Systems Research on Residential Ventilation

Armin Rudd and Joseph Lstiburek, Building Science Corporation

ABSTRACT

Over a decade of applied research and field experience in residential whole-house ventilation is discussed. System pros and cons are evaluated and preferred systems are highlighted. The initial cost of most ventilation system variants are presented on a relative, cost-index basis. Energy consumption and operating cost results from simulations of different central-fan-integrated supply (CFIS) ventilation systems, including new control methods, are compared to that of a reference house without mechanical ventilation. High-performance houses with controlled mechanical ventilation and without use of setback thermostat cost less to operate than standard houses without ventilation and with setback thermostat in warm climates with significant cooling. Electrical energy cost to operate the ventilation systems ranged from 2% to 11% of total HVAC system annual operating cost in the warm climates of Houston and Phoenix, and ranged from 1% to 6% in Charlotte, Kansas City, Seattle, and Minneapolis. Central fan operation of 20 minutes per hour minimum, for ventilation air distribution and whole-house comfort mixing, amounted to 15% to 20% of annual hours, and annual cost ranged between $0 in Minneapolis to $85 in Houston.

Introduction

One purpose of this paper is to provide a historical chronology of our experience with residential mechanical ventilation in production homebuilding since the 1990's. The focus is on beyond-code to high-performance home programs. Issues framing the development of our process are discussed, and explanations of our current positions are given as a point of reference. A second purpose is to present recent ventilation system performance simulations conducted for six U.S. cities in six International Energy Conservation Code (IECC) climate zones (IECC 2006). Residential mechanical ventilation consists of two main categories: 1) local exhaust for removal of concentrated pollutants; and 2) whole-house dilution ventilation for dispersed pollutants. This paper pertains to the second category. Detailed review of common residential ventilation technologies can be found in Rudd (1998), Rudd (2006), and Russell et al. (2007).

Discussion of Past Applied Research and Field Experience

Phase 1

In the early 1990's, a number of whole-house mechanical ventilation systems and control strategies were evaluated with different homebuilders in different climates. While ideal from an engineering and performance point of view, we found that separately ducted, balanced heat recovery and energy recovery systems were not able to overcome the high initial cost barrier for most builders. Supply ventilation was identified as being a preferred whole-house ventilation strategy over exhaust. With supply ventilation, outside air comes from a known source that can be planned based on the expected air quality; outside air can be filtered, tempered with recirculation air, and conditioned if desired; and outside air supply pressurizes rather than depressurizes the building interior with respect to outdoors. House pressurization helps rather than hinders combustion appliance venting; it avoids drawing polluted air from a garage, crawl space, attic, or from below floor slabs in soil contact; it avoids airflow related moisture/mold problems in humid climates under cooling conditions; it works better with natural forces of stack and wind to increase air exchange (houses are generally depressurized under natural conditions, exhaust ventilation works harder against this depressurization than supply ventilation) (BSC 2007); and it can be applied in all climates including cold climates with airtight enclosure and insulated sheathing, or cavity spray foam insulation.

Inline-type supply ventilation fans were tried first. Those required a field constructed filter arrangement. Then, a manufacturing partner developed a new product to integrate the supply fan and air filtration. However, challenges for both systems were as follows:

- Required tempering of outdoor air, using 1 part outdoor air mixed with 2 to 3 parts recirculation air (less in mild-dry climates, more in cold and humid climates). This extra airflow for tempering meant more fan power and energy consumption.
- Separate ducting required at least two pickups (outside air and recirculation air) and at least one supply. If outside air was supplied directly into the main supply or return ducts of a central system, then condensation could occur in cool ducts after cooling cycles. If outside air was supplied directly to conditioned space, then at least two supply points were needed to avoid comfort complaints.
- Required easy access to change another filter
- Occupant resistance to the fan running continuously
- Initial cost was still too high for most production builders

Some homebuilders that were resistant to the cost of the separate supply fan ventilation system chose to install single-point exhaust-only ventilation instead. Concerns soon arose with that system as follows:

- Dust marking on light carpets
- Activation of living space carbon monoxide alarms due to air coming from garages where car engines were the source
- Lack of filtration, evidenced by increased dirt/dust particles settling on window sills, and countertops
- Lack of ventilation air distribution, evidenced by odor buildup in bedrooms remote from the exhaust fan location
- Objection to continuous fan noise

At about the same time, cooling system right-sizing began to take hold with a number of builders who were interested in constructing high-performance homes following a systems engineered design. Over-sized cooling systems were common since the HVAC industry often felt that was necessary in order to compensate for many unknowns in building performance factors out of their control. By not over-sizing cooling systems, cost savings could be realized that nearly paid for the building enclosure improvements that were the basis of warranting the right-sizing. Such improvements included:

- Tightening the air pressure boundary of the building enclosure against air leakage, which: 1) eliminated draft discomfort complaints and frozen pipes due to air infiltration; 2) eliminated excessive energy demand and energy consumption due to high infiltration; and 3) made controlled mechanical ventilation the dominant force for outdoor air exchange
- Moving the entire space conditioning air distribution system inside conditioned space
- High-performance glass (SHGC < 0.35, U-value < 0.35), having benefits of: 1) significantly reduced cooling and heating loads; 2) increased comfort near outside walls; 3) allowing more compact duct systems which made it easier to get ducts inside conditioned space which eliminated duct losses
- More uniform and better performing insulation with sprayed or netted-and-blown cellulose, or spray polyurethane foam

Interestingly, along with high customer satisfaction, due to the improved energy efficiency and comfort performance, came even higher expectations. In other words, homeowners recognized a good thing, and wanted more of it. While most comfort issues were resolved through the efficiency measures described above and proper mechanical system design, temperature variations between the thermostat location and other floors or distant rooms could still be large. While the on-time for heating and cooling equipment is related to the load and equipment capacity, a challenging characteristic of high-performance houses is that the off-time between cooling and heating cycles is longer than for less efficient houses. Consistent air mixing for thermal comfort is needed as internally generated loads vary spatially and as solar heat gain moves with the sun moves around the house. This is especially evident during part-load periods when thermostat demand is low.

Phase 2

These factors brought us to consider central-fan-integrated supply (CFIS) ventilation, such as was already being used to comply with Washington State ventilation code requirements, and in HUD Code (manufactured) homes. These ventilation systems involved outside air ducted to the return side of a central space conditioning system air handler. The amount of outside air supplied throughout the house by the air handler was dependent on the amount of thermostat driven fan runtime for heating and cooling plus an amount controlled by a timer that periodically energized the fan without accounting for prior heating and cooling operation. Those systems showed problems with:

- overlapping and excessive fan runtime that drove up operating cost and caused unusual/objectionable fan operation sequences; and
- fan operation after shut-down of the cooling compressor causing unwanted evaporation of condensed water from the cooling coil, contributing to poor indoor humidity control in humid climates.

A new controller was then developed that built on the central-fan-integrated supply ventilation system strategy but accounted for prior fan operation due to heating and cooling. It assured a programmable minimum fan runtime and avoided fan-only operation for a programmable time after a cooling cycle ended.

This improved CFIS ventilation system, with central fan cycling that took into account prior cooling and heating operation, was first used in high-efficiency production homes in the Chicago area in 1996. The design outside air flow was 10 cfm per person, with the number of people being equal to the number of bedrooms plus one. The system operated on a 33% duty cycle (10 minutes every half hour), so the intermittent outside air flow rate was three times the continuous ventilation requirement. During the first winter season, we found that some homeowners complained that the air in their house was too dry. Our monitored data showed relative humidity to be around 20% RH. We then reduced the outside air flow rate by an amount that accounted for a low level of natural air exchange (about 0.07 ach) during the two-thirds time that the fan was off (Rudd 1998). With the low level infiltration adjustment for the fan off-cycle, the averaged intermittent mechanical ventilation rate was equivalent to a continuous rate of about 7 cfm per person. This generally required a 6" diameter outside air duct rather than the previous 8" duct. At that lower ventilation flow rate, complaints of wintertime dryness went away and complaints of odor buildup did not arise. That result has persisted for over a decade representing thousands of installations. It should be noted that a publication by Fonorow et al. (2007) stated that he found acceptable results in hundreds of installations in the warm-humid climate with even a much lower ventilation rate, using "runtime ventilation", which was a central-fan-integrated supply ventilation system with a 4" diameter outside air duct and no central fan cycling (operation only coincident with thermostat demand for heating and cooling).

Use of the CFIS ventilation system then began in high-efficiency production homes in the predominantly cooling climates of Las Vegas and Tucson. There it was confirmed that the fan cycling strategy for delivery and distribution of ventilation air also solved a common thermal comfort problem of room-to-room temperature variation by making the space conditions more homogeneous. Room-to-room temperature variations were reduced to the extent that builders began to guarantee not more than 3^{o}F variation from the thermostat to any other room.

Similar to Chicago, where a symptom of over-ventilation was found, in a Tucson project, a symptom of under-ventilation was found. Due to lack of adherence to the specified ventilation system design, the average ventilation air flow was about 25% of the design amount. The outside air duct was connected to the return air system in such a way that only about half the design amount of outside air was being drawn in (insufficient negative pressure), and the timer was set so that the minimum fan on-time was only half the design amount (10 minutes per hour instead of 10 minutes per half hour). Occupants complained of odor buildup and elevated interior moisture in winter. When the problems were corrected, and the ventilation amount was increased to the design amount, the occupant complaints were resolved.

Above-code production homebuilding projects with CFIS ventilation then began to take hold in locations across the U.S. In the very cold climates, the system was often combined with exhaust ventilation to create a balanced system that minimized any effect on house pressure with respect to outdoors. Use of the CFIS system became a standard for the Building Science Consortium, Building America projects (Rudd & Lstiburek 2001), and for the Environments for Living™ national private sector program by Masco which offered an energy-use and comfort guarantee. That includes over 150,000 such houses since 1996.

An additional control strategy (outside air damper cycling) was developed to operate a motorized damper in the outside air duct such that the damper would stop the ventilation air supply if the central system fan was on longer than the programmed time for ventilation. In that way, excessive ventilation would not occur during periods of extended fan runtime (such as when recovering from heating setpoint setback or cooling setup, or during peak heating and cooling periods, or if an occupant switched the fan from Auto to On).

A number of CFIS ventilation system ducting configurations were developed for different equipment installations commonly used in practice (Rudd & Lstiburek 2007). These configurations depended on factors such as the air handler location, the location and number of central return duct terminations, the location and type of air filtration, and accessibility to route the outside air duct to a planned fresh air location. Lab measurements of outside air flow rates using realistic configurations were compared to field measurements resulting in charts to guide designers, installers, and raters in the process of design and field verification. Research conducted by Rudd & Lstiburek (2000) showed that ventilation systems that used a programmed minimum amount of whole-house distribution and mixing via the central air distribution system had much less room-to-room variation in outside air delivery than systems that did not. A publication by Moyer et al. (2004) compared seven ventilation systems, all tested in the same lab house in Cocoa, Florida. The CFIS system with 33% fan cycling showed the lowest cooling and ventilation power (watts) usage as a function of temperature difference across the building envelope. Guidelines were developed to help HVAC contractors steer away from potential problems and to assure repeatable and successful results (Rudd 2006).

Phase 3

In 2003, ASHRAE published its first Standard dealing only with residential ventilation. 2004 and 2007 versions have followed (ASHRAE 2007). The ASHRAE 62.2 Standard created a challenge in that it required about two times the mechanical ventilation airflow that we had been successfully using for more than five years in tens of thousands of houses. So, with a desire to support the Standard in principle, the question to be answered was: Should we increase the minimum fan duty cycle, or increase the size of the outside air duct to increase the flow rate?

Increasing the minimum fan duty cycle much above 33% was determined to be not a good option for a number of reasons:

- The neighborhood of 33% duty cycle fits well with normal system cycling during the heart of the heating and cooling seasons. Staying at or below that duty cycle maximizes "free" ventilation air distribution that occurs along with heating and cooling operation. Going much above that starts to add significantly to extra fan runtime which increases energy consumption. Fan energy consumption can also be significantly reduced through use of electronically commutated motor (ECM) blowers compared to permanent split capacitor (PSC) motor blowers. However, an ECM blower adds about $300 to $350 to equipment cost.
- The higher the fan duty cycle the more potential for unwanted re-evaporation of moisture from wet cooling coils in humid climates. This has been shown to be an important factor with constant fan operation, but a minimal factor at fan duty cycle rates that generally match normal cooling system operation (Henderson et al. 2007)

Increasing the size of the outside air duct to increase the intermittent ventilation flow rate was also determined to be less than desirable for the following reasons:

- Doubling the ventilation flow rate could increase the cooling equipment cost in some cases by pushing the equipment size to the next higher half-ton or one-ton increment
- Two 6" or one 8" outside air duct would normally be required to achieve the higher ventilation flow rate. Additional wall penetrations cost more, and large penetrations can be aesthetically objectionable.

The successful solution was to upgrade any bathroom fan in the house to meet the ASHRAE Standard 62.2 mechanical air flow and sound level requirements (i.e. 1 sone or less). This would add about $50 to $75 initial cost, but the exhaust plus CFIS system would still provide top performance at minimal cost compared to other options. Existence of a compliant exhaust fan and an on/off switch meets the Standard regardless of how much the fan runs. The fan should at least be activated by the occupants anytime the bathroom is in use. The CFIS system would still provide assurance of full distribution of ventilation, along with whole-house mixing for thermal comfort, but the exhaust ventilation system would be the ventilation system of record for ratings, or for code compliance if the ASHRAE Standard 62.2 was required by local code.

A new ventilation controller has recently been developed with enhanced capabilities to integrate the functions of a bathroom exhaust fan (or other ventilation fan such as an HRV) and central-fan-integrated supply ventilation with or without fan and damper cycling. The amount of outside air provided by the air handler can be dependent on normal heating and cooling operation or a minimum amount can be assured by the ventilation controller. The ventilation controller communicates with a wall switch that operates a separate ventilation fan (exhaust or HRV/ERV). Programmed operation of the separate ventilation fan can be coincident with CFIS, for intermittent balanced ventilation, or can provide continuous ventilation by alternating between CFIS and exhaust or balanced. The separate ventilation fan can be operated both manually and automatically, independent of or dependent on the central system operation.

This control approach has the benefit of optimizing low operating costs. Fully distributed ventilation is provided at no additional cost by the air handler as it operates in response to normal calls for heating and cooling, then low-cost exhaust fan operation (using an existing bath fan) can automatically provide the balance of ventilation as needed. A minimum amount of air handler runtime can be programmed to maintain whole-house thermal comfort mixing and ventilation air distribution.

Ventilation System Initial Cost

For production (high volume) new construction, the initial cost of purchased equipment and installation is often the most significant factor affecting ventilation system choices. The longer-term impacts on both the building and occupants are then factored in to create an overall perceived value as the basis of choosing a ventilation system design.

For example, upgrading from a standard builder-grade bathroom exhaust fan to a quiet higher-quality bathroom exhaust fan is clearly the lowest initial cost. But to gain whole house distribution, without adding fans or ductwork, a value decision might be made to add a fan cycling control on the central system fan to assure periodic mixing throughout the house. The initial cost of a fan cycling control is about the same as one additional upgraded bathroom fan or one additional duct run (inlet or supply).

Continuing with that example, to avoid problems sometimes associated with continuous exhaust, such as the unknown source of unfiltered/unconditioned air and dust marking on carpets, an additional value decision might be made to use supply ventilation with an outside air duct to the central system return for at least part of the time.

Table 1 gives an approximate ranking of initial cost for a number of ventilation systems. The ranking is relative to the initial cost of a single-point upgraded bathroom exhaust fan (low sone rating), which has a ranking of 1.0. Costs were derived from best estimates of current contractor pricing to new construction production builders. The cost factor ranking system yields a non-dated comparison (assuming relative costs remain similar over time).

The ranking trends show a first group of systems that are between 1 and 4 times the cost of the single-point exhaust upgraded bath fan. Central-fan-integrated supply without central fan cycling has a cost factor of 1, meaning that it costs about the same as a single upgraded bathroom exhaust fan. Adding fan cycling to that yields a cost factor of 2. Adding motorized damper cycling control to that yields a cost factor of 3. A second group is in the 8 to 14 cost factor range. Those systems include exhaust and supply systems with remote fans. The last group includes single-point and multi-point Heat Recovery Ventilators and Energy Recovery Ventilators which have cost factors in the 15 to 21 range.

Table 1. Initial Cost Factors for Ranking Various Whole-House Ventilation Systems Relative to Single-Point Exhaust

Description	Variants	Initial Cost Factor
Single-point exhaust, upgraded bath fan	no central fan cycling	1.0
	central fan cycling	2
Central-fan-integrated supply	no central fan cycling	1
	central fan cycling	2
	central fan+damper cycling	3
Multi-point exhaust, 2 upgraded bath fans	no central fan cycling	2
	central fan cycling	3
Central-fan-integrated supply + single-point exhaust	no central fan cycling	2
	central fan cycling	3
	central fan+damper cycling	4
Single-point supply, remote fan	no central fan cycling	8
	central fan cycling	9
Multi-point exhaust, remote fan	3 pickup points	12
	4 pickup points	13
Multi-point supply, remote fan	3 supply points	12
	4 supply points	14
Single-point HRV (1 supply, 1 exhaust)	no central fan cycling	15
	central fan cycling	16
Single-point ERV (1 supply, 1 exhaust)	no central fan cycling	16
	central fan cycling	17
Multi-point HRV	1 exhaust, 3 supply points	18
	2 exhaust, 3 supply points	20
Multi-point ERV	1 exhaust, 3 supply points	20
	2 exhaust, 3 supply points	21

System Performance Simulations

Computer simulations were performed to analyze the typical HVAC operating cost of different central-fan-integrated supply (CFIS) ventilation systems. Data presented here are a subset of a larger simulation study report (BSC 2007) where complete details of the computer model can be found. An important feature of the model was a short (1 minute) time step which allowed accurate tracking of the control strategies. Six U.S. cities in different IECC Climate Zones were evaluated. Those cities and climate zones were: Houston 2A; Phoenix 2B, Charlotte 3A, Kansas City 4A, Seattle 4C, Minneapolis 6A.

The standard house was a 2000 ft^2 two-story house meeting the 2006 IECC code requirements, but the high-performance house had additional improvements, including:

- entire space conditioning air distribution system located inside conditioned space;
- glazing U-value and SHGC equal to 0.35; and
- about one-half the building enclosure leakage of the standard house (3.0 air changes per hour at 50 Pa pressure differential (ach50) instead of 5.8 ach50).

As a reference case, System 0 was a standard house modeled without whole-house mechanical ventilation. All of the other cases were a high-performance house with mechanical ventilation. Only Systems 5 and 5a did not meet the ASHRAE Standard 62.2 air flow rate requirement. System 5 supplied 50 cfm of outside air through the central system whenever the air handler was on for heating or cooling. System 5a was the same except it assured that the air handler was on and supplying ventilation air for a minimum of 20 minutes per hour, and the outside air damper would close (stopping the supply of outside air) if the central fan runtime exceeded 20 minutes per hour. Systems 6 and 6a were the same as Systems 5 and 5a, respectively, except that a 50 cfm exhaust fan operated continuously to assure that the ASHRAE Standard 62.2 flow rate was always met. System 7 supplied 50 cfm of outside air through the central system whenever the air handler was on for heating or cooling, and exhausted 50 cfm whenever the air handler was off. System 7a was the same as System 7 except that the air handler was on (and supplying ventilation air) for a minimum of 20 minutes each hour, and the outside air damper would close (stopping the supply of outside air) if the central fan runtime exceeded 20 minutes per hour.

Table 3 lists the building enclosure and duct system parameters for the standard IECC house and the high-performance houses. Table 4 lists the central heating and cooling system parameters including capacity (sized using ACCA Manual J version 8 [ACCA 2006]), airflow rate, and fan power for the standard IECC house and the high-performance houses. Table 5 lists the ventilation fan parameters including the power draw at the stated rated static pressure. Table 6 lists the electric and natural gas utility rates used to calculate operating cost.

The thermostat setpoints for the standard house were modeled with a nighttime heating setback from 70°F to 68°F and a daytime cooling setup from 76°F to 80°F, while the thermostat setpoints for the high-performance house were modeled as constant 70°F for heating and constant 76°F for cooling. Constant thermostat setpoints are often used in high-performance houses because, with the thermal air distribution ducts inside conditioned space, there is less practical incentive for using setback and setup thermostat control compared to the inconveniences. There are also benefits to using constant setpoints related to cooling equipment sizing since a tendency to over-size equipment for quicker temperature recovery after setup is unnecessary.

Table 3. Building Enclosure and Duct System Parameters for the Standard and High-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 SHGC	Glazing U-value	Insulation Ceiling	Insulation Wall	Insulation Foundation	Insulation 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 SHGC	Glazing U-value	Insulation Ceiling	Insulation Wall	Insulation Foundation	Insulation 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

Table 4. Central Heating and cooling system parameters Including Capacity, Air Flow, and Fan Power for the Standard IECC Houses and the High-Performance Houses

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

	Heating AFUE	Heating Input Capacity (kBtu/h)	Heating Output Capacity (kBtu/h)	Heating Fan Flow (cfm)	Heating Temp Rise (F)	Heating Fan Power (W)	Cooling SEER	Cooling Total Capacity (kBtu/h)	Cooling Fan Flow (cfm)	Cooling Fan Power (W)	Fan cycling Fan Flow (cfm)	Fan cycling 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

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

	Heating AFUE	Heating Input Capacity (kBtu/h)	Heating Output Capacity (kBtu/h)	Heating Fan Flow (cfm)	Heating Temp Rise (F)	Heating Fan Power (W)	Cooling SEER	Cooling Total Capacity (kBtu/h)	Cooling Fan Flow (cfm)	Cooling Fan Power (W)	Fan cycling Fan Flow (cfm)	Fan cycling 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

Table 5. Ventilation Fan Parameters

Fan description	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		45
Kitchen exhaust	Ventamatic Nuvent RH160	160		99
Whole-house				
Continuous Exhaust	Panasonic FV-07VQ2	50	0.25	18.1
Central air handler	(used 0.5 W/cfm, indicative of 0.5 inch w.c. external static pressure)			

Table 6. Electric and Natural Gas Utility Rates

Location	Electric rate ($/kW-h)	Gas rate ($/therm)
Houston	0.115	1.55 (national avg used for all)
Phoenix	0.095	
Charlotte	0.093	
Kansas City	0.085	
Seattle	0.069	
Minneapolis	0.066	
source: www.eia.doe.gov, October 2005		

As shown in the data presented in Table 7, not all ventilation systems provided the same service in terms of annual average air change rate (including the combined effects of ventilation and infiltration). The more airtight high-performance house with ventilation Systems 5 and 5a had 20% to 35% lower annual average air change rate than the leakier standard house without mechanical ventilation (System 0). The annual average air change rate for System 5 was about 45% less than that of Systems 6 and 7 in Houston and Phoenix, ranging down to about 30% less in Minneapolis. The annual average air change rate for the standard house ranged from about 25% less than that of Systems 6 and 7 in Houston and Phoenix to about 4% more in Minneapolis.

Table 7 shows the simulation results for each climate. Annual energy-use is broken out by 1) heating therm; 2) heating kW-h, which includes heating fan electric use and heating therm converted to kW-h; 3) heating cost; 4) cooling kW-h, which includes cooling compressor and fan electric use; 5) cooling cost; 6) ventilation kW-h, which includes electric use for operating the kitchen and bath local exhaust fans, the whole-house exhaust fan, and the central air handler fan whenever it was used for ventilation air distribution only; 7) ventilation cost; 8) total kW-h, which includes heating and cooling electric use and heating therm converted to kW-h; 9) total heating, cooling, and ventilation cost; and 10) annual average air change rate (combined ventilation and infiltration).

In Seattle and Kansas City, the annual HVAC operating cost goes up from the standard house to the high-performance house with ventilation System 5 because the low solar heat gain glass increases heating load more than it reduces cooling load in those heating dominated climates. For those locations, that effect is greater than the reduced air change rate effect. The operating cost further increases with Systems 6 and 7 because of the increased air change rate due to exhaust fan operation. In all of the other locations, annual HVAC operating cost goes down from the standard house to the high-performance house with CFIS (System 5), then increases for the other systems with more air exchange.

Table 7. HVAC Energy-Use Simulation Results for Houston, Phoenix, Charlotte, Kansas City, Seattle, and Minneapolis

Ventilation System Number	Description	Operating cost and average air change rate for 2000 ft² higher performance house — Houston									
		heat			cool		vent		total		annual
		(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	avg ach
0	No ventilation, 6 ach50	427	12881	704	3609	415	56	6	16547	1125	0.18
5	CFIS	382	11476	624	1868	215	56	6	13400	845	0.12
5a	CFIS+20/60	373	11188	608	1977	227	850	98	14014	933	0.14
6	CFIS+Exh cont	398	11935	649	2148	247	216	25	14299	921	0.24
6a	CFIS+20/60+Exh cont	387	11614	631	2263	260	994	114	14871	1006	0.24
7	CFIS+Exh switch	398	11956	650	2157	248	185	21	14298	919	0.24
7a	CFIS+20/60+Exh swit	387	11620	632	2264	260	927	107	14811	999	0.24

Ventilation System Number	Description	Operating cost and average air change rate for 2000 ft² higher performance house — Phoenix									
		heat			cool		vent		total		annual
		(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	avg ach
0	No ventilation, 6 ach50	328	9891	535	6624	629	56	5	16571	1170	0.18
5	CFIS	307	9235	499	4060	386	56	5	13352	890	0.12
5a	CFIS+20/60	296	8904	481	4124	392	925	88	13953	961	0.13
6	CFIS+Exh cont	324	9744	527	4231	402	216	21	14191	949	0.24
6a	CFIS+20/60+Exh cont	313	9425	509	4311	410	1067	101	14804	1020	0.24
7	CFIS+Exh switch	323	9732	526	4231	402	177	17	14141	945	0.24
7a	CFIS+20/60+Exh swit	312	9401	508	4309	409	999	95	14710	1012	0.24

Ventilation System Number	Description	Operating cost and average air change rate for 2000 ft² higher performance house — Charlotte									
		heat			cool		vent		total		annual
		(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	avg ach
0	No ventilation, 6 ach50	756	22787	1231	1895	176	56	5	24737	1412	0.20
5	CFIS	682	20467	1102	905	84	56	5	21428	1191	0.13
5a	CFIS+20/60	667	20029	1079	977	91	739	69	21745	1238	0.15
6	CFIS+Exh cont	708	21247	1144	1035	96	216	20	22498	1260	0.24
6a	CFIS+20/60+Exh cont	694	20836	1122	1106	103	885	82	22827	1307	0.25
7	CFIS+Exh switch	708	21235	1144	1030	96	180	17	22446	1256	0.24
7a	CFIS+20/60+Exh swit	695	20848	1123	1108	103	818	76	22774	1302	0.24

Ventilation System Number	Description	Operating cost and average air change rate for 2000 ft² higher performance house — Kansas City									
		heat			cool		vent		total		annual
		(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	avg ach
0	No ventilation, 6 ach50	934	28149	1514	1847	157	56	5	30053	1676	0.24
5	CFIS	1011	30352	1629	1105	94	56	5	31514	1727	0.17
5a	CFIS+20/60	990	29713	1594	1146	97	610	52	31469	1744	0.18
6	CFIS+Exh cont	1046	31385	1684	1221	104	216	18	32822	1806	0.27
6a	CFIS+20/60+Exh cont	1026	30794	1652	1293	110	755	64	32841	1826	0.26
7	CFIS+Exh switch	1046	31400	1685	1219	104	166	14	32784	1803	0.27
7a	CFIS+20/60+Exh swit	1026	30794	1652	1290	110	680	58	32763	1820	0.26

Ventilation System Number	Description	Operating cost and average air change rate for 2000 ft² higher performance house — Seattle									
		heat			cool		vent		total		annual
		(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	avg ach
0	No ventilation, 6 ach50	730	22007	1174	256	18	56	4	22319	1196	0.24
5	CFIS	802	24053	1282	79	5	56	4	24188	1291	0.16
5a	CFIS+20/60	793	23804	1268	85	6	495	34	24384	1308	0.18
6	CFIS+Exh cont	841	25230	1344	90	6	216	15	25536	1365	0.25
6a	CFIS+20/60+Exh cont	832	24957	1330	97	7	638	44	25691	1380	0.25
7	CFIS+Exh switch	841	25233	1344	88	6	182	13	25502	1363	0.25
7a	CFIS+20/60+Exh swit	832	24975	1331	99	7	578	40	25652	1377	0.25

Table 7. HVAC Energy-Use Simulation Results for Houston, Phoenix, Charlotte, Kansas City, Seattle, and Minneapolis (cont'd)

Ventilation System Number	Description	Operating cost and average air change rate for 2000 ft² higher performance house — Minneapolis									
		heat			cool		vent		total		annual
		(therm)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	(kW-h/yr)	($/yr)	avg ach
0	No ventilation, 6 ach50	1310	39484	2103	794	52	56	4	40335	2159	0.29
5	CFIS	1314	39542	2105	345	23	56	4	39942	2132	0.20
5a	CFIS+20/60	1290	38802	2066	375	25	429	28	39604	2119	0.20
6	CFIS+Exh cont	1363	41001	2183	389	26	216	14	41605	2223	0.28
6a	CFIS+20/60+Exh cont	1337	40222	2142	414	27	581	38	41216	2207	0.28
7	CFIS+Exh switch	1362	40989	2182	388	26	163	11	41540	2219	0.28
7a	CFIS+20/60+Exh swit	1337	40228	2142	414	27	506	33	41148	2203	0.28

High-performance homes in hot-dry and hot-humid climates, even with the highest electric rates, cost far less to operate than high-performance homes in colder climates. Even in Houston and Phoenix, the cost of heating at the minimum required efficiency of 78% AFUE exceeds the cost of cooling at the minimum efficiency of 13 SEER.

Electrical energy cost to operate the ventilation systems ranged from 2% to 11% of total HVAC system annual operating cost in the warm climates of Houston and Phoenix, and ranged from 1% to 6% in Charlotte, Kansas City, Seattle, and Minneapolis. For a given climate, there was less than $150 per year difference in total HVAC operating cost between all of the ventilation options evaluated. There was less than $100 per year difference in total cost between Systems 5a through 7a. There was less than $10 per year difference between Systems 6, 6a (with continuous exhaust operation) and Systems 7, 7a (with the exhaust fan switching off while the central fan was on), but it is expected that avoiding objections to continuous exhaust fan operation will have significant benefits in customer acceptance of the more efficient switched system.

In Phoenix, Houston, and Charlotte, the high-performance house with all options of whole-house mechanical ventilation had lower operating cost than the standard house without mechanical ventilation. That was mainly due to locating the ducts inside conditioned space and the low solar heat gain glass.

In Minneapolis, System 5 and 5a had lower operating cost than the standard house because of the lower average air exchange rate. Systems 6, 6a, 7, and 7a in Minneapolis showed slightly higher operating cost than the standard house even though the annual average air change rate was nearly the same. This was due to: 1) the low solar heat gain glass in the high performance house reduced heat gain in winter; 2) the constant heating and cooling thermostat setpoints used for the high-performance houses versus the programmed heating setback and cooling setup used for the standard house; and 3) half the ducts in the standard house were already within the building thermal enclosure (insulated basement), so the benefit of locating ducts inside conditioned space was only realized for the other half of the ducts, which were in the attic in the standard house.

In Kansas City and Seattle, the high-performance house operating cost was always higher than the standard house because: 1) the average air change rate was higher with mechanical ventilation; 2) the constant, rather than programmed, heating and cooling setpoints; and 3) half the ducts in the standard house were already within the building thermal enclosure (insulated basement) for Kansas City.

Breaking out central fan cycling cost (i.e., the additional cost of ensuring the central fan runs at least 20 minutes per hour for ventilation and mixing), including the effect on heating and cooling, one can see that central fan cycling in Minneapolis saved energy. That is because gas heating was offset by heat from the central fan, and at 78% furnace efficiency, the cost of natural gas per delivered kW-h was slightly higher than the electric rate in Minneapolis. The cost of central fan cycling in Seattle and Kansas City was less than $20 per year. The low cost was due to: 1) for most of the year, heat from the central fan is not lost but contributes to heating; 2) electric rates were not high; and 3) there was fairly consistent demand for heating in Seattle and for both heating and cooling in Kansas City. The cost of central fan cycling in Charlotte was about $50 per year. The highest central fan cycling costs were in Phoenix and Houston at $70 to $85 per year, due to higher electric rates and because heat from the central fan adds to the cooling load. Use of efficient ECM fans could reduce central fan cycling costs significantly for all systems.

The total annual hours of central air handler activity, also converted to the annual air handler runtime fraction, is shown in Table 8. The percentage of annual central air handler activity time was then broken down into heating, cooling, and ventilation. The runtime fractions for the standard houses and the high-performance houses without central fan cycling were in the range of about 0.25 to 0.30. The runtime fractions for the high-performance houses with minimum central air handler operation of 20 minutes per hour were in the range of 0.40 to 0.45. The reason that the annual runtime fraction was higher than the controlled minimum of 0.33 (20 minutes per hour) was due to periods where heating and cooling runtime exceeded the minimum.

The increase in annual air handler runtime fraction required to reach the minimum 20 minutes per hour ventilation runtime for Systems 5a, 6a, and 7a ranged from about 15% in the heating dominated climates to about 20% in the cooling dominated climates.

Table 8. Annual Hours of Central Air Handler Activity and Annual Central Air Handler Runtime Fraction Broken Down into Contributions from Heating, Cooling, and Ventilation

	Total AHU active hrs	Runtime fraction	Percentage of runtime for: heating	cooling	ventilation
Standard house					
Houston	2103	0.24	45%	55%	0%
Phoenix	2067	0.24	23%	77%	0%
Charlotte	2447	0.28	69%	31%	0%
Kansas City	2466	0.28	76%	24%	0%
Seattle	2859	0.33	95%	5%	0%
Minneapolis	2198	0.25	86%	14%	0%
Higher-performance houses					
Avg of Systems 5, 6, 7 with no central fan cycling					
Houston	1647	0.19	52%	48%	0%
Phoenix	2092	0.24	33%	67%	0%
Charlotte	1909	0.22	80%	20%	0%
Kansas City	2719	0.31	83%	17%	0%
Seattle	1854	0.21	98%	2%	0%
Minneapolis	2852	0.33	93%	7%	0%
Avg of Systems 5a, 6a, 7a with 20/60 central fan cycling					
Houston	3626	0.41	23%	23%	54%
Phoenix	3805	0.43	18%	37%	45%
Charlotte	3594	0.41	42%	11%	47%
Kansas City	4060	0.46	55%	12%	34%
Seattle	3265	0.37	55%	2%	44%
Minneapolis	4041	0.46	65%	5%	30%

Summary and Conclusions

The initial cost of residential mechanical ventilation systems can vary by a factor of up to 20. An evolution of applied building science, field experience with production homebuilders, performance monitoring, and computer simulations have shown the central-fan-integrated supply ventilation system with fan and outside air damper cycling to provide highly effective whole-house dilution ventilation and thermal comfort mixing at affordable initial cost and operating cost.

In warm climates with significant cooling, high-performance houses with controlled mechanical ventilation and without use of setback thermostat cost less to operate than standard houses without ventilation and with use of setback thermostat. Reduction of cooling energy use came mainly from low solar heat gain glass and from getting the air distribution system inside conditioned space.

Electrical energy cost to operate the ventilation systems ranged from 2% to 11% of total HVAC system annual operating cost in the warm climates of Houston and Phoenix, and ranged from 1% to 6% in Charlotte, Kansas City, Seattle, and Minneapolis. For a given climate, there was less than $150 per year difference in HVAC operating cost between all of the ventilation systems evaluated. The greatest part of that difference was due to the differences in the annual average air exchange rate.

Simulations showed that additional central fan operation for fan cycling of 20 minutes minimum per hour for ventilation and whole-house mixing amounted to about 15% to 20% of annual hours across the climates. The total annual cost, including the impact on heating and cooling, was $0 to $20 in the climates having a relatively long (dominant) heating season and low electric rates (Minnesota, Seattle, Kansas City). The low cost was also due to fan heat contributing to heating. The annual cost was about $50 in Charlotte, where more of a balance of heating and cooling operation is needed. The annual cost was $75 to $85 in the climates with dominant cooling operation and high electric rates (Phoenix, Houston), where fan heat mostly added to the cooling load. Those results are in general agreement with previous publications based on monitored data or simulations (Rudd 1998, Rudd 1999, EDU 2001, NAHB RC 2001, Henderson et al. 2006).

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