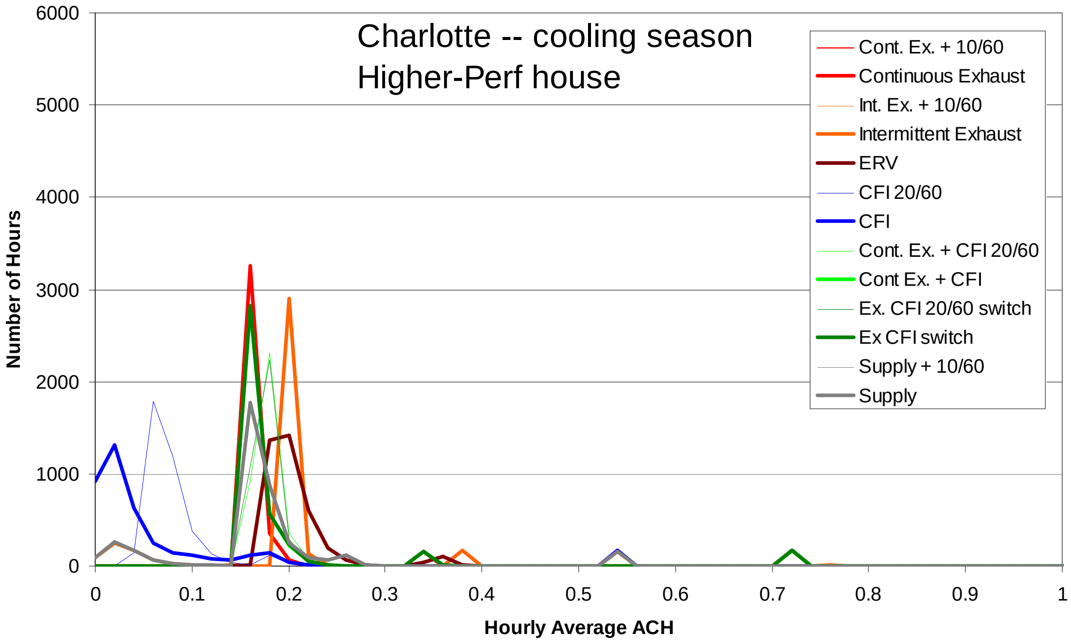


Figure B.33 Frequency distribution of hourly average air change rate (0.02 ach bins) for the cooling season in Charlotte

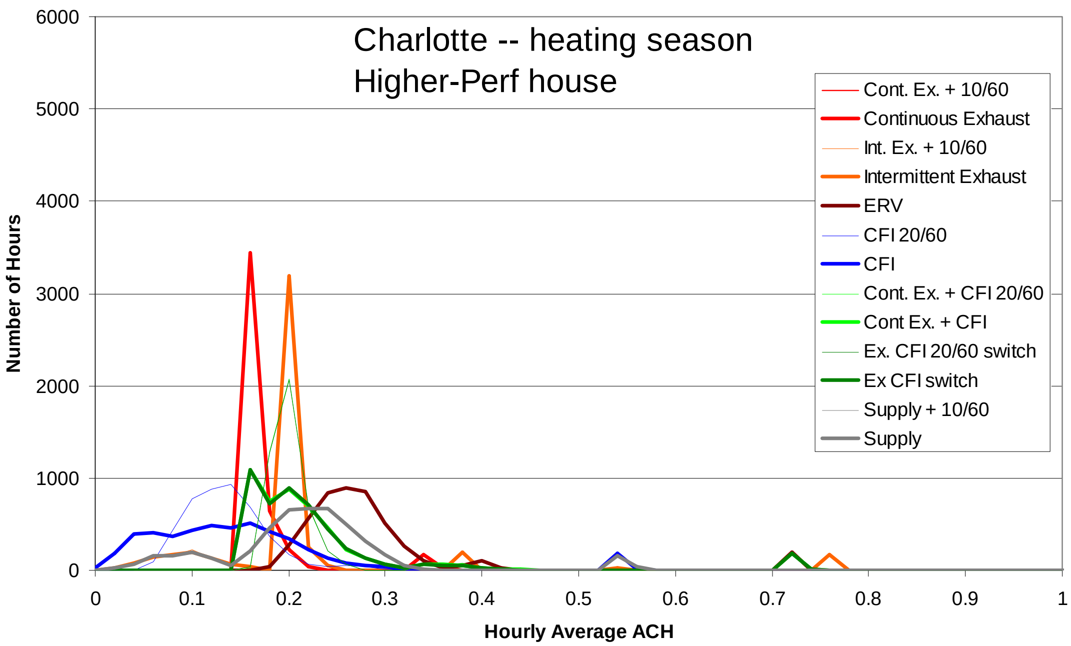


Figure B.34 Frequency distribution of hourly average air change rate (0.02 ach bins) for the heating season in Charlotte

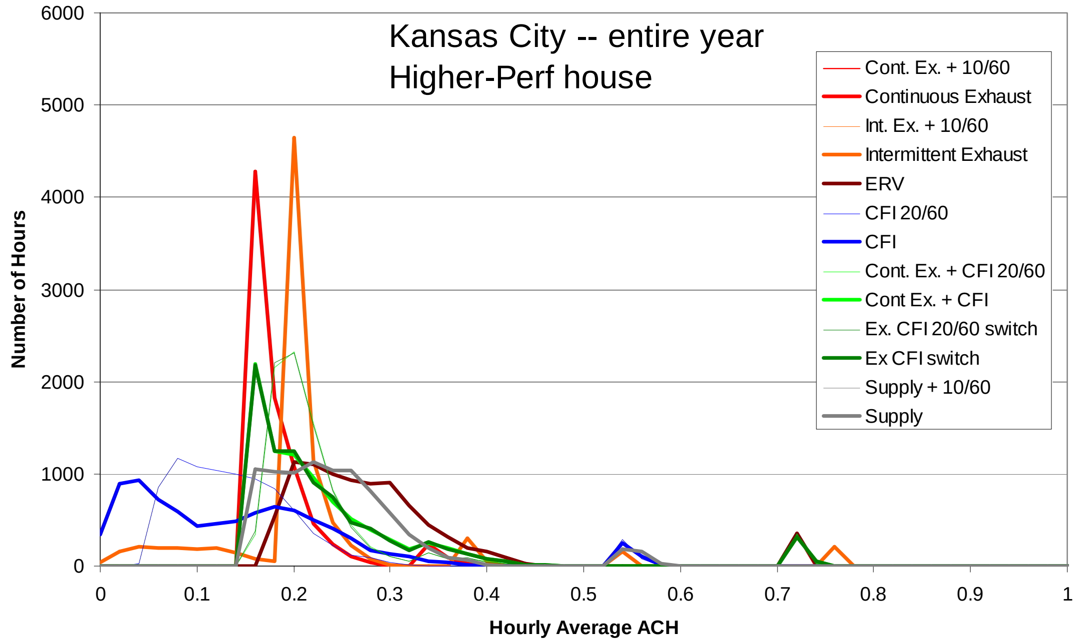


Figure B.35 Frequency distribution of hourly average air change rate (0.02 ach bins) for the entire year in Kansas City

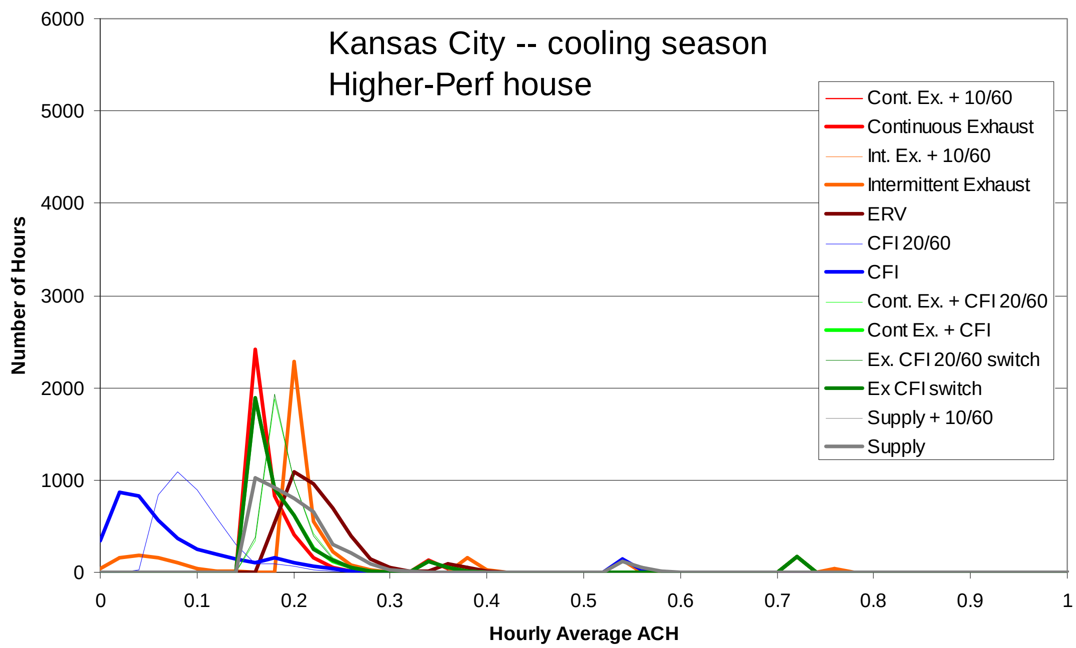


Figure B.36 Frequency distribution of hourly average air change rate (0.02 ach bins) for the cooling season in Kansas City

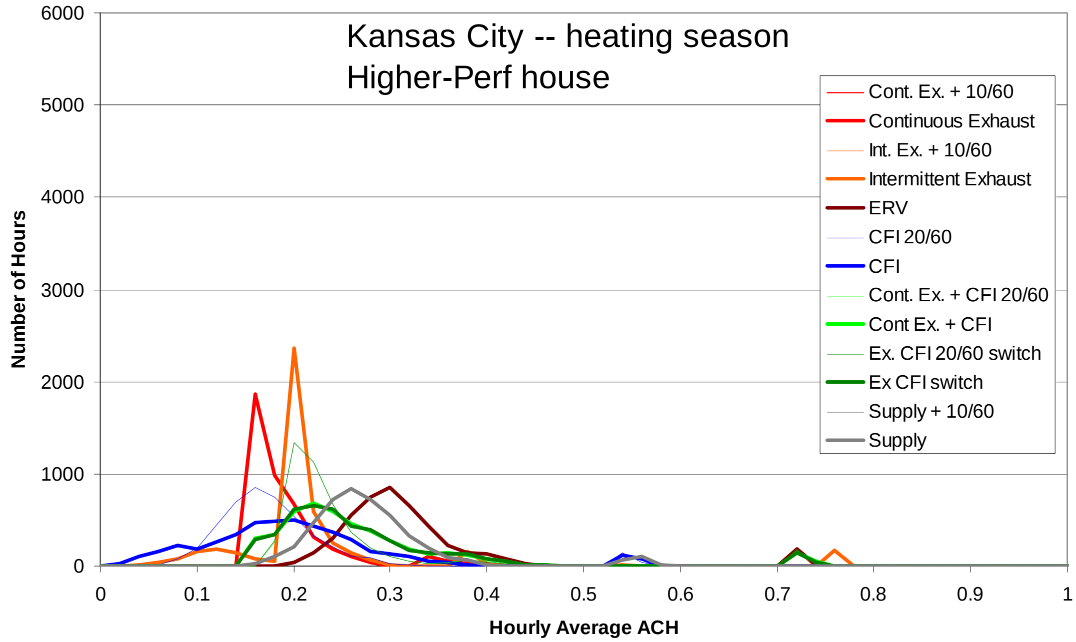


Figure B.37 Frequency distribution of hourly average air change rate (0.02 ach bins) for the heating season in Kansas City

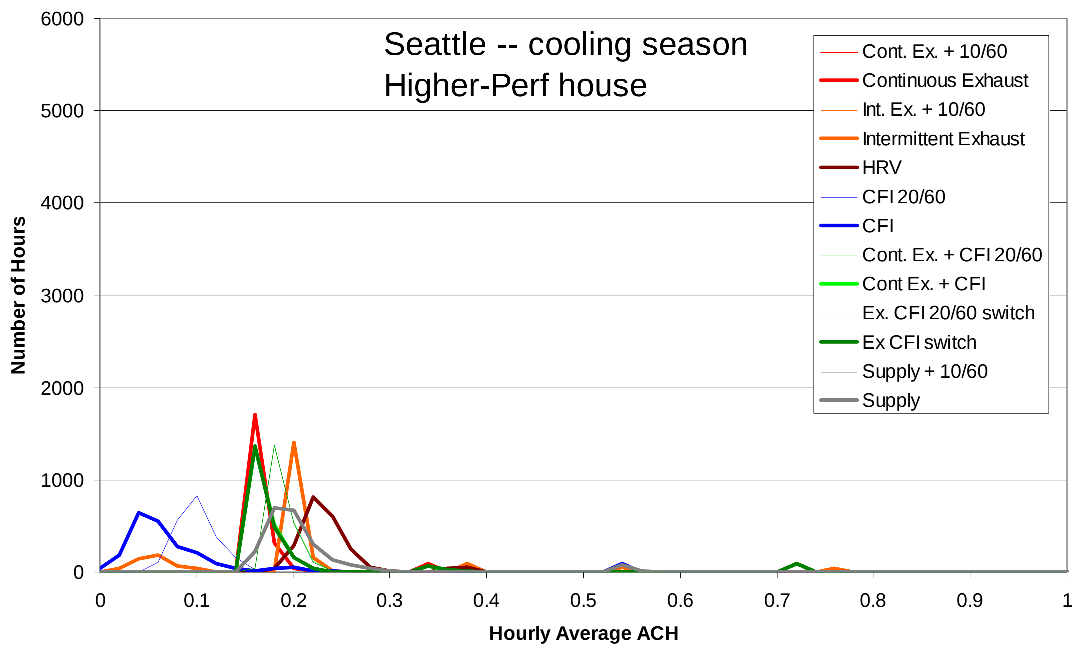


Figure B.38 Frequency distribution of hourly average air change rate (0.02 ach bins) for the cooling season in Seattle

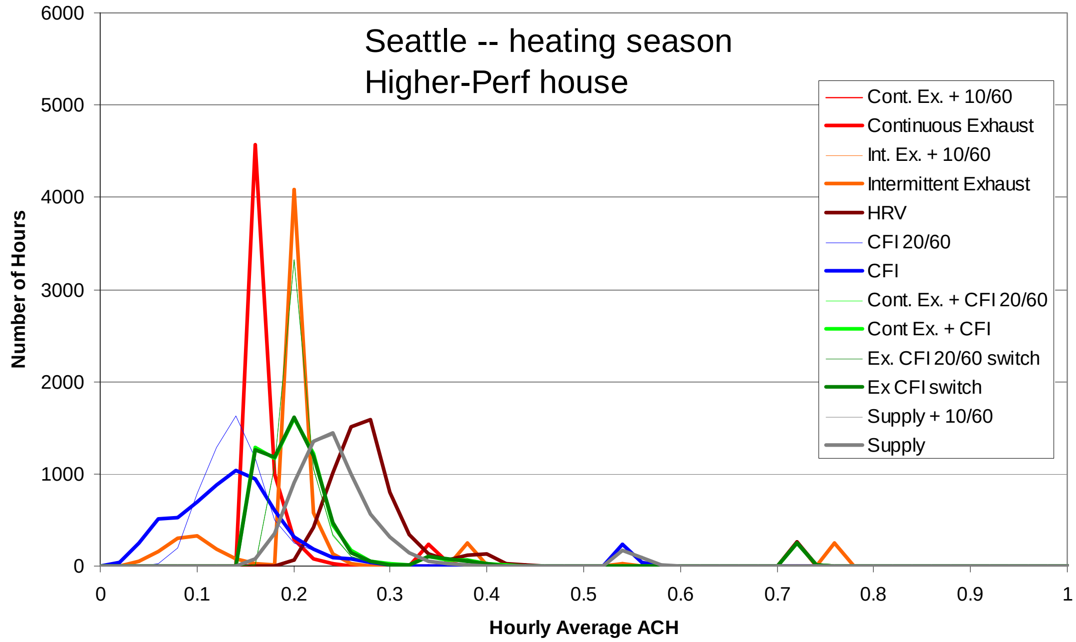


Figure B.39 Frequency distribution of hourly average air change rate (0.02 ach bins) for the heating season in Seattle

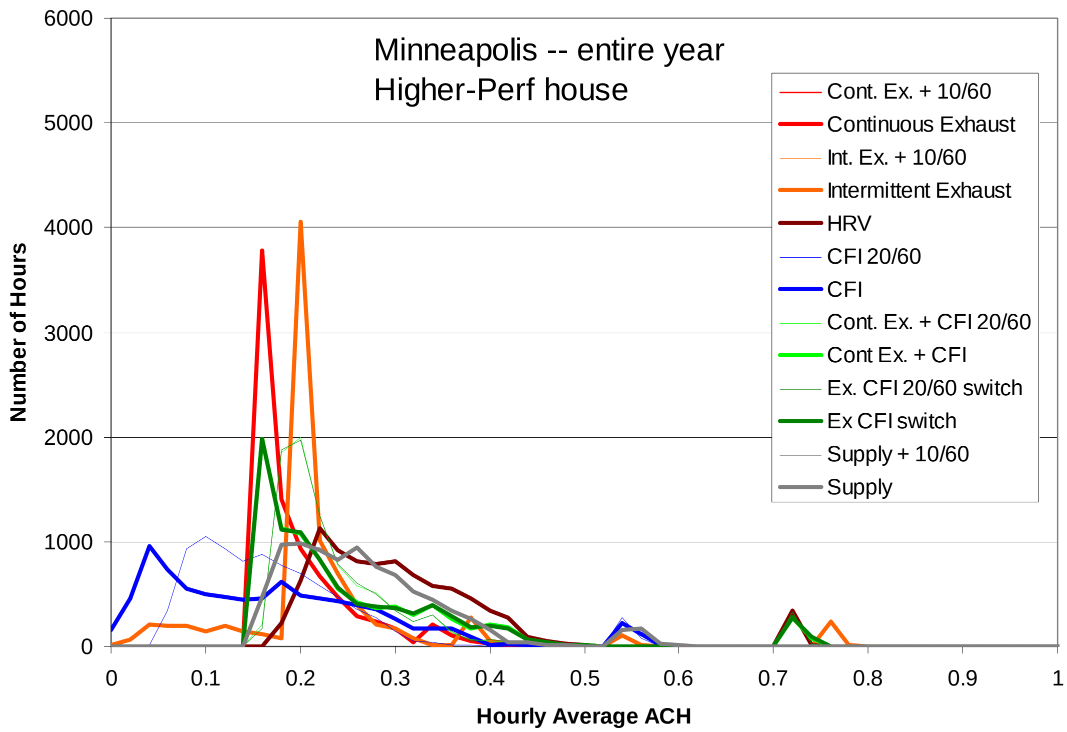


Figure B.40 Frequency distribution of hourly average air change rate (0.02 ach bins) for the entire year in Minneapolis

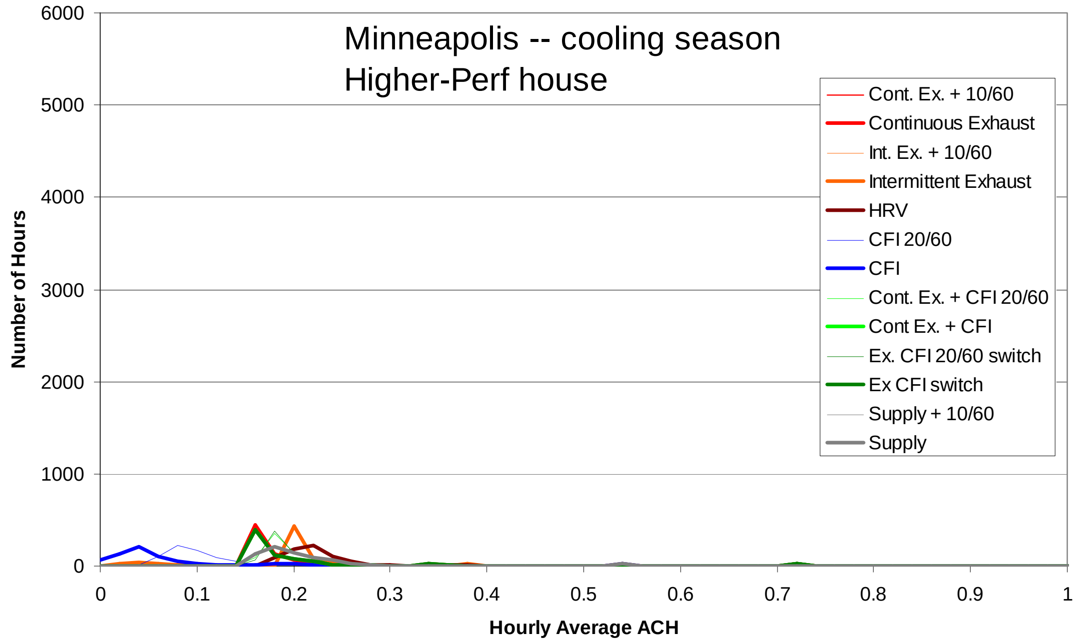


Figure B.41 Frequency distribution of hourly average air change rate (0.02 ach bins) for the cooling season in Minneapolis

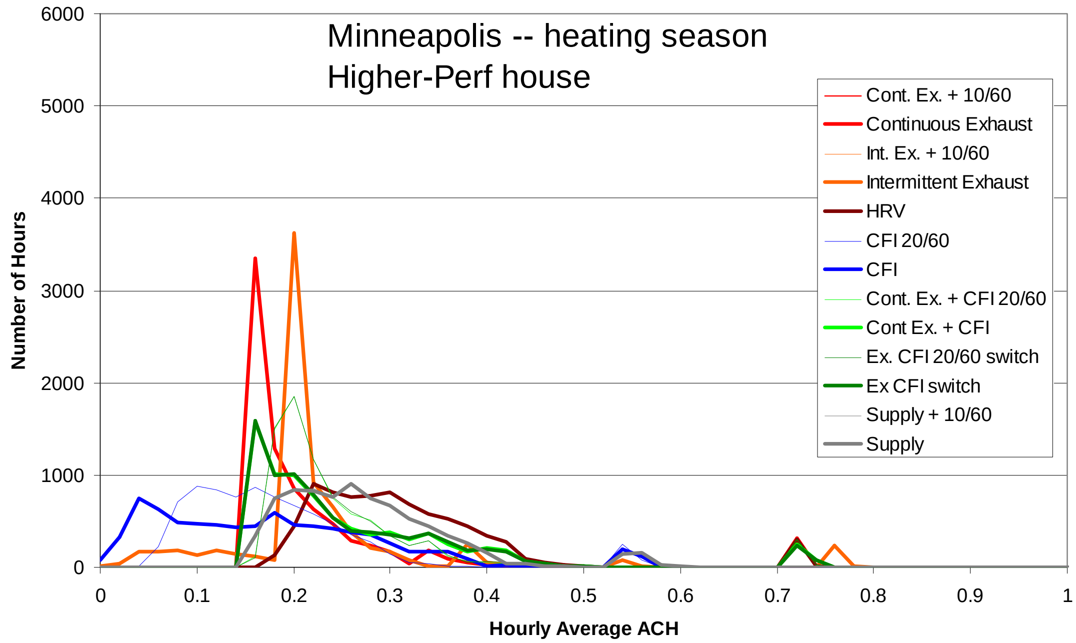


Figure B.42 Frequency distribution of hourly average air change rate (0.02 ach bins) for the heating season in Minneapolis

APPENDIX C

REGCAP model outline

Introduction

The REGCAP model combines a ventilation model, a heat transfer model and a simple moisture model.

The ventilation model developed here is a two zone model, in which the two zones are the attic and the house below it and they interact through the ceiling flow. Both zones use the same type of flow equations and solution method. The total building and attic leakage is separated into components and a flow equation is developed for each leakage site. The envelope flow components are illustrated in Figure C1.

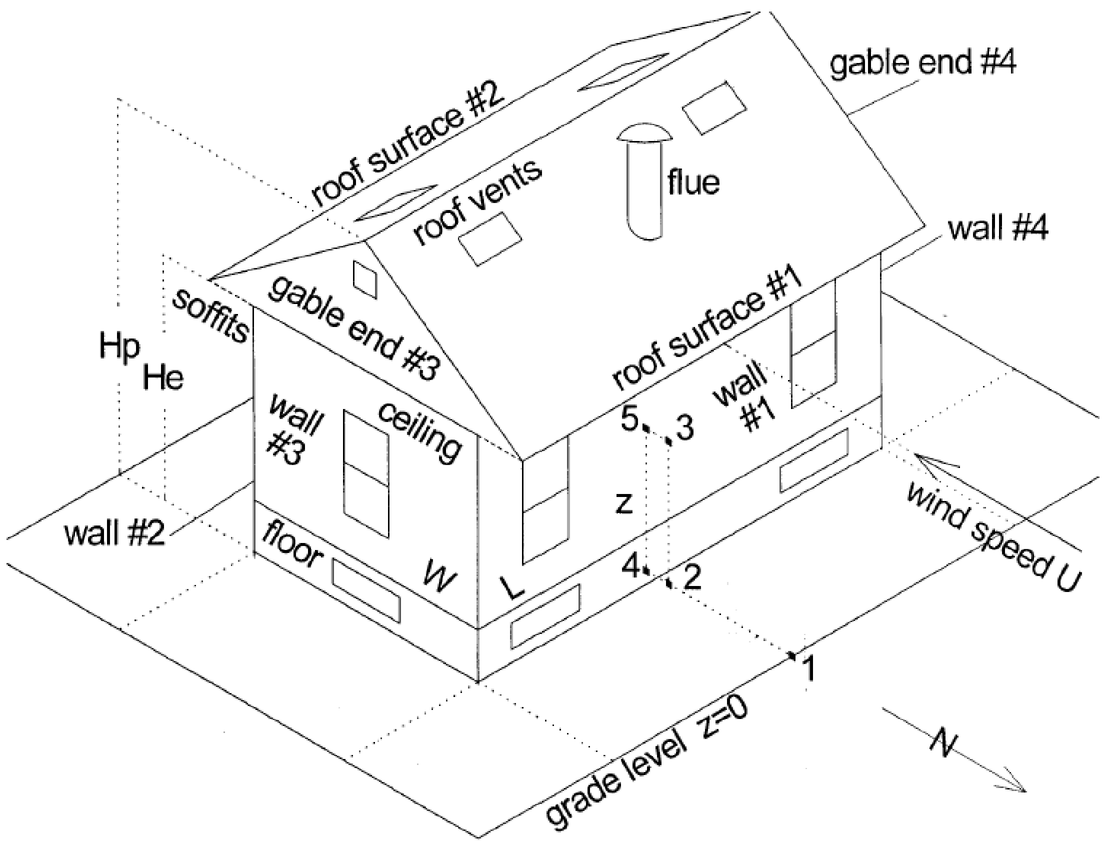


Figure C1. Illustration of house and attic airflow components

The flow at each leakage site is determined by a power-law pressure - flow relationship. This relationship has a flow coefficient, C, that determines the magnitude of the flow and an exponent for pressure difference, n, that determines how the flow through the leak varies with pressure difference. For each zone the total leakage is divided into distributed leakage that consists of the small cracks inherent in the building construction and intentional openings (e.g. furnace flues and open windows). Following the work of Sherman and Grimsrud (1980) the distributed envelope leakage is further divided into specific locations based on the height of the leak (i.e. floor, ceiling and walls). The building is assumed to have a rectangular planform with a user specified length, width and height. The attic has the same floor plan as the house and a pitched roof with soffits and gable ends.

In addition to the envelope leakage, the airflows in and out of attic ducts are included in the mass balances. The ducts are modelled differently depending on if the air handler is on or off. When the air handler is off, the duct leaks are assumed to experience the same pressure difference as the ceiling. Air then flows between the house and the attic via these leaks. When the air handler is on, supply leaks enter the attic and return leak flows are form the attic to the return duct and there are register flows between the ducts and the house.

The ventilation rate of the house and the attic is found by determining the internal pressures for the house and attic that balances the mass flows in and out. Because the relationship between mass flow and pressure is non-linear, the solution is found by iteration.

The attic heat transfer model determines the temperature of the attic air and the other components (e.g., pitched roof surfaces and ducts). A lumped heat capacity method is used to divide the attic into several nodes, and an energy balance is performed at each node to determine the temperatures. The attic air temperature is used to find the attic air density used in the ventilation calculations. The attic ventilation rate changes the energy balance for the attic air and the surface heat transfer coefficients. Fortunately this coupling of the attic ventilation model and the heat transfer model is weak because attic ventilation rates are not a strong function of attic air temperature.

A simple building load model is used to determine indoor air temperature. It uses the total UA for the building together with solar loads (including window orientation – i.e., the area of windows in facing north, south, east and west). A critical part of the house model is the coupling of the house air to the thermal mass of the structure and furnishings. The model uses a combination of thermal mass and surface area together with natural convection heat transfer coefficients.

An equipment model is used to determine heating and cooling system capacities, efficiencies and energy consumption. For gas or electric furnace heating the capacity is fixed for all conditions. For air conditioning, the indoor and outdoor air conditions, together with air handler flow and refrigerant charge are used to determine the cooling system performance.

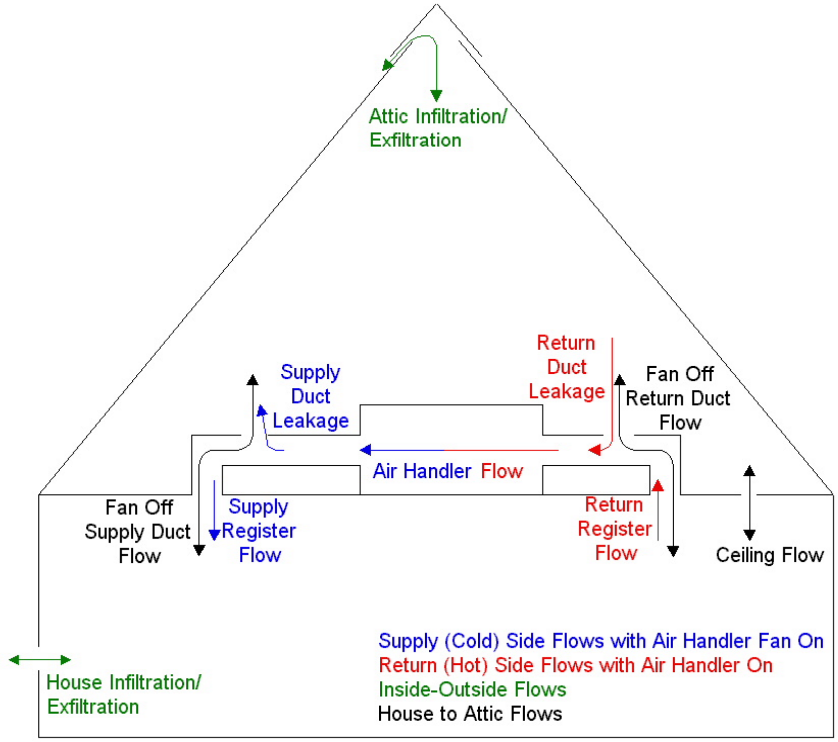


Figure C2: Schematic of duct related airflows (Arrows indicate direction). House and attic air infiltration/exfiltration is the sum of local and distributed leakage.

Ventilation model

The flow through each leakage path is found by determining the internal pressure in the house and attic that balances the mass flow rates. The house and attic interact through the pressure difference and flowrate through the ceiling and duct leaks, and the combined solution is found iteratively. The calculated ventilation rates are used as inputs to the heat transfer model and the building load model. The ventilation model and the heat transfer model are coupled because the ventilation rate effect the amount of outside and house air convected through the attic (as well as convective heat transfer coefficients) and the attic air temperature changes the attic air density. This change in density changes the mass flow rates and the stack effect driving pressures for attic ventilation. The combined ventilation and heat transfer model solution is found iteratively, with the ventilation rate being passed to the heat transfer model that then calculates an attic air temperature. This new attic air temperature is then used in the ventilation model to recalculate ventilation rates. The initial temperature estimate for the attic air used in the first iteration for the ventilation model is the outside air temperature. Most of the time the attic air is within a few degrees of the outside air temperature and the combined ventilation and heat transfer model requires only a few iterations (five or less).

Some significant limitations and assumptions for the ventilation model are listed below:

- There is assumed to be no valving action in the building and attic leakage so that flow coefficients are independent of flow direction.
- The building has a rectangular planform. The planform must not have the longest side greater than about three times the shorter side because the wind pressure coefficients used in the model will be incorrect.

- The attic has two pitched roof surfaces and gable ends. This assumption affects the leakage distribution and the pressure coefficients applied to the attic leakage sites.
- The interior of both the house and the attic are well-mixed zones.
- There are no indoor or outdoor vertical temperature gradients, so that the indoor and outdoor air densities are independent of location.
- Air behaves as an incompressible ideal gas. This allows density and viscosity to be functions of temperature only.
- Wall and pitched roof leakage is evenly distributed so as to allow simple integration of height dependent mass flow equations.
- All wind pressure coefficients are averaged over a surface. This means that extremes of wind pressure occurring at corner flow separations are not included.

General flow equation

The general flow equation for each leak is given by:

$$M = \rho C \Delta P^n \tag{C1}$$

where M = Mass flow rate [kg/s]
ρ = Density of airflow [Kg/m^3]
C = Flow coefficient [m^3/(sPa^n)]
ΔP= Pressure difference across the leak [Pa]
n = Pressure exponent

The flow direction is determined by ΔP where a positive ΔP produces inflow and a negative ΔP produces outflow. A density and viscosity correction factor is applied to C to account for changes due to the temperature of the airflow.
Neglecting atmospheric pressure changes:

$$C = C_{ref}\left(\frac{T}{T_{ref}}\right)^{3n-2} \tag{C2}$$

where T_{ref} is the absolute reference temperature (K) at which C_{ref} was measured, and T is the temperature of the airflow. For many buildings the distributed background leakage has n~2/3, which means that this correction is unity. For simplicity this temperature correction was therefore not applied to distributed leakage. For localised leakage sites including furnace flues, passive vents and attic vents n is typically 0.5 and this correction can become significant and therefore it is included in the ventilation calculations.

Each leak is then defined by its flow coefficient, pressure exponent, height above grade, wind shelter, and wind pressure coefficient. For distributed leakage on walls and pitched roof surfaces, an integral closed form equation is used. Similarly, for open doors and windows and integrated Bernoulli relation is used that includes interfacial mixing effects. For duct leakage with the air handler on, fixed user specified flow rate is used. For ventilation fans, a simple fan law is used so that the flow through the fan changes with the pressure difference across the fan. In the future these ventilation fan flows can simply be fixed values as the relationship between pressure difference and airflow is not generally known.

Wind Pressures

To find the outside surface wind pressure for each leak a wind pressure coefficient, Cp, is used that includes a windspeed multiplier, S_U to account for shelter. The wind speed, U, is the eaves height wind speed. The following equation is then used to calculate the pressure difference due to wind effect:

$$\Delta P_U = \rho_{out} Cp \frac{(S_U U)^2}{2} \tag{C3}$$

where ΔP_U is the difference between the pressure on the surface of the building due to the wind and the atmospheric reference pressure P_∞ (at grade level, z=0). ρ_{out} is chosen as the reference density for pressures, because pressure coefficients are measured in terms of the external flow and the outdoor air density is used to calculate pressure coefficients from measured surface pressures. P_∞ is the pressure in the atmosphere far away from of the building where the building does not influence the flow field. S_U is a windspeed multiplier that accounts for windspeed reductions due to upwind obstacles. $S_U = 1$ implies no shelter and $S_U = 0$ implies complete shelter and there is no wind effect. Because each leak has a different Cp and S_U it is convenient to define a reference wind pressure P_U as

$$P_U = \rho_{out} \frac{U^2}{2} \tag{C4}$$

and then Equation C3 can be written in terms of P_U:

$$\Delta P_U = Cp\, S_U^2\, P_U \tag{C5}$$

This definition is used later in the equations for the flow through each leak.

Indoor-Outdoor Temperature Difference Pressures

The hydrostatic pressure gradient inside and outside the building depends on the air temperature. Different temperatures inside and outside result in a differential pressure across the building envelope, ΔP_T. ΔP_T is defined as the outside pressure minus the inside pressure. This convention is applied so that positive pressures result in flow into the building (the same as for wind effect). Integrating the resulting pressure difference means that the stack effect pressure difference at height z above grade is given by

$$\Delta P_T(z) = -zg\rho_{out}\left(\frac{(T_{in} - T_{out})}{T_{in}}\right) \tag{C6}$$

where z is the height above a reference (grade level) [m]
g is gravitational acceleration (9.81 [m/s^2]).

Each leak is at a different height, z, above grade, and so for convenience in writing the mass flow equations P_T is defined as follows:

$$P_T = g\rho_{out}\left(\frac{(T_{in} - T_{out})}{T_{in}}\right) \tag{C7}$$

P_T is the pressure gradient and is multiplied by the height of each leak above grade to find the stack effect pressure difference at that location. Substituting Equation C6 in C5 gives:

$$\Delta P_T(z) = -z P_T \tag{C8}$$

Total Pressure Difference

The total pressure difference is due to a combination of these wind and indoor-outdoor temperature difference effects, together with ventilation fan and HVAC system airflows, and the indoor to outdoor pressure shift (ΔP_I) that acts to balance the inflows and outflows. ΔP_I is the only unknown in this equation, and is the same for every leak in each zone. The total pressure difference is given by:

$$\Delta P = Cp\, S_U^2\, P_U - z\, P_T + \Delta\, P_I \tag{C9}$$

Equation C9 is applied to every leak for the building and the attic with the appropriate values of Cp, S_U and z.

The linear change in pressure, ΔP, with height, z, due to the stack effect term in Equation C9 means that when inflows and outflows are balanced there is a location where there is no pressure difference. This is called the neutral level, H_{NL}. For $T_{in} > T_{out}$ flow is in below H_{NL} and out above H_{NL}, and the flow directions are reversed for $T_{out} > T_{in}$. In general the neutral level is different for each wall due to the inclusion of wind pressures which can drive H_{NL} above the ceiling or below the floor. In those cases there is one way flow through the wall. The neutral level is found for the i^{th} vertical by setting $\Delta P = 0$ in Equation 9 and solving for $z = H_{NL,i}$:

$$H_{NL,i} = \left(\frac{\Delta\, P_I + S_{U,i}^2\, Cp_i\, P_U}{P_T} \right) \tag{C10}$$

Wind Pressure Coefficients For the house

Wind pressure coefficients are taken from wind tunnel tests and depend on the wind direction. For closely spaced houses in a row the pressure coefficients also change due to the change in flow around the building. Walker and Wilson (1994) discuss these vary in greater detail. Table C1 contains the wall averaged wind pressure coefficients used for the house by the ventilation model for wind perpendicular to the upwind wall. For the closely spaced row, the wind is blowing along the row of houses.

Table C1. Wall averaged wind pressure coefficients for a rectangular building with the wind normal to upwind wall from Akins, Peterka and Cermak (1979) and Wiren (1985).

Shelter Configuration	Cp, Wind Pressure Coefficient		
	Upwind Wall	Side Walls	Downwind Wall
Isolated House	+0.60	-0.65	-0.3
In-Line Closely-Spaced Row	+0.60	-0.2	-0.3

When the wind is not normal to the upwind wall a harmonic trigonometric function is used to interpolate between these normal values to fit the variation shown by

Akins, Peterka, and Cermak and Wiren. For each wall of the building the harmonic function for Cp from Walker and Wilson (1994) is used:

$$Cp(\theta) = \frac{1}{2}[(Cp(1) + Cp(2))(\cos^2\theta)^{\frac{1}{4}} + (Cp(1) - Cp(2))(\cos\theta)^{\frac{3}{4}}$$
$$+(Cp(3) + Cp(4))(\sin^2\theta)^2 + (Cp(3) - Cp(4))\sin\theta] \qquad \text{(C11)}$$

where Cp(1) is the Cp when the wind is at 0^o (+0.60)
Cp(2) is the Cp when the wind is at 180^o (-0.3)
Cp(3) is the Cp when the wind is at 90^o (-0.65 or -0.2)
Cp(4) is the Cp when the wind is at 270^o (-0.65 or -0.2)

and θ is the wind angle measured clockwise from the normal to the wall.
This function is shown in Figure C3 together with data from Akins et. al. for a cube. The error bars on the data points in Figure C3 represent the uncertainty in reading the measured values from the figures of Akins, Peterka and Cermak.

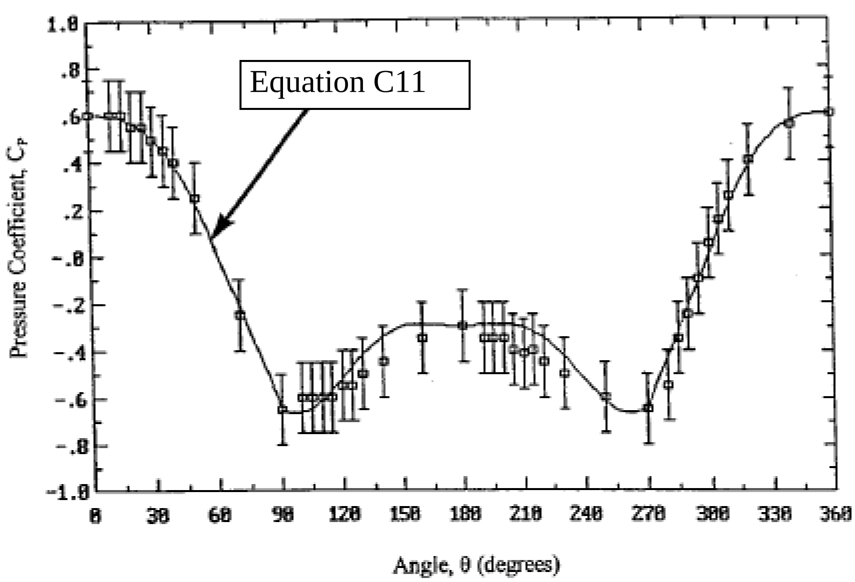


Figure C3. Angular variation in wind pressure coefficient for a rectangular building

Wind Pressure Coefficients for the Attic

The attic simulation model has been developed for a gable end attic with two pitched roof surfaces. The Cp's for gable ends or soffits are assumed to be the same as those on the walls below them and are calculated using the same procedure as for house walls. The pitched roof surfaces have Cp's that are also a function of roof slope. Table C2 gives values of Cp measured by Wiren (1985) for upwind and downwind pitched roof surfaces with wind normal to the upwind surface for different roof pitches. For wind flow parallel to the roof ridge Cp's change in the same way as for houses with Cp = -0.6 for an isolated building and Cp = -0.2 for row houses for both roof pitched surfaces. The Cp is independent of roof pitch for flow parallel to the roof ridge.

Table C2. Pitched roof wind pressure coefficients for wind normal to the upwind surface (Wiren (1985))

Roof Pitch	Cp, Wind Pressure Coefficient	
	Upwind Surface	Downwind Surface
$<10^o$	-0.8	-0.4
10^o to 30^o	-0.4	-0.4
$>30^o$	+0.3	-0.5

To account for the variation on roof Cp with wind angle a similar empirical relationship to that for houses is used (from Walker, Forest and Wilson (1995)):

$$Cp(\theta) = \frac{1}{2}[(Cp(1) + Cp(2))\cos^2\theta + (Cp(1) - Cp(2))F$$
$$+(Cp(3) + Cp(4))\sin^2\theta + (Cp(3) - Cp(4))\sin\theta] \qquad (C12)$$

where Cp(1) is the Cp when the wind is at 0^o
Cp(2) is the Cp when the wind is at 180^o
Cp(3) is the Cp when the wind is at 90^o
Cp(4) is the Cp when the wind is at 270^o
θ is the wind angle measured clockwise from the normal to the roof surface.
F is a switching function to account for changes in roof pitch.

$$F = \frac{1-(|\cos\theta|)^5}{2}\left(\frac{28-\psi}{28}\right)^{0.01} + \frac{1+(|\cos\theta|)^5}{2} \qquad (C13)$$

where ψ is the roof pitch in degrees measured from horizontal. Equation C13 acts like a switch with $F \sim 1$ up to $\psi = 28^o$ and $F \sim \cos\theta$ when $\psi > 28^o$. The switch point of 28^o is chosen so that this relationship produces the same results as the wind tunnel data in Table C2. Equation C13 is not used to change the pressure coefficients shown in Table C2, but it changes the functional form of Equation C12 so that the interpolation fits the measured pressure coefficients.

Equation C12 is compared with pitched roof Cp's from Liddament (1986) in Figures C4 through C6 for roof pitches $>30^o$, 10^o to 30^o, and $<10^o$ respectively.

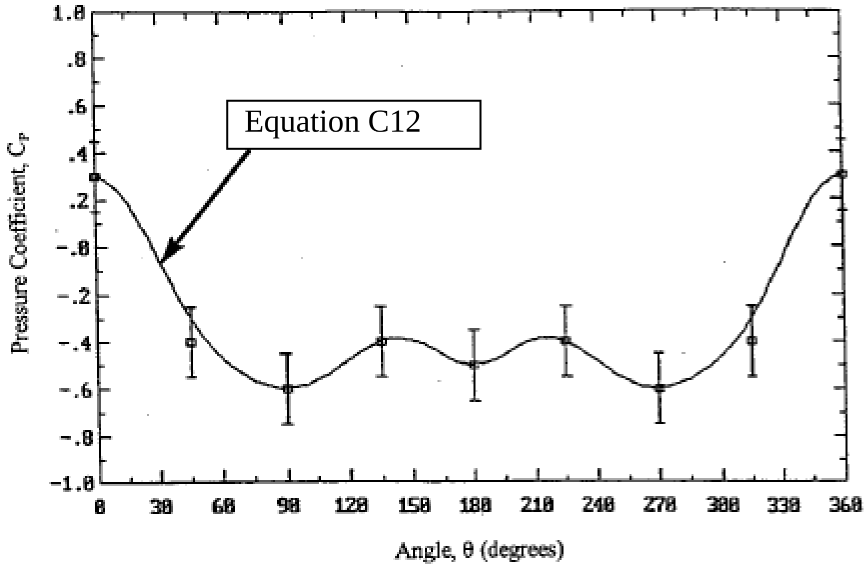


Figure C4. Roof pressure coefficients for a steep sloped roof (pitch > 30°)

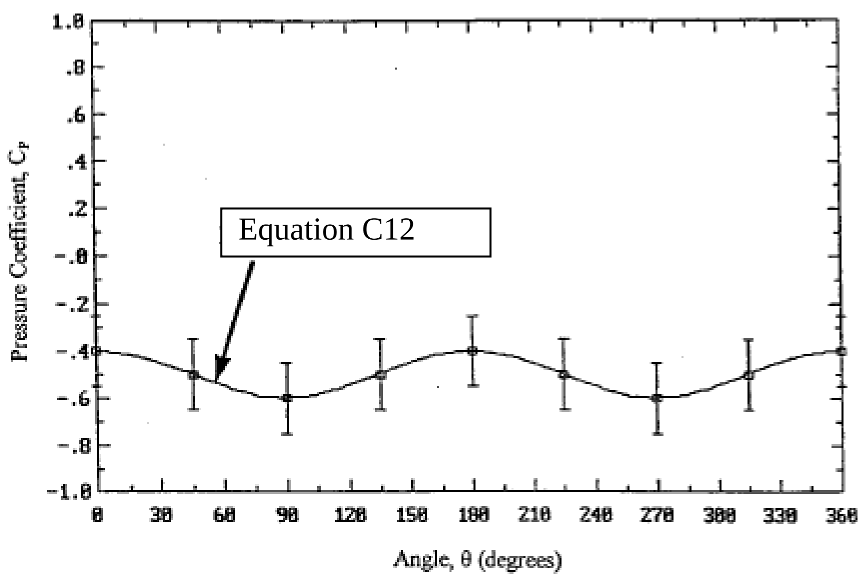


Figure C5. Roof pressure coefficients for a moderate sloped roof (10° < pitch < 30°)

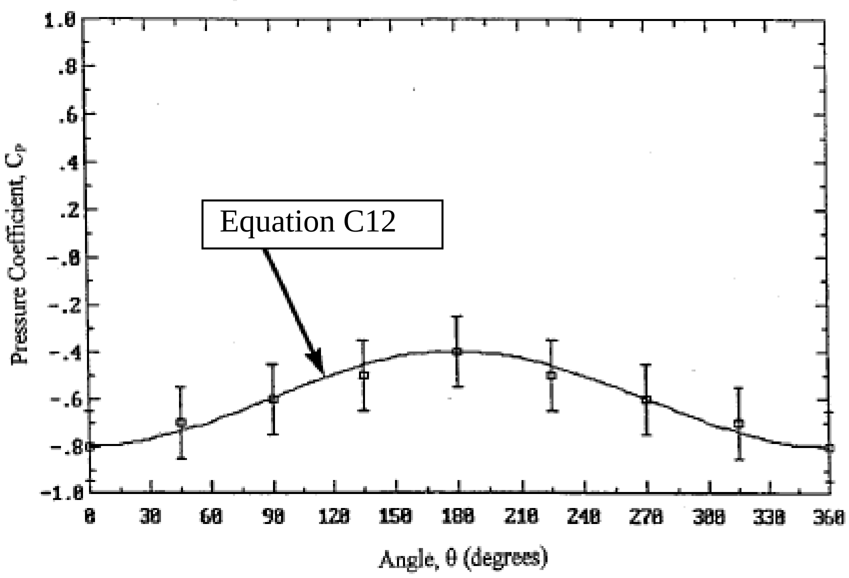


Figure C6. Roof pressure coefficients for a low sloped roof (pitch < 10°)

Wind Shelter

Shelter effects are separated from the effects of changing Cp's with wind direction and flow field changes. The windspeed multiplier, S_U, acts to reduce the effective windspeed generating surface pressures on the building such that:

$$U_s = S_U U \qquad \text{(C14)}$$

where U is the free stream windspeed with no sheltering effects.

S_U has the limits where $S_U = 1$ implies no shelter and $S_U = 0$ implies total shelter and there are no wind pressures on the building.

U_S is the effective windspeed used for calculating surface pressures.

The coefficients used to find U_S and S_U are based on measured surface pressures and not on measured wake velocities.

REGCAP has the following three options for wind shelter.

1. Fixed shelter for all wind directions.

2. Interpolation Function.

The interpolation function determines shelter for all wind angles given shelter for four cardinal directions so that for each wall:

$$S_U = \frac{1}{2}[(S_U(1) + S_U(2))\cos^2\theta + (S_U(1) - S_U(2))\cos\theta$$
$$(S_U(3) + S_U(4)\sin^2\theta + (SU(3) - S_U(4))\sin\theta] \qquad \text{(C15)}$$

where S_U is the windspeed multiplier

$S_U(1)$ is the S_U when the wind is at 0^o

$S_U(2)$ is the S_U when the wind is at 180^o

$S_U(3)$ is the S_U when the wind is at 90^o

$S_U(4)$ is the S_U when the wind is at 270^o

and θ is the wind angle measured clockwise from the normal to the upwind wall.

3. Input from data file:

A file of shelter values for every degree of wind direction for all four faces of a house was generated using sophisticated wind shelter calculation techniques discussed in Walker, Wilson and Forest (1996). The following figure illustrates the values of shelter coefficient in the pre-calculated data file for one wall.

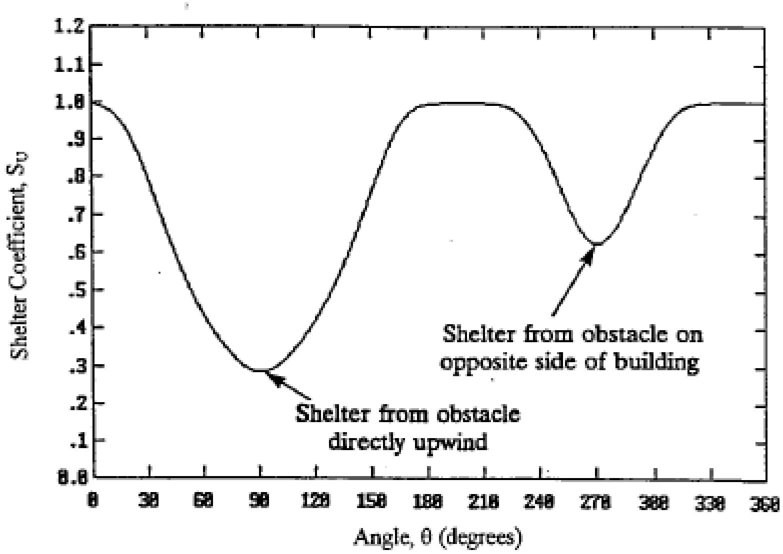


Figure C7. Wind shelter for a typical urban house

Flow Through each Leak for the Attic

The total leakage is divided into distributed leakage and localised leakage. All the distributed leakage sites are assumed to have the same flow exponent. The flow coefficients for the roof and soffit must be estimated as fractions of the total distributed leakage such that

$$C_{d,a} = \sum_{i=1}^{4} C_{s,i} + C_r \tag{C16}$$

where C_r is the total leakage in the two pitched roof surfaces and $C_{s,i}$ is the leakage in the soffit or gable ends above each wall.

Pitched roof Leakage

The two pitched roof surfaces are assumed to have equal leakage. Therefore there is $C_r/2$ leakage in each surface. Cp for the pitched roof surfaces is found using Equation C12 and Table C2. If the surrounding obstacles are taller than the building in question then S_U for the pitched roof surfaces is estimated to be the same as the wall below them, otherwise there is no shelter and S_U=1. For example, a south facing roof pitch would then have the same S_U as calculated for the south facing wall below it. For the attic roof the neutral level, $H_{NL,r}$, is calculated for the two roof pitches using the appropriate Cp and S_U values in Equation C10.

The change in pressure with height, z, on the roof surfaces makes the flow through the roof a function of height which must be integrated to find the total mass flow in and out of each roof surface $M_{r,i}$.

$$M_{r,i} = \int dM_{r,i}(z)dz \tag{C17}$$

where

$$dM_{r,i}(z) = \rho\, dC_{r,i}(\Delta P_{r,i}(z))^{n_r} \tag{C18}$$

where $\Delta P_{r,i}(z)$ is given by Equation C9. Assuming evenly distributed leakage allows easy integration over the roof because the fractional leakage $dC_{r,i}$ is given by:

$$dC_{r,i} = C_{r,i}\frac{dz}{(H_p - H_e)} \tag{C19}$$

where H_p is the roof peak height and H_e the eave height. Substituting Equations C19 and C18 in C17 gives

$$M_{r,i} = \frac{\rho C_{r,i}}{(H_p - H_e}\int \Delta P_{r,i}^{n_r} dz \tag{C20}$$

where the limits of integration depend on the neutral level height, $H_{NL,r}$, that is found for each wall using Equation C18.

When $H_{NL,r}$ is on the roof there is flow both in and out of the roof and upon integrating Equation C20 the masses flowing in and out are kept separate. This is important for the total mass balance and for keeping track of all the flows through the building envelope. There are several different cases of flow through the pitched roof surfaces depending on the location of $H_{NL,r}$, T_a and T_{out}. The pressure differences at the eave height, ΔP_e, and at the roof peak, ΔP_p, are defined as follows and are convenient to use when calculating the mass flow rates.

$$\Delta P_p = \Delta P_{I,a} + S_U^2 Cp\, P_U - H_p P_{T,a} \tag{C21}$$

$$\Delta P_e = \Delta P_{I,a} + S_U^2 Cp\, P_U - H_e P_{T,a} \tag{C22}$$

An example case given by Equations C23 and C24 is for $T_a > T_{out}$ with $H_{NL,r}$ somewhere on the pitched roof surface between the eave height, H_e, and the peak height H_p. There is two way flow through the roof surface in this case with flow in below $H_{NL,r}$ and flow out above $H_{NL,r}$:

$$M_{r,out} = \frac{\rho_a \frac{C_r}{2}\Delta P_p^{(n_r+1)}}{(H_p - H_e)P_{T,a}(n_r+1)} \tag{C23}$$

$$M_{r,in} = \frac{\rho_{out} \frac{C_r}{2}\Delta P_e^{(n_r+1)}}{(H_p - H_e)P_{T,a}(n_r+1)} \tag{C24}$$

Other cases are given in Appendix D.

Soffit and Gable Leakage

The soffit and gable leakage are treated identically. The soffit and gable leakage is split into four parts, one for each side of the building. $C_{s,i}$ is the estimated fraction of the total attic distributed leakage in the soffit or gable on the i^{th} side of the building. H_s is the height of the leakage above grade and usually $H_s = H_e$ for soffits. For the gable leakage H_s is assumed to be H_e plus half of the attic height (H_p - H_e). The wind pressure coefficient (Cp_i) and shelter factor ($S_{U,i}$) are assumed to be the same as for the wall below each soffit or gable. The pressure difference across each soffit or gable above wall i is then given by:

$$\Delta P_{s,i} = \Delta P_{I,a} + Cp_i \; S_{U,i}^2 \; P_U - H_s P_{T,a} \qquad \text{(C25)}$$

Attic Vent Leakage

Attic vents provide extra ventilation leakage area in addition to the background distributed leakage. There can be multiple attic vents at different locations on the attic envelope, each with their own C_V and n_V. C_V and n_V are user specified leakage characteristics of each vent. Usually the vent can be assumed to act like an orifice with $n_V = 0.5$. In that case C_V can be estimated from the vent area multiplied by the discharge coefficient, K_D. The vent area should be corrected for any blockage effects e.g. by insect screens. $S_{U,V}$ and Cp_V for each vent are the same as for the attic surface they are on, either the gable ends (which have the same S_U and Cp as the wall below them) or the roof pitches. H_V is the height above grade of the vent and the pressure difference across each attic vent is given by:

$$\Delta P_{V,a} = \Delta P_{I,a} + S_{U,V}^2 \; Cp_V \; P_U - H_V \; P_{T,a} \qquad \text{(C26)}$$

$\Delta P_{V,a}$ is calculated for each attic vent and the flow through each attic vent is given by Equation 1.

Attic Floor Leakage

The mass flow rate through the attic floor is calculated by the house zone part of the ventilation model. The resulting $\Delta P_{I,a}$ from balancing the mass flows for the attic zone is returned to the house zone to calculate pressure across the ceiling, and then to recalculate the mass flow through the attic floor.

Ventilation Fans in Attics

Fans are included by using a fan performance curve. The operating point on the curve is determined by the pressure across the fan. The stack and wind pressures across each fan are found by specifying which attic surface the fan is located in and its height above grade, H_{fan}. Cp_{fan} and $S_{U,fan}$ are the same as the surface the fan is are located in. There can be multiple fans each with their own rated flowrates, Q_{rated}, and rated pressure differences, ΔP_{rated}. The pressure difference across each attic fan, $\Delta P_{fan,a}$, is given by:

$$\Delta P_{fan,a} = \Delta P_{I,a} + S_{U,fan}^2 Cp_{fan} P_U - H_{fan} P_{T,a} \qquad \text{(C27)}$$

Approximating the fan performance curve by a power law using p_{fan} gives the following equation for mass flow through each fan:

$$M_{fan,a} = \rho Q_{rated} \left(\frac{\Delta P_{rated} + \Delta P_{fan,a}}{\Delta P_{rated}} \right)^{p_{fan}} \qquad \text{(C28)}$$

where ρ is equal to ρ_a for outflow and ρ_{out} for inflow.

Duct Leaks with air handler off (M_{soff} and M_{roff})
Both the supply and return leaks have the same pressure difference as the attic floor/house ceiling. The supply leakage pressure exponent is a required input, but typically a value of 0.6 is used. The flow coefficient is calculated from the leakage airflow rate, assuming a reference pressure of 25 Pa and using the pressure exponent:

$$C_{soff} = \frac{Q_{ah}\alpha_s}{25^{n_s}} \tag{C29}$$

$$C_{roff} = \frac{Q_{ah}\alpha_r}{25^{n_r}} \tag{C30}$$

where, C_{soff} is the supply leak flow coefficient, Q_{ah} is the air handler flow, n_s is the supply leak pressure exponent and α_s is the supply leakage expressed as a fraction of air handler flow. C_{roff} is the return leak flow coefficient, n_r is the return leak pressure exponent and α_r is the return leakage expressed as a fraction of air handler flow.

Duct leaks with air handler on (M_{son} and M_{ron})
All the air handler on flows: air handler flow, duct leakage flows and register flows are converted from the input volumetric flows to mass flows using the indoor air density. The supply leak mass flow is added to the inflow into the attic and the return leaks are treated as airflows out of the attic. These are fixed mass flows independent of wind, stack or internal pressures and simply appear as mass flows in the mass balance equation.

Flow through Each Leak for the House

The flow coefficients for the ceiling, floor level leaks and walls are estimated as fractions of the total distributed leakage such that

$$C_d = \sum_{i=1}^{4} C_{f,i} + \sum_{i=1}^{4} C_{w,i} + C_c \tag{C31}$$

where $C_{f,i}$ is the floor level leakage below wall i, $C_{w,i}$ is the leakage in wall i and C_c is the ceiling leakage.

Furnace Flues and Fireplaces

Furnace flues and fireplaces are usually the largest openings in the building envelope and typically have a flow exponent, n_F, close to 0.5. The flue leakage coefficient, C_F, can be calculated from diameter, D_F, of the flue or fireplace assuming orifice flow, with a discharge coefficient of $K_D = 0.6$. The pressure coefficient of $Cp_F = -0.5$ is from Haysom and Swinton (1987). The change in wind velocity with height above grade may be significant for furnace flues that protrude above the reference eaves height. A corrected Cp_F is then given by:

$$Cp_F = (-0.5)\left(\frac{H_F}{H_e}\right)^{2p} \tag{C32}$$

where H_f is the flue top height and p is the exponent used in the atmospheric boundary layer wind profile (typically p=0.3 for urban surroundings and p=0.17 for rural sites). Shelter for the flue, $S_{U,F}$, is the shelter factor at the top of the flue. If the surrounding buildings and other obstacles are below the flue height then it is assumed that $S_{U,F} = 1$. If the surrounding obstacles are higher than the flue then the flue is sheltered and $S_{U,F}$ is calculated using Equation C15. The general pressure difference Equation C8 can be written specifically for the furnace flue as:

$$\Delta P_F = \Delta P_I - P_T \; H_F + P_U \; S_{U,F}^2 \; Cp_F \tag{C33}$$

and the mass flow rate, M_F, for the flue is given by Equation C1. Note that this is for an unheated flue or a natural draft furnace (flue without a draft inducing fan) well connected to the conditioned space. In new construction most furnace flues will be outside conditioned space in a well vented closet, garage or attic (or will be direct vented), in which case the flue leak is set to zero and only open fireplaces need to be considered, or we need to know the flow rate through the forced combustion fan for furnaces.

Floor Level Leakage

The leakage at floor level, $C_{f,i}$, is estimated as a fraction of the total distributed leakage and n_f is the same as n for the other distributed leaks. There are two cases of floor level leakage that require different assumptions about wind pressure effects. The cases depend on house construction.

a. Basements and Slab on Grade

In this case the total floor level leakage is split into four parts, one for each side of the building. On each side the floor level leakage is given the same Cp and S_U as the wall above it. For the i^{th} side of the building

$$\Delta P_{f,i} = \Delta P_I + Cp_i \; S_{U,i}^2 \; P_U - H_f \; P_T \tag{C34}$$

where floor height, H_f, is measured from grade level. For a house with a basement this is the height of the main level floor above grade and the leakage coefficient, $C_{f,i}$ includes the leakage around basement windows, dryer vents etc. The mass flow rate for these floor level leaks is given by Equation C1.

b. Crawlspaces (flow through house floor to and from the crawslpace)

As an estimate of the wind pressure in a crawl space the shelter and pressure coefficients for the four walls of the building are averaged. The average is weighted for non square plan buildings by the length of each side, L_i, so that for the i^{th} side.

$$Cp_f = \sum_{i=1}^{4} S_{U,i}^2 Cp_i \left(\frac{L_i}{L_\pi} \right) \tag{C35}$$

where L_π is the perimeter of the building (the sum of the L_i's) and then the pressure across the crawlspace is given by

$$\Delta P_f = \Delta P_I + Cp_f P_U - H_f P_T \tag{C36}$$

and the mass flow rate through the crawlspace leakage is given by Equation C1.

Ceiling Leakage

The ceiling flow coefficient C_c is estimated from the total distributed leakage and n_c is the same as n for the other distributed leaks. There are no wind pressures acting on the ceiling except indirectly through the flow balancing pressures ΔP_I (house) and $\Delta P_{I,a}$

(attic) because the ceiling is completely sheltered from the wind. The pressure across the ceiling includes the difference in attic and house buoyancy pressures

$$\Delta P_c = \Delta P_I - \Delta P_{I,a} - \rho_{out} g H_e \left(\frac{T_{in} - T_{out}}{T_{in}} - \frac{T_a - T_{out}}{T_a} \right) \tag{C37}$$

The mass flow rate through the ceiling is given by Equation C1.

Wall Leakage

For each wall $C_{w,i}$ is estimated from the total distributed leakage and the flow exponent, n, for each wall is n_d, the same as for the other distributed leaks. The vertical distributed leakage is treated the same way as attic pitched roof leakage.

Fan Flow

Fans are included in houses the same was as for attics: by using the naturally occurring pressures to determine the operating point on a fan curve.

Vent Leakage

The vent leakage is attributed to deliberately installed leakage sites that are separate from the background leakage. Multiple vents can be described, each with their own flow characteristics and each at a different location on the house envelope. Furnace and fireplace flues are treated separately as they may contain heated air that would produce a different stack effect for that leak only. Vents exiting through the roof use the same Cp and S_U as the furnace flue. The pressure difference and mass flow calculation is the same as for attic vents.

Flow through open Doors and Windows

The flowrates through door and window openings are determined by integrating the flow velocity profiles found by applying Bernoulli's equation along streamlines passing through the opening as shown by Kiel and Wilson (1986). For convenience the following parameters are defined

$$P_b = Cp\, S_U^2 U^2 - 2g\, H_b \left(\frac{T_{in} - T_{out}}{T_{in}} \right) + \frac{2\Delta P_I}{\rho_{out}} \tag{C38}$$

$$P_t = Cp\, S_U^2 U^2 - 2g\, H_t \left(\frac{T_{in} - T_{out}}{T_{in}} \right) + \frac{2\Delta P_I}{\rho_{out}} \tag{C39}$$

where Cp and S_U are for the surface that the opening is in

H_b = Height above grade of the bottom of the opening

H_t = Height above grade of the top of the opening

As with the integrated wall flows the mass flows in and out depend on H_{NL}, T_{in} and T_{out}. All of the possible cases for flow above and below H_{NL} are given in appendix D. Appendix D also contains a derivation for the flow in below H_{NL} for the case where H_{NL} falls in the opening and $T_{in}>T_{out}$, such that

$$M_{out} = (\rho_{out} \rho_{in})^{\frac{1}{2}} \frac{KWT_{in}}{3g(T_{in} - T_{out})} P_t^{\frac{3}{2}} \tag{C40}$$

$$M_{in} = \rho_{out} \frac{KWT_{in}}{3g(T_{in} - T_{out})} P_b^{\frac{3}{2}} \tag{C41}$$

Window and Door Flow Coefficient, K

The flow coefficient, K, accounts for reduction in flow due to flow contraction, viscous losses and interfacial mixing. An estimate for K that accounts for the variation in K due to interfacial mixing generated by atmospheric turbulence is given by Kiel and Wilson (1986) as

$$K = 0.400 + 0.0045 \,|T_{in} - T_{out}| \tag{C42}$$

The flow coefficient must be altered when the interface is near the top or the bottom of the opening so that the iterative solution of flow for the whole building does not have the neutral level oscillating just above and below the top or bottom of the opening. A first order approximation is to let K vary linearly in the top and bottom 10% of the opening between the value of K with the neutral level at 10% or 90% of the opening height and K = 0.6 at the edges of the opening. This is physically realistic because when the interface is near the top or the bottom of the opening the edges of the opening will interfere with the interfacial mixing process. This will make the flow look more like one way flow with an assumed orifice discharge coefficient, $K_D = 0.6$.

Grille Airflows

The supply and return grille airflows are determined by subtracting the leakage from the air handler flow. The volumetric flows are converted to mass flows using the indoor air density.

Airflow Solution Method

All of the flow equations for the house contain the difference between the inside and outside pressure, ΔP_I, that is the single unknown (or $\Delta P_{I,a}$ for the attic). To find ΔP_I all of the flow equations are combined into one equation that is the mass balance for air in the house:

$$\sum M = M_F + M_f + M_c + M_{\sup} + M_{ret} + M_{son} + M_{soff} + \sum_{i=1}^{4} M_{ws,i} + M_v + M_{fan} = 0 \quad \text{(C43)}$$

where the various mass flows are:
M_F : Flue
M_f : floor level leaks
M_c : Ceiling
M_{sup} : supply register air handler on
M_{ret} : return register air handler on
M_{soff} : supply register air handler off
M_{roff}: return register air handler off
M_V : sum of all passive vent flows
M_{fan} : is the sum of all the ventilation fans.

This equation for mass balance is non-linear with ΔP_I as the only unknown. To solve for ΔP_I, an iterative bisection technique was adopted because it is extremely robust and computational simple. This bisection search technique assumes that $\Delta P_I = 0$ for the first iteration and the mass inflow or outflow rates are calculated for each leak. At the next iteration ΔP_I is chosen to be +25 Pa if total inflow exceeds total outflow and -25 Pa if outflow exceeds inflow. Succeeding iterations use the method of bisection in which ΔP_I for the next iteration is reduced by half the difference between the last two iterations, thus the third iteration changes ΔP_I by ±6.25 Pa. The sign of the pressure change is positive if inflow exceeds outflow and negative if outflow is greater then inflow. The limit of solution is determined by stopping when the change in ΔP_I is < 0.01 Pa, which gives mass flow imbalances on the order of 0.001 Kg/s (or 4Kg/hour).

For the attic the mass balance equation is given by

$$\sum M = M_r + M_c + M_{soff} + M_{roff} + M_{son} + M_{soff} + \sum_{i=1}^{4} M_{s,i} + M_{v,a} + M_{fan,a} = 0 \qquad \textbf{(C44)}$$

where
M_r : sum of the in and the out flows through the pitched roof surfaces,
Mc : ceiling
M_{soff} : supply leak air handler off
M_{roff}: return leak air handler off
M_{son} : supply leak air handler on
M_{ron} : return leak air handler on
$M_{s,i}$ flow through soffit (or gable) component i.
$M_{fan,a}$: sum of the mass flows through all the attic fans
$M_{V,a}$: sum of the flows through all the attic vents.

As with the house all of the components of this mass balance equation contain the single unknown, $\Delta P_{I,a}$, the attic to outdoor pressure difference. The attic zone is solved using the same bisection technique as the house zone.

Zone Coupling

The house and attic zones are coupled by the flow through the ceiling and pressure difference across the ceiling. The house zone uses $\Delta P_{I,a}$ to calculate the mass flow through the ceiling. This mass flow is used in the mass flow balance by the attic zone to calculate a new $\Delta P_{I,a}$. This is an iterative procedure that continues until the change in mass flow through the ceiling from iteration to iteration is less than the magnitude of the house leakage coefficient divided by 10 or 0.0001 kg/s, whichever is larger.

Heat Transfer Model

A standard lumped heat capacity analysis is used, and solid material use the standard technique of splitting into surface and inner layers. The surface layer interacts by convection and radiation and the inner layer by conduction to the surface. The north and south sheathing are separated so that they may have different daytime solar gains. Forced convection heat transfer coefficients are used inside the attic using airflows calculated in the ventilation model. Radiation heat transfer inside the attic is simplified to three attic surface nodes: the attic floor and the two pitched roof surfaces plus the supply and return duct surfaces.

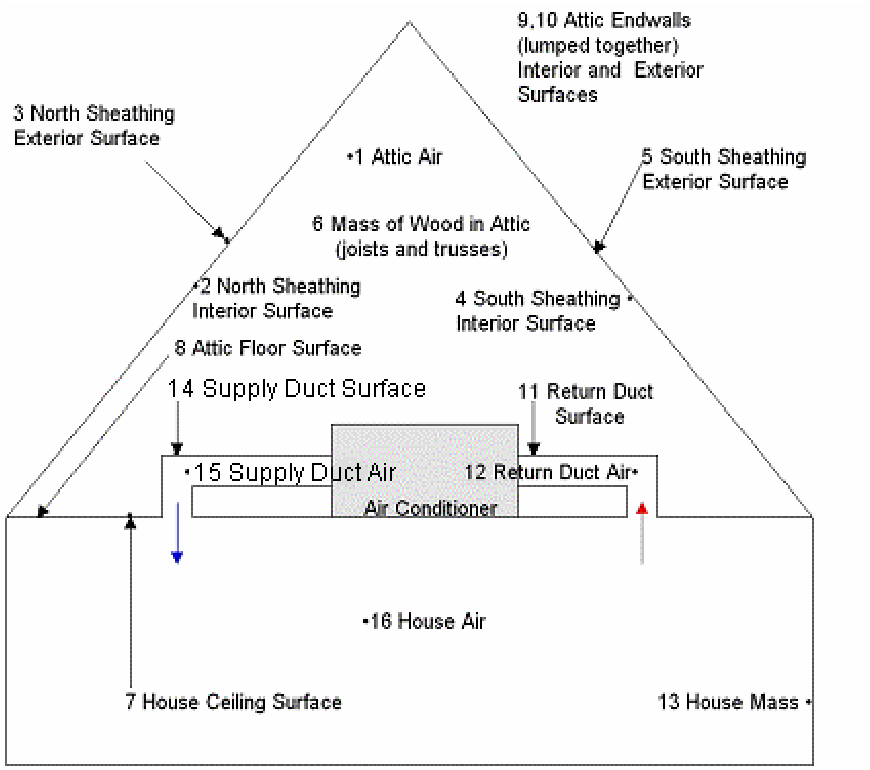


Figure C8: Nodes For Heat Transfer Model

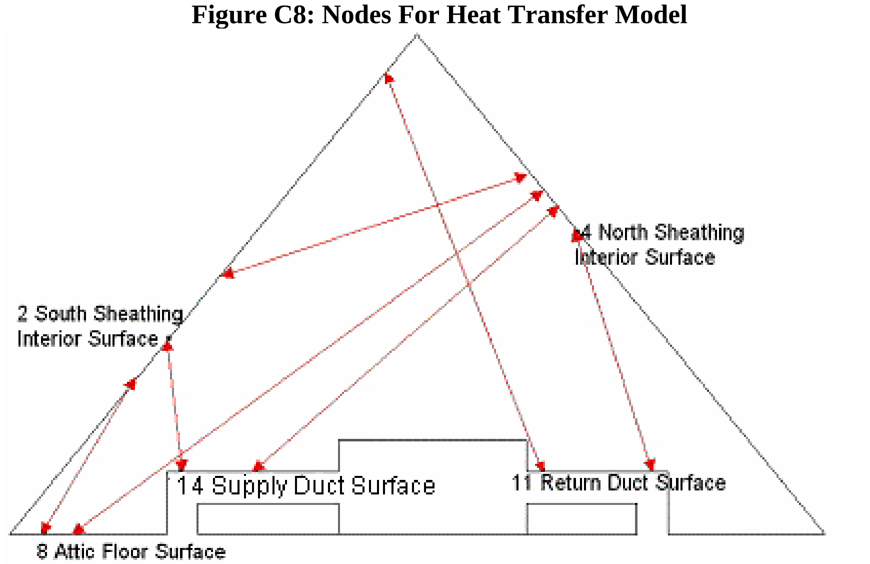


Figure C9: Radiation Transfer for Ducts in Attic

Attic and duct system heat transfer nodes:
1 = Attic Air
2 = Inner Surface1 Sheathing
3 = Outer Surface1 Sheathing
4 = Inner Surface2 Sheathing
5 = Outer Surface2 Sheathing
6 = All of the wood (joists, trusses, etc) lumped together
7 = Ceiling of the house
8 = Floor of the attic
9 = Inner End Wall
10 = Outer End Wall
11 = Return Duct Outer Surface
12 = Return Duct Air
13 = Mass of the house
14 = Supply Duct Outer Surface
15 = Supply Duct Air
16 = House Air

At each node, the rate of change of energy is equal to the sum of the heat fluxes

$$\rho_i V_i C_{sh,i} \frac{dT_i}{dt} = \sum q \tag{C45}$$

where ρ_i is the density [Kg/m^3] , V_i is the volume [m^3], $C_{sh,i}$ is the specific heat [J/KgK], T_i is temperature [K] and q are the heat fluxes [W]. The fluxes are due to convection, radiation and conduction heat transfer. The derivative in this equation is calculated using a finite difference approximation. Only the first term of the finite difference approximation is used so that the equation remains linear with temperature.

$$\rho_i V_i C_{sh,i} \frac{T_i^{\,j} - T_i^{\,j-1}}{\tau} = \sum q \tag{C46}$$

were j refers to the current timestep and j-1 the previous timestep and τ is the length of the time step. The energy balance is performed at each timestep j with the previous hour's (j-1) temperatures used to calculate the rate of change of energy at each node. This results in a linear system of 16 equations and 16 unknowns (the temperatures) that can be solved using simple matrix solutions.

Radiation Heat Transfer

Inside the Attic (Nodes 2,4, 8, 11 and 14)

For simplicity, this model assumes that the radiation heat transfer inside the attic can be simplified to five surfaces: attic floor, two pitched roof sections plus the supply and return duct surfaces. The calculation of radiation exchange inside the attic is based on heat exchange between non-blackbodies.

$$q_i = A_i h_{R,i-j} \left(T_i - T_j\right) + A_i h_{R,i-k} \left(T_i - T_k\right) \tag{C47}$$

where $h_{R,i-j}$ are radiation heat transfer coefficients from node i to node j that are calculated from

$$h_{R,i-j} = \frac{\sigma\left(T_i + T_j\right)\left(T_i^2 + T_j^2\right)}{\frac{1-\varepsilon_i}{\varepsilon_i} + \frac{1}{F_{i-j}} + \frac{\left(1-\varepsilon_j\right)A_i}{\varepsilon_j A_j}} \tag{C48}$$

where ε = emissivity of surface, A is the area of the body and σ is the Stephan-Boltzman constant that is equal to $5.669 * 10^{-8}$, and F_{i-j} are the view factors. These equations represent a linearized solution to the radiant heat transfer between three bodies: i, j and k.

The emissivity of surfaces found in building construction is given by ASHRAE (1989)(Chapter 37). For the inside sheathing surfaces a typical value for wood is ε = 0.90 and for the attic floor that is assumed to be covered with fibreglass insulation the typical emissivity glass (from ASHRAE (1989), Chapter 37) is used, ε = 0.94. The emissivity of glass is also typical of diffuse surfaces, and the fibreglass insulation is a diffuse surface due to its roughness. The geometry factors are determined from the attic dimensions and duct locations. For example, for ducts on the attic floor, it is assumed that 1/3 of the duct surface area sees each pitched roof surface and the remaining third of the duct surface area is not involved in radiation heat transfer.

Solar Radiation (Nodes 3 and 5)

Solar gains are only applied to the external sheathing surfaces. The energy transfer due to solar radiation is

$$q_R = A\alpha G \tag{C49}$$

where q_R is radiation heat transfer rate [W]
A = Surface area [m^2]
α = Surface absorbtivity
G = Total Solar Radiation [W/m^2], both direct and diffuse.

The radiant heat transfer properties (and thermal resistance) change depending on the attic sheathing material either: asphalt shingles (R=0.077, ε=0.91 α=0.92), white coated asphalt shingles(R=0.077, e=0.91 α=0.15), red clay tile (R=0.5, ε=0.58 α=0.92) and low emissivity coated clay tile(R=0.5, ε=0.5 α=0.92).

Radiant Exchange of Exterior Surfaces with Sky and Ground (Nodes 3 and 5)

In addition to the daytime solar gain the outside of the pitched roof sheathing has low temperature long wave radiant exchange with the sky and the ground. This exchange is responsible for cooling of the sheathing at night as it radiates energy to the cooler sky. On a cloudy night the cooling of the sheathing is reduced because the radiation exchange is with clouds that are warmer than the sky temperature. Both the clouds and the ground are assumed to be at the outside air temperature. The view factors that account for the proportion of sky, cloud or ground seen by the pitched roof surface are from Ford (1982). Cloud cover is assumed to be 0.5 for all cases because cloud cover data are generally not available or reliable.

The net radiation exchange for exterior pitched roof sheathing surfaces has the same form as Equations C41 and C42 for the internal radiation because this is a three body problem involving the roof surface, the sky and the ground and the clouds (which are assumed to be at the same temperature). The sky temperature T_{sky} depends on the water vapour pressure in the air. The view factors give the fraction of exposure to the ground (and clouds) and the sky for the pitched roof surfaces. Using the same view factors for both pitched roof surfaces assumes that the cloud cover is uniformly distributed over the sky.

Effective Sky Temperature for Radiation

The sky temperature, T_{sky}, is the equivalent temperature of an imaginary blackbody that radiates energy at the same rate as the sky. The effective sky temperature, T_{sky}, is a function of air temperature, T_{out}, and water vapour pressure P_v. Parmelee and Aubele (1952) developed the following empirical fit to measured data to estimate T_{sky} for horizontal surfaces exposed to a clear sky.

$$T_{sky} = T_{out}\left(0.55 + 5.68\times10^{-3}\sqrt{P_v}\right)^{0.25} \tag{C50}$$

where P_v is in Pascals and the temperatures are in Kelvin. Sample calculations show how T_{sky} can be very different from T_{out}. For example at T_{out} = 273K and 50%RH (so that P_v = 305 Pa) then T_{sky} = 245K, almost 30K difference.

Radiant Exchange of the Ceiling (Node 7) with the Room Below

This is modelled as a two body enclosed system where one body is the ceiling and the other body is the interior surfaces. The interior surfaces are assumed to be all at the same temperature as the inside air, T_{in}. The same linearization as for the pitched roof surfaces and the attic floor is applied so that the radiation heat transfer, $q_{R,7}$, is a linear function of temperature. The heat transfer coefficient is calculated based on the previous timestep temperatures.

Convection Heat Transfer

Natural and forced convection heat transfer coefficients are calculated based on surface temperatures and local air velocities. The natural convection heat transfer is given by

$$q_T = h_T A \Delta T \tag{C51}$$

where q_T is the free convection heat transfer rate [W]

h_T is the free convection heat transfer coefficient [W/m^{2}K] – this is given a fixed value of 3.2 based on a heated plate facing upwards.

A is the surface area

ΔT is the temperature difference

$$h_T = 3.2\left(\Delta T\right)^{\frac{1}{3}} \tag{C52}$$

To keep the heat transfer equations linear, ΔT is evaluated using the previous hours temperatures.

The house ceiling uses a convection heat transfer coefficient of 6 W/m^{2}Kwith the air handler off (based on values in ASHRAE Fundamentals 200, Chapter 3) and 9 W/m^{2}K with the air handler on (based on typical indoor air velocities). These same heat transfer coefficients are used for the exterior surfaces of ducts when they are inside the conditioned space.

Forced Convection

Forced convection heat transfer is calculated using:

$$h_{forced} = (18.192 - 0.37T_{film})U^{0.8} \tag{C53}$$

The constants are based on Nusselt correlations and the velocity, U, is based on local air velocities. For duct interior surfaces this is the average duct air velocity. For outside nodes (pitched roof surfaces) it is based on the windspeed. For the inside of attics a characteristic velocity is calculated based on attic envelope air leakage rates and the attic leakage area:

$$U_{atticconvection} = \frac{(Matticenvin - Matticenvout)}{\rho_{attic}\, 4Al_4} \tag{C54}$$

Where Al_4 is the four pascal attic leakage area. Note that Matticenvout will be a negative number hence the subtraction sign.

For the interior and exterior attic surfaces, the natural and forced convection coefficients are combined by cubing them and taking the cube root.

Equipment Capacity

The equipment capacity is added to the heat balance for the supply duct air (Node 15). The capacity includes the waste heat from the air handler. Currently this waste heat is a required input as there is no air handler performance model in REGCAP.

The capacity, the energy efficiency ratio (EER) and the power consumption (ratio of the capacity and EER) vary with the refrigerant charge, the coil temperature and the air handler flow. This model combines (Proctor.1999) with laboratory data from Texas A&M laboratory studies (Rodriguez et al. (1995)) to determine empirical correction factors that take into account the variation of incorrect charge of refrigerant as well as the temperature of the coil for three control types (capillary tube, orifice and thermostatic expansion valve (TXV)).

Refrigerant charge effects

In the following tables, CD is the charge deviation. So CD=-0.1 is a 10% undercharge.

Table C3. Charge Deviation impacts on air conditioner performance with a wet coil

Valve Type	Charge deviation capacity multiplier with a wet coil			
	CD<-0.316	**-0.316<=CD<=-0.15**	**-0.15<CD<=0**	**CD>0**
TXV	1+(1+CD-0.85)	1+(1+CD-0.85)	1	1
Cap Tube	0.4	1-6*CD^2	1-6*CD^2	1-CD*0.35
Orifice	0.4	1-6*CD^2	1-6*CD^2	1-CD*0.35
Valve Type	**Charge deviation EER Multiplier with a wet coil**			
	CD<=-0.15	**-0.15<CD<=-0.1**	**-0.1<CD<=0**	**CD>0**
TXV	1+(1+CD-0.85)*0.9	1	1	1-CD*0.35
Cap Tube	1+(1+CD-0.9)*1.35	1+(1+CD-0.9)*1.35	1	1-CD*09
Orifice	1	1	1	1-CD*0.25

Figures C10 through C13 show both the measured data from Rodriguez and the model in tables C3 and C4. The "old" model was based on a previous analysis of laboratory data and is not currently used in REGCAP.

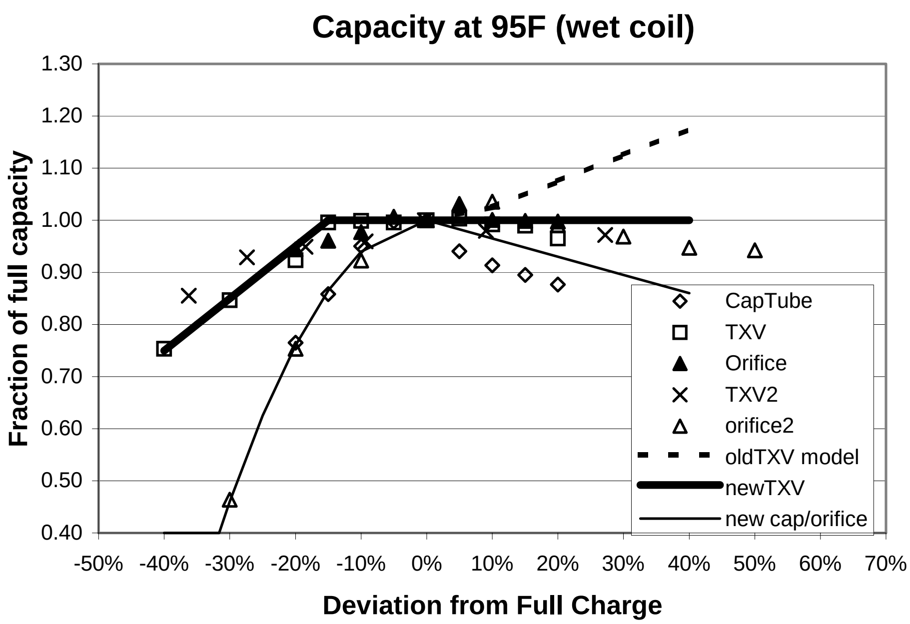


Figure C10. Wet Coil capacity variation with charge

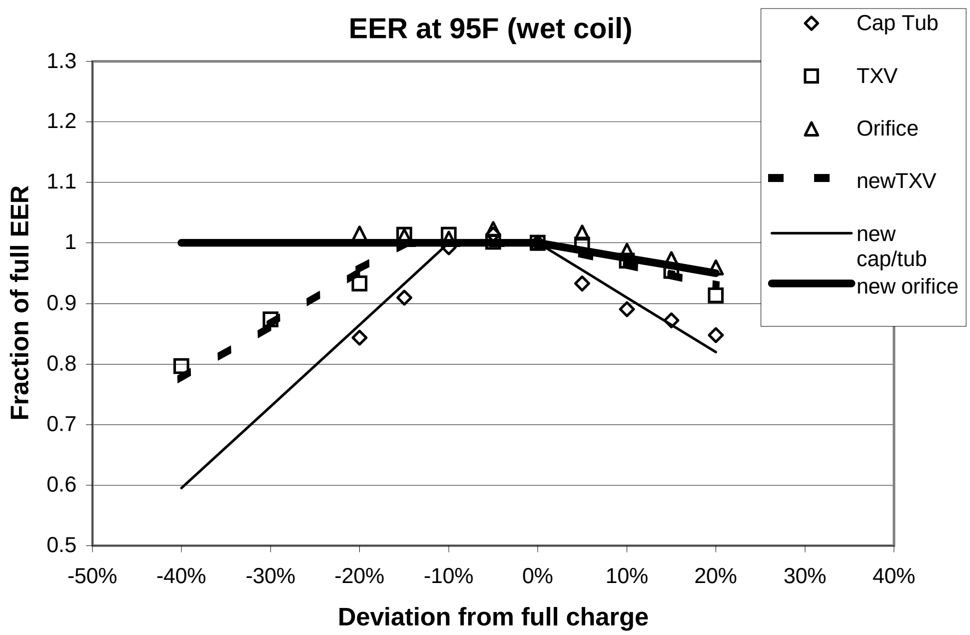


Figure C11. Wet Coil EER variation with charge

Table C4. Charge Deviation impacts on air conditioner performance with a dry coil

Valve Type	Charge deviation capacity multiplier with a dry coil		
	CD<-0.2	**-0.2<=CD<0**	**CD>=0**
TXV	1.2+CD	0.925	0.925
Orifice/cap tube	0.94+CD*0.85	0.94+CD*0.85	0.94-CD*0.15

Valve Type	Charge deviation EER multiplier with a dry coil	
	CD<0	**CD>=0**
TXV	1.04+CD*0.15	1.04-CD*0.35
Orifice/cap tube	1.05+CD*0.5	1.05-CD*0.35

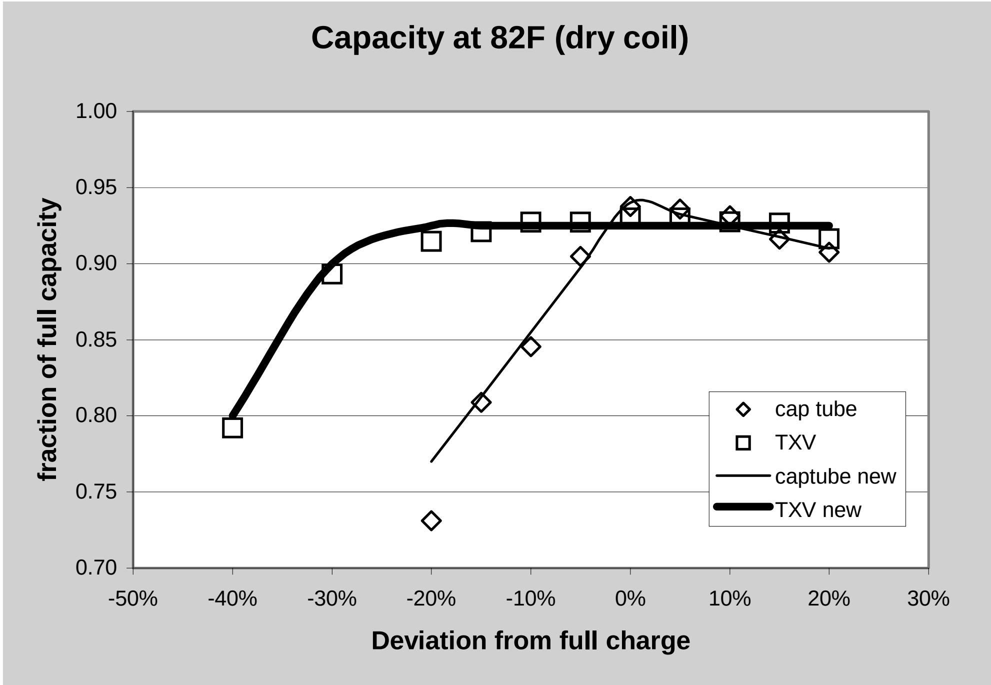


Figure C12. Dry Coil capacity variation with charge

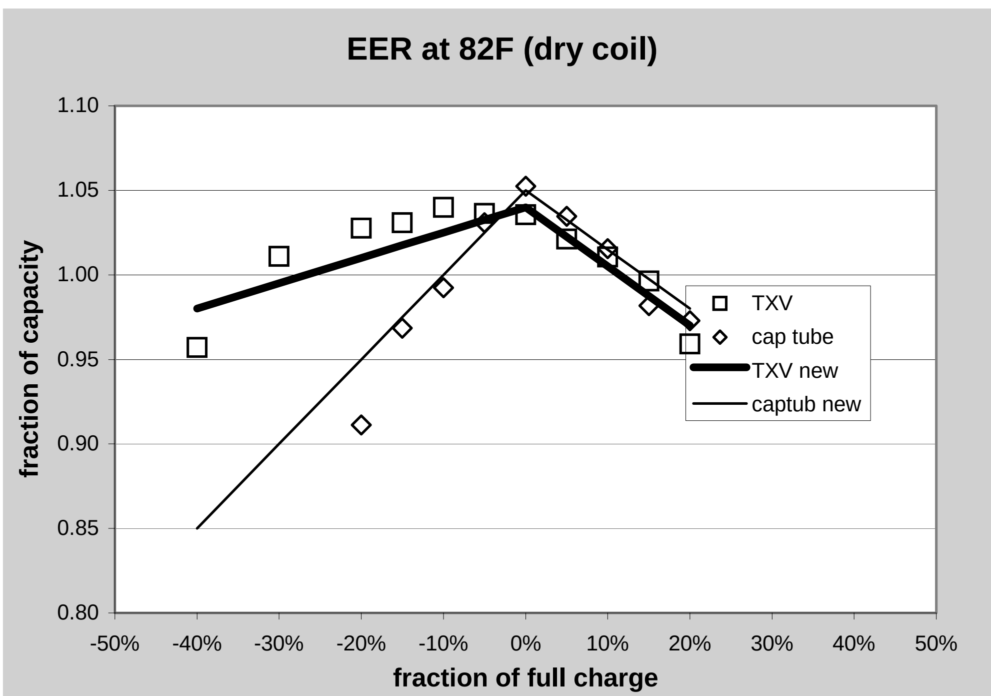


Figure C13. Dry Coil EER variation with charge

Outdoor air temperature

The outdoor air corrections are relative to the reference temperatures used for rating:

- Capacity:
  Correction = (-0.00007)*(T-Tref)^2-0.0067*(T-Tref)+1
- EER:
  Correction = (-0.00007)*(T-Tref)^2-0.0085*(T-Tref)+1

Tref is 95F (35ºC) for a wet coil and 82F (28ºC) for a dry coil.

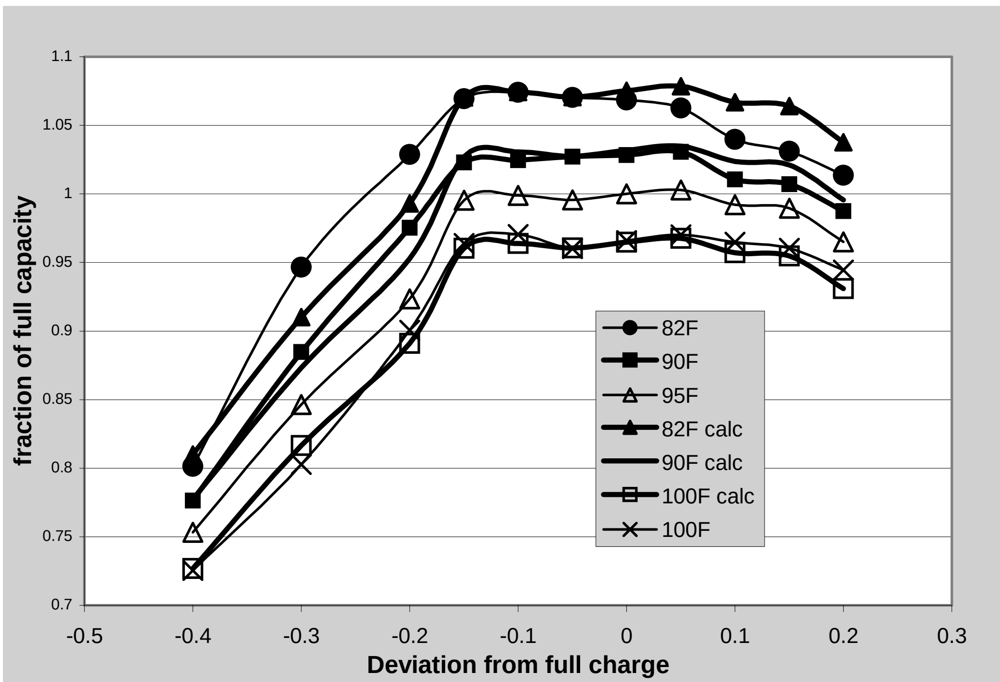


Figure C14. Dry Coil capacity variation with outdoor temperature

Air handler flow

The same multiplier is used for capacity and EER. These are taken from ASHRAE standard 152 and were developed for the standard by John Proctor from correlations to Texas A&M laboratory data.

- for TXV:

$$correction = 1.62 - 0.62\left(\frac{Qactual}{Qrecommended}\right) + 0.647\ln\left(\frac{Qactual}{Qrecommended}\right)$$

- for capillary tube and orifice:
- $$correction = 0.65 + 0.35\left(\frac{Qactual}{Qrecommended}\right)$$

Qrecommended is the airflow recommended by the manufacturer – typically 350 to 400 cfm/ton.

Node Heat Transfer Equations

In each of the following equations the subscript on temperature, T, refers to the node location and the superscript to the timestep.

Node 1. Attic Air

The attic air has convective (the h_U terms) heat transfer from all the interior attic surfaces - nodes 2, 4, 8, 6 and 9 as shown Figure C8. Although each convection term uses the same velocity, U_U, the different temperatures will change the film temperature, T_ε, and thus the heat transfer coefficient. In addition the convective flows in and out of the attic, M_a, and the flow through the ceiling, M_c, duct leakage and duct leak air handler off flows transport heat in and out of the attic air.

Nodes 2,3,4,5,9 and 10

These nodes all experience internal conduction with surface convection and radiation. The differences are that the exterior sheathing surfaces have daytime solar gains and nightime radiation cooling.
The areas of nodes 3 and 5 (exterior surfaces) are increased by 50% for tile roofs.

Node 6. Attic Joists and Trusses

The joists and trusses only exchange heat with the attic air by convection.

Nodes 7 and 8. House Ceiling/Attic Floor

The underside of the ceiling has radiant exchange with the inside surfaces of the house that are assumed to be at T_{in}, i.e. the same temperature as the air in the house. The house is assumed to have internal free convection and so the ceiling exchanges heat with the house air. There is also conduction through the ceiling to the floor of the attic.

The attic floor exchanges heat by radiation to the pitched roof surfaces, forced convection with the attic air and by conduction through the ceiling form the house below. The radiation terms are important because during high daytime solar gains the warm sheathing can raise the attic floor temperature above the attic air and reduce heat loss through the ceiling. Conversely cooler attic sheathing on clear nights will make the attic floor colder.

Node 11. Return duct external duct surface

Exchanges heat by convection plus the thermal resistance of the duct walls with the return duct air, by convection with the attic air and radiation with attic surfaces.

Node 12. Return Duct Air

The return duct with air handler on has air entering at indoor temperature plus leakage at attic temperature and air leaving at the air handler flow rate. There is also forced convection plus the thermal resistance of the duct walls between the return duct air and the return duct surface. With the air handler off the processes are the same but the airflow rate is determined by the leakage area of the duct leaks.

Node 13. House mass

The thermal mass of the house is an empirical approximation based on assuming that the first 5 cm of the concrete slab and 1 cm of the drywall all interact with the attic air. The surface area for heat transfer for the house thermal mass has been empirically adjusted to be 2.5 times the wall and floor surface area. 95% of the solar gain to the house (calculated from the direct and diffuse solar radiation, solar geometry and window area) goes to the thermal mass. The other 5% goes to the house air.

Node 14. Supply duct external duct surface

Exchanges heat by convection plus the thermal resistance of the duct walls with the supply duct air, by convection with the attic air and radiation with attic surfaces.

Node 15. Supply Duct Air

The supply duct with air handler on has air entering at the return temperature (at the air handler flow rate) and air leaving through leaks to the attic and also to the house. There is also forced convection plus the thermal resistance of the duct walls with the duct surface. With the air handler off the processes are the same but the airflow rate is determined by the leakage area of the duct leaks. The equipment capacity is added to the supply duct air (noting that cooling capacity is negative).

Node 16. House Air

House air exchanges energy by convection with the ceiling and the house internal mass. Airflows due to inflows and outflows through the envelope and register grilles are included. Care must be taken to ensure that the appropriate mass fluxes are used when the air handler is on or off and that the flow directions are tracked (particularly for the ceiling and duct air handler off flows) so that the correct air temperature is used for each airflow. The solar temperature is used together with the envelope UA to calculate the heat transfer through the house envelope. Solar loads are dealt with by having 5% of the solar gain go to the air in the house and the other 95% to the house mass. The solar gain is through windows only and includes a shading coefficient and the solar gain through the windows in each of the four cardinal directions. Any internal loads go directly to the house air.

Envelope load

$$Load = UA\left(t_{solair} - t_{in}\right) + 0.05 q_{solgain} \tag{C55}$$

where $q_{solgain}$ is the average solar radiation on the walls over four cardinal directions and includes any shading, and

$$t_{solair} = t_{out} + 0.03 q_{incidentsolar} \tag{C56}$$

$q_{incidentsolar}$ is the average incident solar radiation on each vertical surface for the four cardinal directions.

The factor 0.03 is from ASHRAE Fundamentals SI p. 26.5 (1993).

Solution of the Attic Heat Transfer Equations

At each node the rate of change of thermal energy is equated to the sum of the heat fluxes due to radiation, convection and conduction. This results in the above set of equations that are linear in temperature and must be solved simultaneously. This simultaneous solution is found using Gaussian elimination. When the temperatures have been calculated the attic air temperature (Node 1) is returned to the attic ventilation model so that a new attic ventilation rate can be calculated. This new ventilation rate is then used in the thermal model at the attic air node to calculate temperatures. This iterative process is continued until the attic air temperature changes by less than $0.1^{o}C$. Because the attic ventilation rates are relatively insensitive to the attic air temperature usually fewer than five iterations between thermal and ventilation models are required.

Typical House features

Building Envelope Characteristics		
Envelope Leakage Coefficient	0.057 m^3/sPa^n	Includes ceiling leakage
Envelope pressure exponent	0.65	
Eave height	19 ft (5.8 m)	This is from grade level for a two story house
Envelope Leakage Distribution	30% of leakage in the walls, 64% in the ceiling and 6% at grade level	Wall leakage is equally distributed on all four walls
Combustion appliance flues	N/a	No open flues between conditioned space and outside.
Building Volume	16000 ft^3 (453 m^3)	
Floor Area	2000 ft^2 (186 m^2)	
Plan Area	1000 ft^2 (93 m^2)	
Wind shelter for roof	None	
Continuous mechanical ventilation	None	
Window placement	90ft^2 (8.36 m^2) in each face (NSEW).	Used for solar gain calculations – even distribution on four walls. Based on fraction of floor area – for a total of 18% of floor area
Window SHGC	0.4	Includes medium color and weave interior drapes
Internal gains	2080 Btu/h (611 W)	

Attic Characteristics		
Attic Leakage Coefficient	0.105 m^3/sPa^n	
Attic leak pressure exponent	0.65	
Attic leakage distribution	10% in each pitched surface, 20% in each soffit, 20% in each gable end	
Attic Volume	656 ft^3 (61 m^3)	Based on roof pitch, plan dimensions
Attic vents	None	
Attic fans	None	
Roof Pitch	19 degrees	12/4 pitch
Roof Peak orientation	Parallel to the front of the house	This affects the wind pressure coefficients on the attic used to determine attic ventilation rate – but with this tight attic this is not very critical)
Roof peak height	23 ft (7.1 m)	Based on geometry
Roof deck insulation	None	
Gable end insulation	none	
Roof Type	Composition shingles absorbtivity = 0.75, emissivity = 0.9	This solar/radiation performance is from LBNL's online database (http://eetd.lbl.gov/coolroof/tile.htm#tile) .
Duct Location	All ducts in the attic	

Duct System Characteristics		
Supply duct material	R8 Flex (RSI 1.4)	
Return duct material	R8 Flex (RSI 1.4)	
Supply Leakage Fraction	3% of air handler flow	Based on 5% total from BSC spec.
Return Leakage Fraction	2% of air handler flow	
Supply leak pressure exponent	0.6	
Return leak pressure exponent	0.6	
Supply Duct Leakage Coefficient	3 cfm/Pa^n (0.0015 m^3/sPa^n)	Based on cooling leakage flow converted at 25 Pa reference pressure
Return duct leakage coefficient	2 cfm/Pa^n (0.00103 m^3/sPa^n)	Based on cooling leakage flow converted at 25 Pa reference pressure
Cooling Air Handler Flow	1000 cfm (0.47 m^3/s)	400 cfm/ton
Heating Air Handler Flow	970 cfm (0.46 m^3/s)	42 °F temp rise
Cooling Nominal capacity	2.5 Tons	
Cooling ARI capacity	30 kBtu/h (8.8 kW)	
Cooling EER	11	
Cooling control	TXV	
Cooling Charge	100%	
Heating Capacity	Varies by climate, see Table 6	
Heating AFUE	78%	Std furnace
Air Handler power consumption	Varies by system size, see Table 7	Assuming a standard PSC motor at 2cfm/W

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APPENDIX D

Example calculations for integrated ventilation flows

D.1 Example Calculations for Wall Flow

Consider the case where $T_{in} > T_{out}$ with counterflow where $H_{NL} > H_f$. For the section of the wall below H_{NL}

$$M_{w,i,in} = \frac{\rho_{out} C_{w,i}}{(H_e - H_f)} \int_{H_f}^{H_{NL}} \left(\Delta P_I + S_{U,i}^2 Cp_i P_U - z\, P_{T'} \right) dz \qquad \textbf{(D-1)}$$

Let

$$\Gamma(z) = \Delta P_I + S_{U,i}^2 Cp_i P_U - z P_{T'} \qquad \textbf{(D-2)}$$

then

$$d\,\Gamma = -P_{T'}\, dz \qquad \textbf{(D-3)}$$

Substituting equations D-2 and D-3 in D-1 gives

$$M_{w,i,in} = -\frac{\rho_{out} C_{w,i}}{(H_e - H_f) P_{T'}} \int_{\Gamma(z=H_f)}^{\Gamma(z=H_{NL})} \Gamma^n d\Gamma \qquad \textbf{(D-4)}$$

Integrating equation D-4:

$$M_{w,i,in} = -\frac{\rho_{out} C_{w,i}}{(H_e - H_f) P_{T'}} \frac{1}{(n+1)} \Gamma^{(n+1)} \Big|_{\Gamma(z=H_f)}^{\Gamma(z=H_{NL})} \qquad \textbf{(D-5)}$$

Substituting Γ back into equation D-5:

$$M_{w,i,in} = -\frac{\rho_{out} C_{w,i}}{(H_e - H_f) P_{T'}} \frac{1}{(n+1)} \left(\Delta P_I + S_{U,i}^2 Cp_i P_U - z P_{T'} \right)^{(n+1)} \Big|_{z=H_f}^{z=H_{NL}} \qquad \textbf{(D-6)}$$

Evaluating D-6 at the limits:

$$M_{w,i,in} = -\frac{\rho_{out} C_{w,i}}{(H_e - H_f) P_{T'}} \frac{\left(\left(\Delta P_I + S_{U,i}^2 Cp_i P_U - H_{NL} P_{T'} \right)^{(n+1)} - \left(\Delta P_I + S_{U,i}^2 Cp_i P_U - H_f\, P_{T'} \right)^{(n+1)} \right)}{n+1}$$

(D-7)

The first term is zero by definition of the neutral level and equation D-7 becomes

$$M_{w,i,in} = \frac{\rho_{out} C_{w,i}}{(H_e - H_f)P_{T'}} \frac{1}{(n+1)} \left(\Delta P_I + S_{U,i}^2 Cp_i P_U - H_f P_{T'} \right)^{(n+1)} \quad \textbf{(D-8)}$$

For convenience we define the pressure at the top and the bottom of the wall to be:

$$\begin{aligned} \Delta P_t &= \Delta P_I + S_{U,i}^2 Cp P_U - H_e P_T \\ \Delta P_b &= \Delta P_I + S_{U,i}^2 Cp P_U - H_f P_T \end{aligned} \quad \textbf{(D-9)}$$

Using the definition of the pressure at the bottom of the wall, ΔP_b, equation D-8 becomes:

$$M_{w,i,in} = \frac{\rho_{out} C_{w,i}}{(H_e - H_f)P_{T'}} \frac{\Delta P_b^{(n+1)}}{(n+1)} \quad \textbf{(D-10)}$$

D.2 The six possible cases for wall flow when $T_{in} \neq T_{out}$

For $T_{in} > T_{out}$. There are three possible cases:

Case 1. All wall above H_{NL} - all flow out

$$\mathbf{M_{w,i,in} = 0}$$

$$M_{w,i,out} = \frac{\rho_{in} C_{w,i} (\Delta P_t^{n+1} - \Delta P_b^{n+1})}{(H_e - H_f)P_{T'}(n+1)} \quad \textbf{(D-11)}$$

Case 2. All wall below H_{NL} - all flow in

$$\mathbf{M_{w,i,out} = 0}$$

$$M_{w,i,in} = \frac{\rho_{out} C_{w,i} (\Delta P_t^{n+1} - \Delta P_b^{n+1})}{(H_e - H_f)P_{T'}(n+1)} \quad \textbf{(D-12)}$$

Case 3. H_{NL} on the wall with flow in below H_{NL} and flow out above H_{NL}.

$$M_{w,i,out} = \frac{\rho_{in} C_{w,i} \Delta P_t^{n+1}}{(H_e - H_f)P_{T'}(n+1)} \quad \textbf{(D-13)}$$

$$M_{w,i,in} = \frac{\rho_{out} C_{w,i} \Delta P_b^{n+1}}{(H_e - H_f)P_{T'}(n+1)} \quad \textbf{(D-14)}$$

For $T_{in} < T_{out}$. There are also three possible cases:

Case 1. All wall above H_{NL} - all flow in

$$\mathbf{M_{w,i,out} = 0}$$

$$M_{w,i,in} = \frac{\rho_{out} C_{w,i} (\Delta P_b^{n+1} - \Delta P_t^{n+1})}{(H_e - H_f)P_{T'}(n+1)} \quad \textbf{(D-15)}$$

Case 2. All wall below H_{NL} - all flow out

$$\mathbf{M_{w,i,in} = 0}$$

$$M_{w,i,out} = \frac{\rho_{in} C_{w,i} (\Delta P_b^{n+1} - \Delta P_t^{n+1})}{(H_e - H_f) P_{T'} (n+1)} \qquad \textbf{(D-16)}$$

Case 3. H_{NL} on the wall with flow out below H_{NL} and flow in above H_{NL}.

$$M_{w,i,out} = \frac{\rho_{in} C_{w,i} (-\Delta P_b)^{n+1}}{(H_e - H_f) P_{T'} (n+1)} \qquad \textbf{(D-17)}$$

$$M_{w,i,in} = \frac{\rho_{out} C_{w,i} (-\Delta P_t)^{n+1}}{(H_e - H_f) P_{T'} (n+1)} \qquad \textbf{(D-18)}$$

D.3 Example calculation for flow through open doors and windows

Kiel and Wilson (1986) determined the flows through open doors and windows by integrating the vertical velocity profile in the opening. The velocities were found by applying Bernoulli's equation to streamlines passing through the opening. This assumes steady, irrotational and incompressible flow.

The different inside and outside air densities mean that the reference density used for the flow changes depending on the flow direction. Both inflow and outflow cases are derived below.

D.3.1 Flow into opening

For flow into the opening the flow density is ρ_{out} and the inflow velocity is given by Kiel and Wilson as

$$U_{in} = \left(\frac{2(P_{out,s} - P_{in,s})}{\rho_{out}} \right)^{\frac{1}{2}} \qquad \textbf{(D-19)}$$

where $P_{out,z}$ is the outside pressure, $P_{in,z}$ is the inside pressure, U_{in} is the flow velocity into the opening. The pressure difference driving the flow ($P_{out,z}$ - $P_{in,z}$) is found using Equation C9. Because both $P_{in,z}$ and $P_{out,z}$ depend on height, z, the inflow velocity is also a function of z. The total mass flow, M_{in}, must therefore be found by integrating U_{in} with height, z, over the area of inflow.

$$M_{in} = \rho_{out} KW \int U_{in} dz \qquad \textbf{(D-20)}$$

where W is the width of the opening and K is the flow coefficient for the opening that includes turbulent mixing effects. K is found using equation C42. The limits of integration for equation D-20 depend on the location of the opening with respect to the neutral level H_{NL}. The example calculation here is for inflow below H_{NL} where $T_{in} > T_{out}$ where H_{NL} falls within the opening. The limits of integration are then the height of the bottom of the opening, H_b, and the neutral level, H_{NL}. Substituting equation C9 for the pressure difference (and assuming that air is an ideal gas so that the density differences are expressed in terms of temperature differences) and equation D-19 in equation D-20 gives

$$M_{in} = \rho_{out} KW \int_{z=H_b}^{z=H_{NL}} \left(CpS_U^2 U^2 - \frac{2gz(T_{in} - T_{out})}{T_{in}} + \frac{2\Delta P_I}{\rho_{out}} \right)^{\frac{1}{2}} dz \qquad \textbf{(D-21)}$$

Let

$$\Gamma(z) = CpS_U^2 U^2 - \frac{2gz(T_{in} - T_{out})}{T_{in}} + \frac{2\Delta P_I}{\rho_{out}} \qquad \textbf{(D-22)}$$

then

$$d\,\Gamma = -\frac{2g(T_{in} - T_{out})}{T_{in}} dz \qquad \textbf{(D-23)}$$

Substituting equations D-22 and D-23 into D-21

$$M_{in} = -\rho_{out} KW \frac{T_{in}}{2g(T_{in} - T_{out})} \int_{\Gamma(z=H_b)}^{\Gamma(z=H_{NL})} \Gamma^{\frac{1}{2}} d\,\Gamma \qquad \textbf{(D-24)}$$

Integrating equation D-24 gives

$$M_{in} = -\rho_{out} KW \frac{T_{in}}{2g(T_{in} - T_{out})} \frac{2}{3} \Gamma^{\frac{3}{2}} \Big|_{\Gamma(z=H_b)}^{\Gamma(z=H_{NL})} \qquad \textbf{(D-25)}$$

Substituting Γ back into equation D-24 yields

$$\begin{aligned} M_{in} = & -\rho_{out} \frac{KWT_{in}}{3g(T_{in} - T_{out})} \left(CpS_U^2 U^2 - \frac{2gH_{NL}(T_{in} - T_{out}))}{T_{in}} + \frac{2\Delta P_I}{\rho_{out}} \right)^{\frac{3}{2}} \\ & + \rho_{out} \frac{KWT_{in}}{3g(T_{in} - T_{out})} \left(CpS_U^2 U^2 - \frac{2gH_b(T_{in} - T_{out})}{T_{in}} + \frac{2\Delta P_I}{\rho_{out}} \right)^{\frac{3}{2}} \end{aligned} \qquad \textbf{(D-26)}$$

where the first term is zero by definition of H_{NL} in equation D-10. The final equation for flowrate is

$$M_{in} = \rho_{out} \frac{KWT_{in}}{3g(T_{in} - T_{out})} \left(CpS_U^2 U^2 - \frac{2gH_b(T_{in} - T_{out})}{T_{in}} + \frac{2\Delta P_I}{\rho_{out}} \right)^{\frac{3}{2}} \qquad \textbf{(D-27)}$$

D.3.2 Flow out of opening

In this case the density of the flow is ρ_{in} and equation D-19 becomes

$$U_{in} = \left(\frac{2(P_{out,s} - P_{in,s})}{\rho_{in}} \right)^{\frac{1}{2}} \quad \textbf{(D-28)}$$

For flow out of the opening $P_{in,z} > P_{out,z}$ and the sign of the pressure difference term is negative. Therefore U_{in} will also be negative which implies outflow. This agrees with the convention applied to the other ventilation model leaks where inflow is positive and outflow is negative. The pressure difference driving the flow, $(P_{out,z} - P_{in,z})$, is found using equation C9. The total mass flow, M_{out}, is found by integrating U_{in} over the area of inflow.

$$M_{out} = \rho_{in} KW \int U_{in} dz \quad \textbf{(D-29)}$$

The limits of integration for equation D-29 depend on the location of the opening with respect to the neutral level H_{NL}. The example calculation here is for inflow above H_{NL} where $T_{in} > T_{out}$ where H_{NL} falls within the opening. The limits of integration are H_{NL} and the top of the opening, H_t. Substituting equation C9 for the pressure difference (and assuming that air is an ideal gas so that the density differences are expressed in terms of temperature differences) into equation D-28 for velocity and then using this in equation D-29 gives

$$M_{out} = \rho_{in} KW \int_{z=H_{NL}}^{z=H_t} \left(\frac{\rho_{out}}{\rho_{in}} Cp S_U^2 U^2 - \frac{\rho_{out}}{\rho_{in}} \frac{2gz(T_{in} - T_{out})}{T_{in}} + \frac{2\Delta P_I}{\rho_{in}} \right)^{\frac{1}{2}} dz \quad \textbf{(D-30)}$$

Unlike the inflow case the densities do not cancel in the wind and stack pressure terms. Let

$$\Gamma(z) = \frac{\rho_{out}}{\rho_{in}} Cp S_U^2 U^2 - \frac{\rho_{out}}{\rho_{in}} \frac{2gz(T_{in} - T_{out})}{T_{in}} + \frac{2\Delta P_I}{\rho_{in}} \quad \textbf{(D-31)}$$

then

$$d\,\Gamma = -\frac{\rho_{out}}{\rho_{in}} \frac{2g(T_{in} - T_{out})}{T_{in}} dz \quad \textbf{(D-32)}$$

Substituting equations D-31 and D-32 into D-30

$$M_{out} = -\rho_{in} KW \frac{\rho_{in}}{\rho_{out}} \frac{T_{in}}{2g(T_{in} - T_{out})} \int_{\Gamma(z=H_{NL})}^{\Gamma(z=H_t)} \Gamma^{\frac{1}{2}} d\,\Gamma \quad \textbf{(D-33)}$$

Integrating equation D-33 gives

$$M_{out} = -\rho_{in} KW \frac{\rho_{in}}{\rho_{out}} \frac{T_{in}}{2g(T_{in} - T_{out})} \frac{2}{3} \Gamma^{\frac{3}{2}} \Big|_{\Gamma(z=H_{NL})}^{\Gamma(z=H_t)} \quad \textbf{(D-34)}$$

Substituting Γ back into equation D-34 yields

$$M_{out} = -\rho_{in}\frac{\rho_{in}}{\rho_{out}}\frac{KWT_{in}}{3g(T_{in}-T_{out})}\left(\frac{\rho_{out}}{\rho_{in}}CpS_U^2U^2 - \frac{\rho_{out}}{\rho_{in}}\frac{2gH_t(T_{in}-T_{out}))}{T_{in}} + \frac{2\Delta P_I}{\rho_{in}}\right)^{\frac{3}{2}}$$

$$+\rho_{in}\frac{\rho_{in}}{\rho_{out}}\frac{KWT_{in}}{3g(T_{in}-T_{out})}\left(\frac{\rho_{out}}{\rho_{in}}CpS_U^2U^2 - \frac{\rho_{out}}{\rho_{in}}\frac{2gH_{NL}(T_{in}-T_{out})}{T_{in}} + \frac{2\Delta P_I}{\rho_{in}}\right)^{\frac{3}{2}} \quad \textbf{(D-35)}$$

where the second term is zero by definition of H_{NL} in equation C10. The final equation for flowrate out of the opening is

$$M_{out} = (\rho_{out}\rho_{in})^{\frac{1}{2}}\frac{KWT_{in}}{3g(T_{in}-T_{out})}\left(CpS_U^2U^2 - \frac{2gH_t(T_{in}-T_{out})}{T_{in}} + \frac{2\Delta P_I}{\rho_{out}}\right)^{\frac{3}{2}} \quad \textbf{(D-36)}$$

D.4 The seven possible cases of door/window flow

P_b and P_t are flow coefficients based on the pressures at the bottom and the top of the opening and are defined in equation D-9.

For $T_{in} > T_{out}$ there are three possible cases.

Case 1. All opening above H_{NL} - all flow out

$$\mathbf{M_{in} = 0}$$

$$M_{out} = (\rho_{out}\rho_{in})^{\frac{1}{2}}\frac{KWT_{in}}{3g(T_{in}-T_{out})}\left(P_t^{\frac{3}{2}} - P_b^{\frac{3}{2}}\right) \quad \textbf{(D-37)}$$

Case 2. All opening below H_{NL} - all flow in

$$\mathbf{M_{out} = 0}$$

$$M_{in} = \rho_{out}\frac{KWT_{in}}{3g(T_{in}-T_{out})}\left(P_b^{\frac{3}{2}} - P_t^{\frac{3}{2}}\right) \quad \textbf{(D-38)}$$

Case 3. H_{NL} in the opening with flow in below H_{NL} and flow out above H_{NL}.

$$M_{out} = (\rho_{out}\rho_{in})^{\frac{1}{2}}\frac{KWT_{in}}{3g(T_{in}-T_{out})}P_t^{\frac{3}{2}} \quad \textbf{(D-39)}$$

$$M_{in} = \rho_{out}\frac{KWT_{in}}{3g(T_{in}-T_{out})}P_b^{\frac{3}{2}} \quad \textbf{(D-40)}$$

For $T_{out} > T_{in}$ there are three possible cases.

Case 1. All opening above H_{NL} - all flow in

$M_{out} = 0$

$$M_{in} = \rho_{out} \frac{KWT_{in}}{3g(T_{in} - T_{out})} \left(P_t^{\frac{3}{2}} - P_b^{\frac{3}{2}} \right) \quad \textbf{(D-41)}$$

Case 2. All opening below H_{NL} - all flow out

$M_{in} = 0$

$$M_{out} = (\rho_{out}\rho_{in})^{\frac{1}{2}} \frac{KWT_{in}}{3g(T_{in} - T_{out})} \left(P_b^{\frac{3}{2}} - P_t^{\frac{3}{2}} \right) \quad \textbf{(D-42)}$$

Case 3. H_{NL} in the opening with flow out below H_{NL} and flow in above H_{NL}.

$$M_{out} = (\rho_{out}\rho_{in})^{\frac{1}{2}} \frac{KWT_{in}}{3g(T_{in} - T_{out})} (-P_b)^{\frac{3}{2}} \quad \textbf{(D-43)}$$

$$M_{in} = \rho_{out} \frac{KWT_{in}}{3g(T_{in} - T_{out})} (-P_t)^{\frac{3}{2}} \quad \textbf{(D-44)}$$

For $T_{in} = T_{out}$

In this case there is wind effect only and H_{NL} is undefined. It is not necessary to integrate a velocity profile over the opening height and an orifice flow equation is used to compute the flow through the opening.

$$M = \rho K_D W(H_t - H_b) \sqrt{\frac{2\Delta P}{\rho}} \quad \textbf{(D-45)}$$

where $\rho = \rho_{in}$ for outflow and $\rho = \rho_{out}$ for inflow, and $\Delta P = P_{out,z} - P_{in,z}$. The sign of ΔP determines if the flow is in or out. Following the same convention as for other leaks, a positive ΔP results in inflow and a positive mass flow, M. The pressure difference, ΔP, across the opening is found using equation C9. Then

$$M = \rho K_D W(H_t - H_b) \left(CpS_U^2 U^2 + \frac{2\Delta P_I}{\rho_{out}} \right)^{\frac{1}{2}} \quad \textbf{(D-46)}$$

where W is the width of the opening and K_D is a flow coefficient assumed to be 0.6.

D.5 The seven possible cases of attic pitched roof surface flow

For $T_a = T_{out}$

$P'_{T,a} = 0$ and there are wind pressures only. $H_{NL,r}$ is undefined and for each roof pitch

$$\Delta P_r = P_{I,a} + S_U^2 Cp P_U \quad \textbf{(D-47)}$$

and

$$M_r = \rho \frac{C_r}{2} (\Delta P_r)^{n_r} \quad \textbf{(D-48)}$$

where $M_r = M_{r,in}$ and $\rho = \rho_{out}$ for inflow (ΔP_r positive)
$M_r = M_{r,out}$ and $\rho = \rho_{in}$ for outflow (ΔP_r negative).

For $T_a \neq T_{out}$

To find the total flow through each roof pitch the flow must be integrated to allow for the change in pressures. The limits of integration for pressure are found at the roof peak height, H_p, and eave height, H_e and are

$$\Delta P_p = \Delta P_{I,a} + Cp S_U^2 P_U - H_p P_{T',a} \quad \textbf{(D-49)}$$

$$\Delta P_e = \Delta P_{I,a} + Cp S_U^2 P_U - H_e P_{T',a} \quad \textbf{(D-50)}$$

For $T_a > T_{out}$ there are three possible cases:
Case 1. All roof pitch above $H_{NL,r}$ - all flow out

$$\mathbf{M_{r,in} = 0}$$

$$M_{r,out} = \frac{\rho_a \frac{C_r}{2} (\Delta P_p^{(n_r+1)} - \Delta P_e^{(n_r+1)})}{(H_p - H_e) P_{T',a} (n_r + 1)} \quad \textbf{(D-51)}$$

Case 2. All roof pitch below $H_{NL,r}$ - all flow in

$$\mathbf{M_{r,out} = 0}$$

$$M_{r,in} = \frac{\rho_{out} \frac{C_r}{2} (\Delta P_p^{(n_r+1)} - \Delta P_e^{(n_r+1)})}{(H_p - H_e) P_{T',a} (n_r + 1)} \quad \textbf{(D-52)}$$

Case 3 $H_{NL,r}$ on the pitched surface with flow in below $H_{NL,r}$ and flow out above $H_{NL,r}$.

$$M_{r,out} = \frac{\rho_a \frac{C_r}{2} \Delta P_p^{(n_r+1)}}{(H_p - H_e) P_{T',a} (n_r + 1)} \quad \textbf{(D-53)}$$

$$M_{r,in} = \frac{\rho_{out} \frac{C_r}{2} \Delta P_e^{(n_r+1)}}{(H_p - H_e) P_{T',a} (n_r + 1)} \quad \textbf{(D-54)}$$

For $T_{out} > T_a$ there are three possible cases:
Case 1. All roof pitch above $H_{NL,r}$ - all flow in.

$$\mathbf{M_{r,out} = 0}$$

$$M_{r,in} = \frac{\rho_{out} \frac{C_r}{2} (\Delta P_e^{(n_r+1)} - \Delta P_p^{(n_r+1)})}{(H_p - H_e) P_{T',a} (n_r + 1)} \quad \textbf{(D-55)}$$

Case 2. All roof pitch below $H_{NL,r}$ - all flow out.

$$\mathbf{M_{r,in} = 0}$$

$$M_{r,out} = \frac{\rho_a \frac{C_r}{2} (\Delta P_e^{(n_r+1)} - \Delta P_p^{(n_r+1)})}{(H_p - H_e) P_{T',a} (n_r + 1)} \quad \textbf{(D-56)}$$

Case 3. $H_{NL,r}$ on the pitched surface with flow out below $H_{NL,r}$ and flow in above $H_{NL,r}$.

$$M_{r,out} = \frac{\rho_a \frac{C_r}{2} (-\Delta P_e)^{(n_r+1)}}{(H_p - H_e) P_{T',a} (n_r + 1)} \quad \textbf{(D-57)}$$

$$M_{r,in} = \frac{\rho_{out} \frac{C_r}{2} (-\Delta P_p)^{(n_r+1)}}{(H_p - H_e) P_{T',a} (n_r + 1)} \quad \textbf{(D-58)}$$

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