Design/Sizing Methodology and Economic Evaluation of Central-Fan-Integrated Supply Ventilation Systems

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ABSTRACT

Ventilation systems for residential buildings can be generally categorized as supply, exhaust, or balanced systems. Subcategories include: integration into central air distribution ducts, or single- or multi-point air distribution. This effort focused on establishing a design methodology for central-fan-integrated supply ventilation systems using an outside air duct to the return side of a central air distribution fan, with a specialized fan recycling control. A measurement protocol was developed, and air flow measurements were taken for 25' lengths of 5" through 9" diameter flexible ducts, with a 6" wall-cap, at duct pressures of -10 Pa to -120 Pa. Using these measurements, a five-step method was developed as a guide for correctly designing, sizing, and installing the components of the ventilation system. An economic evaluation was made by conducting hourly computer simulations to determine the impact on heating, cooling, and fan energy use for four U.S. climates: cold (Chicago), mixed (Charlotte), hot-dry (Las Vegas), and hot-humid (Orlando). It was found that an effective ventilation system can be achieved using a 5" to 9" diameter insulated duct from outdoors to the return side of a central air distribution fan, with a specialized fan control that automatically cycles the fan if the fan has been inactive for a period of time. The advent of this specialized fan recycling control has made this type of ventilation system viable and more energy efficient.

Introduction

Energy efficient homes with low air leakage rates require mechanical ventilation for acceptable indoor air quality (ASHRAE 1989). A number of residential mechanical ventilation system types exist. These systems can be generally categorized as follows:

- Supply ventilation, with central-, single- or multi-point distribution
- Exhaust ventilation, with single- or multi-point exhaust, and with or without passive inlet vents
- Balanced ventilation, with single- or multi-point supply and exhaust, and with or without heat recovery or energy recovery (heat and moisture)

Recent related publications include (Reardon and Shaw 1997), (Lubliner et al. 1997), (Sherman and Matson 1997), (Holton et al. 1997).

Supply Ventilation

Supply ventilation systems draw outside air from a known location and deliver it to the interior living space. This known location should be selected to maximize the ventilation air quality. The air can be treated before distribution to the living space (heated, cooled, dehumidified, filtered, cleaned). If supply ventilation air is not pre-treated, it should be mixed with recirculated indoor air to mitigate discomfort effects of the outside air. Supply ventilation will tend to pressurize an interior space relative to the outdoors, causing inside air to be forced out through leak sites (cracks, holes, etc.) located randomly in the building envelope. This strategy is advantageous in warm, humid climates to minimize moisture entry into the building structure from outdoors. Care should be taken with building envelope design and workmanship when using supply ventilation in climates with cold winters.

In cold climates, interior humidity control is important to reduce condensation potential. As a first course of action, areas of high moisture generation, such as kitchens and baths, should be exhausted at the source. Controlled ventilation then serves to dilute remaining interior moisture with dryer outdoor air. In some cold climate houses, depending on the building envelope design and the quality of workmanship, an exhaust fan may be advisable to balance supply ventilation air to avoid pressurizing the building. To avoid potential problems with supply ventilation in cold climate buildings: 1) the building envelope must be constructed to avoid air leakage that will transport interior moisture into the building envelope; 2a) interior humidity must be controlled such that surfaces inside wall and unvented ceiling cavities remain above the interior air dewpoint temperature; (2b) surfaces inside wall and unvented ceiling cavities can be elevated through use of exterior insulating sheathing.

Central-fan-integrated supply ventilation system. Ventilation systems that provide ventilation air through a duct that extends from outdoors to the return side of a central air distribution fan have an advantage in that they achieve full distribution of ventilation air using already existing ducts (Reardon and Shaw 1997). However, these systems only supply ventilation air when the fan is operating. During mild outdoor conditions, the central fan may not be activated by the thermostat for long periods of time, creating a problem for adequate distribution of ventilation air. Until the introduction of a specialized fan recycling control, the only options to remedy that problem were, 1) run the central fan continuously; or, 2) operate the fan in parallel with a separate timer that had no relation to the fan's operation due to thermostat demand, causing overlapping or short-cycling operation. Both of these options are inefficient, and will shorten the life of the central fan. In addition, operating the central system fan continuously can also lead to moisture related problems in humid climates. A commercially available and patented control (*EDU* 1997, *Builder* 1998) can be set to periodically distribute ventilation air during stagnant periods when there is no thermostat demand to circulate air for purposes of heating or cooling. The specialized fan control cycles the fan only after a delay time from the last operation of the fan. This is an energy efficiency strategy that utilizes the normal cycling of the fan, as the fan operates to distribute conditioned air in response to calls from the thermostat or humidistat, to also distribute ventilation air at the same time. Only if the fan has been inactive for a period of time will the fan be operated by the fan recycling control to distribute ventilation air and provide mixing. Another feature of the control is that a motorized outside air damper can be opened when the central fan is activated, and closed when either the central fan is deactivated or the fan has operated for a given period of time. This will serve to disconnect ventilation air from the house when the fan is not operating, and to limit ventilation air flow if the fan is operating for long continuous periods.

As a prerequisite for energy efficiency in any forced air system, the entire air distribution system must be substantially airtight, including all ducts, dampers, fittings, and the air handler cabinet itself. If the air distribution system is leaky to unconditioned space, this will defeat the purpose of intentionally sizing an outside air duct to provide a controlled amount of ventilation air. A reference leakage rate for "substantially airtight" is debatable and somewhat subjective, however, it could be defined as leakage to outdoors of less than five percent of the rated fan flow (Lstiburek 1997). A good alternative is to locate the entire air distribution system inside conditioned space.

This type of central-fan-integrated supply ventilation system can also provide enhanced temperature and humidity comfort control in conditioned spaces. Thermostats are typically located in a central area and are expected to serve an entire zone that usually includes closed rooms, and often, more than one floor level. Temperature conditions can vary widely between the thermostat location and extremities of the space the thermostat serves. Likewise, humidistats are usually located in a central area and suffer the same control problems. A practical solution to this could be to intermittently utilize the central air distribution fan to average the overall space conditions by mixing, while possibly at the same time supplying ventilation air. This fan recycling operation can also improve the performance of air cleaning or special filtration systems that locate the cleaning or filtration media at the return side of the central fan. Since, in this case, air cleaning can only be performed when the central air distribution fan is operating.

As an alternative, some have suggested using electronically commutated (ECM) fan motors, having variable speed controls, to operate the fan at low speed all of the time to continuously draw and distribute outdoor air for ventilation. The very low pressures induced at low fan speeds will require large outside air duct sizes which may be impractical. A flow regulator, or other flow control, would also be required to assure that the right amount of ventilation air was being drawn, regardless of fan speed. Such an automatic flow regulator is commercially available, however, this device requires at least 50 Pa pressure differential (0.2 inch water column) to operate. Depending on the fan and duct system, this much pressure may not be available even at higher fan speeds. During the cooling season, in humid climates, this running of the air distribution fan immediately after the cooling/dehumidifying apparatus has been deactivated is counter-productive, in that, moisture on the wet cooling/dehumidifying apparatus is returned to the interior space by the recirculating air. In addition, immediately after a cooling cycle, depending on the percentage of outdoor air and the outdoor humidity level, moisture could condense inside cool supply ducts. A delay should occur after a cooling cycle, wherein the fan does not operate, to allow water on the cooling/dehumidifying apparatus to drip off to the condensate drain and to allow cool supply ducts to warm up.

Single-point supply ventilation systems using a separate fan. Single-point supply ventilation systems usually supply ventilation air to a location in the central area of the house. These systems usually do not achieve whole house distribution of ventilation air. Closed rooms, such as bedrooms, will probably not receive adequate ventilation air unless the central system fan operates to mix air between the central area and the closed rooms.

Alternatively, the outdoor ventilation air can be distributed through the supply ducts of the central system, but this requires that a fan be used that is powerful enough to overcome the static pressure in the supply duct when the central fan is operating, and it requires that care be taken to avoid condensation in the supply ducts in humid climates. Also, if the supply ventilation fan does not operate continuously, a back-draft damper must be installed to eliminate a large supply duct leak that would occur if the central fan operates and the supply ventilation fan does not. This system will usually not achieve whole house distribution of ventilation air when the central fan is not operating, since the supply ventilation air will follow the path of least resistance, likely flowing to the main central return grille and the closest supply register.

Multi-point supply ventilation systems using a separate fan. Multi-point supply ventilation systems usually supply ventilation air to a ceiling location in the central area and to all rooms of the house where a door can be closed and wherein occupants spend significant time. These systems usually do achieve whole house distribution of ventilation air (Reardon and Shaw 1997). These systems require separate ducts for distributing ventilation air only.

Exhaust Ventilation

Exhaust ventilation systems expel conditioned inside air directly to outdoors. Exhaust ventilation will tend to depressurize an interior space relative to the outdoors. This strategy can be advantageous in climates with cold winters, but care should be taken when using exhaust ventilation in warm, humid climates. Exhaust ventilation systems draw outside air from leak sites (cracks, holes, etc.) located randomly in the building envelope. In exhaust ventilation systems, it is not possible to treat the outside air before it enters the living space since it is not known from where it is comes. The "ventilation" air could be coming from pollutant sources and cause indoor air quality problems. For instance, air drawn from cracks in a concrete slab, or a basement, or a crawl space may allow entry of insecticides, radon gas, and fungal or mold spores. In humid cooling climates, if interior negative pressure causes moisture laden outdoor air to enter the building envelope, the moisture will condense on cool surfaces, and if not allowed to dry to the inside, material durability and indoor air quality problems will result.

Single-point exhaust systems. Single-point exhaust ventilation systems usually exhaust ventilation air from a ceiling location in the central area of the house. They may or may not include passive air inlet vents in all room. These systems without air inlet vents in all rooms usually do not achieve whole house distribution of ventilation air, and these systems with passive inlet vents in all rooms are only marginally better (Reardon and Shaw 1997). Closed rooms, such as bedrooms, will probably not receive adequate ventilation air unless the central system fan operates to mix air between the central area and the closed rooms, or unless the closed rooms are made to be very leaky to the central area, such as in manufactured homes in the Northwest U.S.A. (Lubliner et al. 1997).

Multi-point exhaust ventilation systems using a separate fan. Multi-point exhaust ventilation systems usually exhaust air from a ceiling location in the central area and from all rooms of the house where a door can be closed and wherein occupants spend significant time. These systems usually do achieve whole house distribution of ventilation air. These systems require separate ducts for distributing ventilation air only.

Balanced Ventilation

Balanced ventilation systems exhaust conditioned inside air to outdoors and supply outside ventilation air to the inside space. Balanced ventilation, by definition, should not effect the pressure of an interior space relative to the outdoors. Although, in reality the balance may never be perfect due to fluctuations in wind induced pressure and stack pressure. This ventilation strategy can be used effectively in any climate. It is possible to filter or condition the outside air before it enters the living space.

Single-point balanced ventilation systems. Single-point balanced ventilation systems usually exhaust ventilation air from a location in the central area of the house and supply outdoor ventilation air to another location in the central area of the house. These systems usually do not achieve whole house distribution of ventilation air. Closed rooms, such as bedrooms, will probably not receive adequate ventilation air unless the central system fan operates to mix air between the central area and the closed rooms.

Multi-point balanced ventilation systems. Multi-point balanced ventilation systems usually supply and exhaust ventilation air at ceiling location in the central area and to all rooms of the house where a door can be closed and wherein occupants spend significant time. These systems usually do achieve whole house distribution of ventilation air and require separate ducts for distributing ventilation air only.

Alternatively, for both single- and multi-point balanced ventilation systems, the outdoor ventilation air can be distributed through the supply ducts of the central system, but this requires that a fan be used that is powerful enough to overcome the supply duct static pressure when the central fan is operating, and it requires that care be taken to avoid condensation in the supply ducts in humid climates. This system will usually not achieve whole house distribution of ventilation air when the central fan is not operating, since the supply ventilation air will likely flow to the main central return grille and the closest supply register.

Balanced ventilation with heat recovery. Balanced ventilation systems with heat recovery operate the same as the balanced ventilation systems described above with the exception that a heat exchanger transfers heat between the exhaust air stream and the outside ventilation air stream. No moisture is exchanged between the air streams. This means that in cold months, the heating load due to ventilation will be less, and in hot months, the sensible cooling load due to ventilation will be less.

Balanced ventilation with energy recovery. Balanced ventilation systems with energy recovery operate the same as the balanced ventilation systems described above with the exception that heat and moisture is exchanged between the exhaust air stream and the outside ventilation air stream. This means that in cold, dry months, the heating load due to ventilation will be less, and the house interior moisture level will be higher than it otherwise would have been without energy recovery. In hot, humid months, the total cooling load (sensible and latent) due to ventilation will be less. A common misconception is that energy recovery ventilation systems can be used to dehumidify the interior space. This can happen only when outside air is much drier than inside air. When outside air is dry with respect to inside air, the moisture transfer effect in the energy recovery process transfers moisture from the outgoing exhaust air stream to the incoming fresh air stream, limiting the drying potential of the outside air. To put it another way, energy recovery ventilation will keep the interior space more humid than it otherwise would have been without energy recovery when the outside air is dry relative to inside air. Energy recovery will keep the interior space less humid than it otherwise would have been without energy recovery, by reducing the incoming moisture load, when the outside air is wet relative to inside air.

Ventilation Requirements

Two code jurisdictions in the United States require whole house mechanical ventilation for homes. One is the HUD Manufactured Home Construction and Safety Standards (HUD 1994), and the other, the Washington State Ventilation and Indoor Air Quality Code (WSBCC 1993).

Hud manufactured home construction and safety standards. Part 3280.103 (b) *Whole house ventilation* of the HUD Manufactured Home Construction and Safety Standards states:

(1) Natural infiltration and exfiltration shall be considered as providing 0.25 air changes per hour.
(2) The remaining ventilation capacity of 0.10 air change per hour or its hourly average equivalent shall be calculated using 0.035 cubic feet per minute per square foot of interior floor space. This ventilation capacity shall be in addition to any openable window area.

Washington state building code council, washington state ventilation and indoor air quality code (second edition). The Washington State Ventilation and Indoor Air Quality Code (Second Edition), Chapter 51-13 WAC, section 302.2.2 Whole House Ventilation Systems states:

> "Each dwelling unit shall be equipped with a whole house ventilation system which shall be capable of providing at least 0.35 air changes per hour, but not less than fifteen cubic feet per minute per bedroom plus an additional fifteen cubic feet per minute. Whole house ventilation systems shall be designed to limit ventilation to a level no greater than 0.5 air changes per hour under normal operating conditions."

The ventilation system designer can calculate volume and air change rate (between 0.35 and 0.5) to arrive at the required ventilation flow, or, Section 304.1, Table 3-2 shows prescriptive Whole House Ventilation Flow Requirements dependent on number of bedrooms as listed. It is noted that the values listed as minimums in this list are greater than the previously stipulated minimum of 15 cfm per bedroom plus 15 cfm.

	CFM	
Bedrooms	**Minimum**	**Maximum**
2 or less	50	75
3	80	120
4	100	150

In the Washington State Ventilation and Indoor Air Quality Code, Section 304.1, Table 3-5 shows Prescriptive Integrated Forced Air Supply Duct Sizing as listed.

Number of Bedrooms	Minimum Smooth Duct	Minimum Flexible Duct Diameter
2 or less	6"	7"
3	7"	8"
4 or more	8"	9"

Notes: 1. For lengths over 20 feet increase duct diameter 1 inch.
2. For elbows numbering more than 3 increase duct diameter 1 inch.

Some economic analysis has been published by (Miller and Conner 1993), (Lubliner et al. 1997), (Sherman and Matson 1997). However, hourly computer simulations, with real buildings designed to meet national energy standards, and with properly sized heating and cooling equipment has not been published.

Test Protocol For Determining Outside Air Flow Rates

A test protocol was made to establish outside ventilation air flow rates for a central-fan-integrated supply ventilation system having an outside air duct connected to the return side of a central air distribution fan. Figure 1 illustrates the schematic for this system. The test protocol included the measurement of air flow through a series of outside air duct configurations at a range of negative pressures. Outside air duct configurations included: 6" wall jacks (outside air inlets), from 5" to 9" diameter flexible duct sizes, and round to rectangular duct transitions (register box). Tested outside air duct pressures ranged from -10 to -120 Pascal. This range of duct pressures was based on measurement of over 30 central-fan-integrated ventilation systems installed in production homes in Chicago and Las Vegas as part of the U. S. Department of Energy Building America Program. The heating and cooling air distribution systems were fully ducted (no return ducts in framed or panned cavities) and were sealed with mastic. The range of pressures resulted primarily from the location of the connection between the outside air duct and the central return air duct. In some cases, the outside air duct was located at the main return air filter grille assembly (lower pressures), while in other cases, the outside air duct was located in the return air plenum near the air handler unit (higher pressures).

Test Apparatus Description

All outside air ducts with the associated wall cap, balancing damper, register box, and filter were constructed using off-the-shelf components from local HVAC suppliers or builder suppliers. All flexible ducts were uncut 25' lengths. All joints between ducts and fittings were sealed with temporary tape (for permanent field installations, fiberglass mesh and mastic should be used). A 10"x10" register box (12"x12" outside dimensions) was modified with a one inch slot to hold a 12"x12" outside air filter, and was used as a transition between the flexible duct and the return plenum A calibrated fan was used to

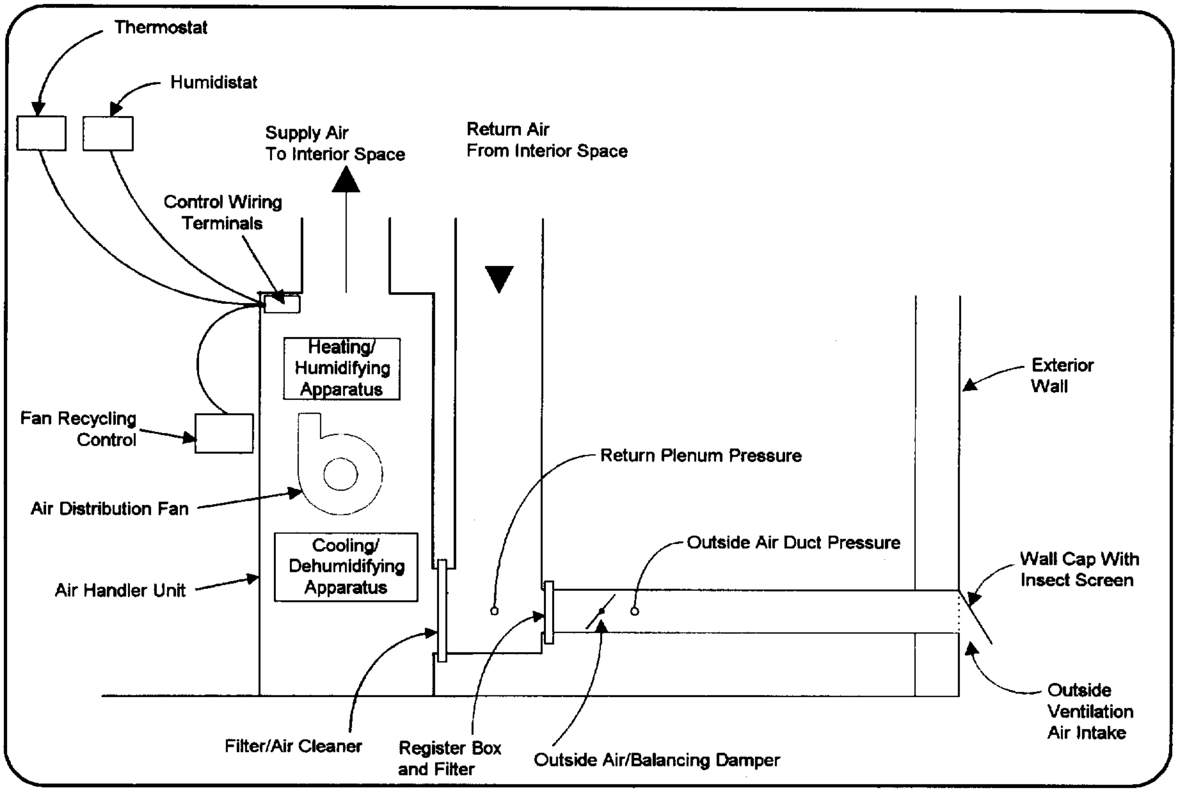

Figure 1. Diagram for one configuration of a central-fan-integrated supply ventilation system using an outside air duct to the return side of a central air distribution fan

create the range of negative pressures and to measure air flow. Digital pressure manometers were used to measure the outside air duct pressure upstream of the balancing damper, and to measure the fan pressures required to calculate the air flow rate by the fan calibration formulas. The tests were conducted inside a laboratory building to eliminate wind effects.

Testing Results

Figure 2 graphically shows the measured relationship between air flow rate and negative pressure in the outside air duct, for the various duct sizes. In all cases, a 6" wall jack and the appropriate reducer/enlarger, if required, was used. This is the primary size available and leaves a minimal visual impact on the building exterior. As shown in Figure 2, for outside air duct pressures between -10 Pa and -120 Pa, outside air flow rates ranged from a low of 40 ft^3/min for 5" diameter duct to a high of 290 ft^3/min for 9" diameter duct. Since duct size and pressure determine air flow, it is important to know those two parameters when designing and installing central-fan-integrated supply ventilation systems. If the return duct pressure is unknown at the time of design, measurement of outside air duct pressure and adjustment of a balancing damper after installation is very important. To illustrate that point, to achieve 120 cfm of outside air when the central fan was operating, one could use any of the following duct size and duct pressure combinations: a 5" duct at -73 Pa, a 6" duct at -55 Pa, a 7" duct at -27 Pa, an 8" duct at -21 Pa, or a 9" duct at -18 Pa.

When checking the measured air flow values of Figure 2 against the prescriptive duct sizing requirements of the Washington State Ventilation And Indoor Air Quality Code (Second Edition), given

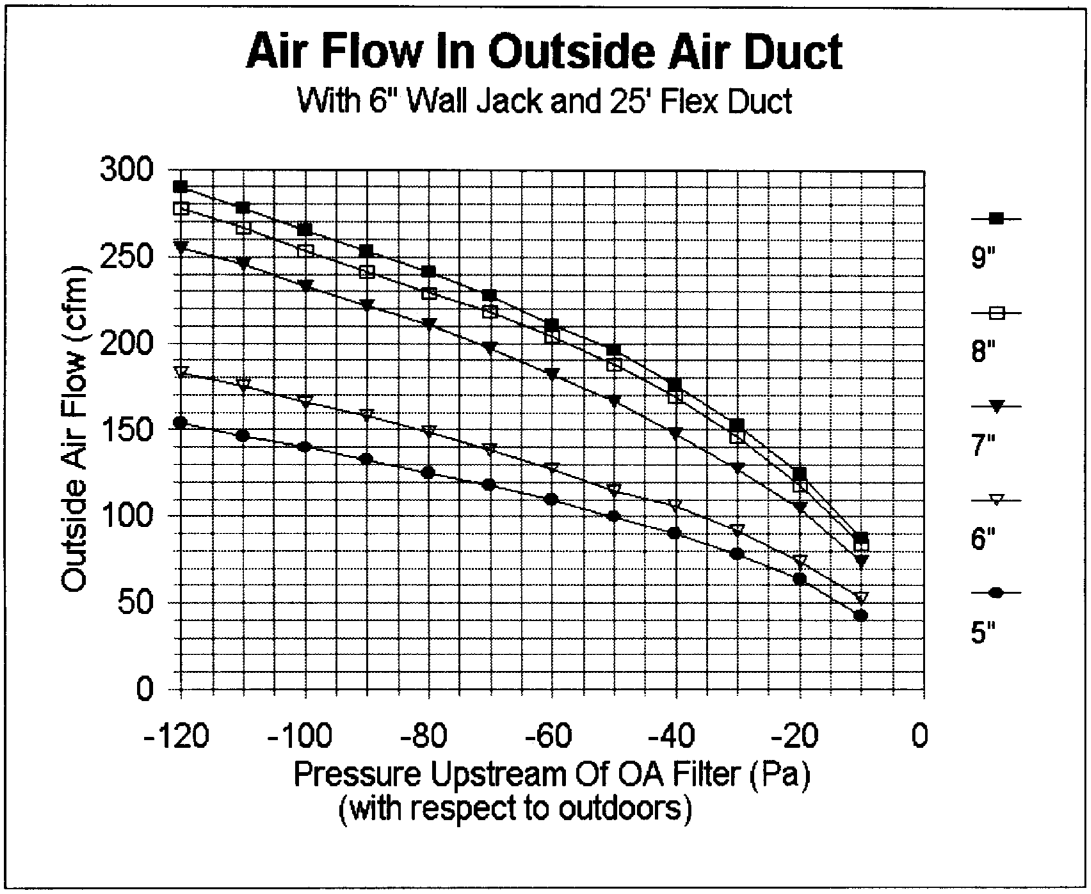

Figure 2. Air flow rate in outside air duct versus outside air duct pressure, for flexible duct sizes between 5 inches and 9 inches in diameter, using a 6" wall jack in all cases

previously in this report in section 2.6.2, it appears that the authors of that code assumed low outside air duct pressures–around -10 to -15 Pa. Even for furnace-only systems that have lower air flow than those with cooling systems, outside air duct pressures with reasonably good duct design and duct sealing can be at least -30 Pa. If the central air distribution system return ducts are sized adequately, and if they are substantially airtight, the Washington State prescriptive outside air duct sizing may tend to cause over-ventilation. (According to one author of the Washington State code, measured available negative pressures in the return air plenum were rarely more than 25 Pa because of duct leakage (1998). The prescriptive sizing table was based on negative pressures between -12 and -25 Pa (-0.05 to -0.1 inch water column).

Discussion

As a prerequisite for energy efficiency in any forced air system, the entire air distribution system must be substantially airtight, including all ducts, dampers, fittings, and the air handler cabinet itself. If the air distribution system is leaky to unconditioned space, this will defeat the purpose of intentionally sizing an outside air duct to provide a controlled amount of ventilation air. A good alternative is to locate the entire air distribution system inside conditioned space.

A five-step procedure was created for designing, sizing, and correctly installing a central-fan-integrated supply ventilation system. The procedure uses Tables 1 and 2, Eq. 1, and Figure 2. Interpolation between listed values on Tables and Figures is acceptable.

Step 1: Using Table 1, select the continuous outside air flow requirement for your house design as a function of the number of bedrooms, plus one, and the air flow per bedroom. (The method of Table 1 sizes the ventilation air flow rate as required by people occupying the space, rather than based on building volume. It assumes that there will be two people for the first bedroom, and one person for each additional bedroom. Other sizing methods can be used. ASHRAE Standard 62-1989 recommends the greater of 15 cfm per person or 0.35 air changes per hour. Table 1 gives a range of cfm/bedroom values to make it easy to choose a different value based on personal choice, for example, if people have pets or want to smoke inside, they may want to choose 20 cfm/bedroom. To hold energy costs and equipment sizing lower, others may want to choose 10 cfm/bedroom.)

Table 1. Outside air flow requirements based on the number of bedrooms plus one, at various flow rates per bedroom

Number of Bedrooms In House	Continuous Outside Air Flow Based On Number Of Bedrooms Plus 1 At Listed Flow Per Bedroom (cfm/bdrm)		
	10	15	20
1	20	30	40
2	30	45	60
3	40	60	80
4	50	75	100
5	60	90	120

Step 2: Using Table 2, select the fan duty cycle percentage based on the specialized fan recycling control FAN OFF time and FAN ON time you want to use. (Table 2 is based on a commercially available product.)

Step 3: Using Eq. 1, calculate the intermittent outside air flow that will be equivalent to the continuous outside air flow selected in **Step 1**, for the fan duty cycle you chose in **Step 2**.

$$Q_{int} = [(100\ Q_{cont}) - (I\ V_{space}\ (100-C))/60]/C \qquad \textbf{Eq. 1}$$

where: Q_{int} = intermittent outside air flow rate in ft^3/min
Q_{cont} = continuous outside air flow rate desired, from Table 1, in ft^3/min
I = estimate of natural air change when central fan is not operating, 1/h (ach)
V_{space} = volume of conditioned space, ft^3
C = fan duty cycle percentage from Table 2

The estimate of natural air change rate when the central fan is not operating is difficult to predict since it changes with environmental conditions. To be conservative, it should be set to a low value (not greater than 0.15). If it is set to zero, the space may be over-ventilated resulting in higher energy costs.

Step 4: Using Figure 2, select the outside air duct size that will give you the required intermittent outside air flow, calculated in **Step 3**, based on the expected negative pressure in the outside air duct upstream of the outside air balancing damper. (This "expected" outside air duct pressure can be estimated to be between 5% and 20% less than the return air plenum pressure from the lower air flow to higher air flow shown on Figure 2. If the return air plenum pressure is unknown, select a 6" diameter duct, but be sure to adjust the balancing damper upon installation.)

Step 5: After installation of the entire air distribution system, including outside air duct with balancing damper, measure the pressure in the outside air duct upstream of the balancing damper, with respect to outside. Adjust the balancing damper until the combination of the duct pressure and the duct size match with the required outside air flow in Figure 2. Then lock the damper in place and permanently mark its proper position.

Table 2. Fan duty cycle percent, based on fan recycling control settings for FAN OFF time and FAN ON time.

		Specialized Fan Recycling Control FAN ON Time (min)															
		6	**8**	**10**	**12**	**14**	**16**	**18**	**20**	**25**	**30**	**35**	**40**	**45**	**50**	**55**	**60**
F	**15**	29	35	40	44	48	52	55	57	63	67	70	73	75	77	79	80
A	**20**	23	29	33	38	41	44	47	50	56	60	64	67	69	71	73	75
N	**25**	19	24	29	32	36	39	42	44	50	55	58	62	64	67	69	71
	30	17	21	25	29	32	35	38	40	45	50	54	57	60	63	65	67
O	**35**	15	19	22	26	29	31	34	36	42	46	50	53	56	59	61	63
F	**40**	13	17	20	23	26	29	31	33	38	43	47	50	53	56	58	60
F	**45**	12	15	18	21	24	26	29	31	36	40	44	47	50	53	55	57
	50	11	14	17	19	22	24	26	29	33	38	41	44	47	50	52	55
T	**55**	10	13	15	18	20	23	25	27	31	35	39	42	45	48	50	52
I	**60**	9	12	14	17	19	21	23	25	29	33	37	40	43	45	48	50
M	**80**	7	9	11	13	15	17	18	20	24	27	30	33	36	38	41	43
E	**100**	6	7	9	11	12	14	15	17	20	23	26	29	31	33	35	38
	120	5	6	8	9	10	12	13	14	17	20	23	25	27	29	31	33
	140	4	5	7	8	9	10	11	13	15	18	20	22	24	26	28	30
	160	4	5	6	7	8	9	10	11	14	16	18	20	22	24	26	27
	180	3	4	5	6	7	8	9	10	12	14	16	18	20	22	23	25

An example follows, stepping through the procedure to size a central-fan-integrated supply ventilation system that uses an outside air duct and a specialized fan recycling control:

<u>Example Givens</u>:

Number of bedrooms = 3

Continuous ventilation air flow per bedroom = 10 cfm

Expected outside air duct pressure relative to outside = -35 Pa

Fan recycling control settings: FAN OFF = 20 min; FAN ON = 10 min

Conditioned space volume = 12000 ft^3

Estimated natural air change rate while central fan is not running = .15 1/h (ach)

<u>Design/Installation Solution</u>:

Step 1: From Table 1, 3 bedrooms at 10 cfm per bedroom (plus 1) gives 40 cfm.

Step 2: From Table 2, a FAN OFF setting of 20 min, and a FAN ON setting of 10 min gives a fan duty cycle percentage of 33%. This can also be calculated by dividing the ON time by the sum of ON and OFF time, converted to percent (10/(10+20)*100).

Step 3: From **Eq. 1**, an intermittent flow of 60 cfm of outside air is calculated for ventilation [(40)(100) - (.15)(12000)(100-33)/60]/33

Step 4: From Figure 2, a 5" outside air duct will provide 60 cfm at an outside air duct pressure of -20 Pa (measured just upstream of the balancing damper with respect to outdoors)

Step 5: After installation of the entire air distribution system, the pressure measured in the outside air duct upstream of the balancing damper was -40 Pa. Close the damper until the pressure is -20 Pa. Lock the damper in place by tightening the wing nut, or screw the damper handle to the duct wall, and mark the damper position as the proper setting with a permanent marker.

Furnace manufacturers often require that the furnace heat exchanger not be exposed to air below a minimum air temperature to protect the heat exchanger from excessive thermal expansion and contraction. For cold outdoor conditions, for a central-fan-integrated supply ventilation system with a 7% outside air fraction, if the outdoor temperature was -25 F and the indoor setpoint temperature was a low 66 F, the mixed air temperature at the furnace heat exchanger would be 60 F. This mixed air temperature would not negatively impact a furnace heat exchanger.

During the heating season, under part load conditions, the specialized recycling control may operate the central fan between heating cycles to distribute ventilation air. Depending on the location of supply registers, and the supply air velocity, and the sensitivity of the occupants to essentially room temperature air being circulated, comfort may be a concern (Lubliner 1997). However, this type of central-fan-integrated supply ventilation system with fan recycling control has been in place during two heating seasons in production housing in the Chicago area and no occupant complaints have been encountered. During the cooling season, if the specialized fan recycling control operates the fan between cooling cycles, it is unlikely that supplying air slightly above room temperature will negatively affect occupant comfort, in fact, it is likely that due to the mixing of air in the entire conditioned space, the thermostat will have better feedback resulting in improved temperature control and comfort.

Economic Evaluation

An economic evaluation was made by conducting hourly computer simulations to determine the impact of central-fan-integrated supply ventilation on heating, cooling, and fan energy use for four U.S. climates: cold (Chicago), mixed (Charlotte), hot-dry (Las Vegas), and hot-humid (Orlando).

Simulation Model Setup

The hourly simulation program used was DOE2.1E. A special function was written to calculate the load placed on the heating and cooling equipment due to a specified amount of ventilation air coming into the central return whenever the central fan was operating. This function also accounted for the additional fan power and associated sensible load. The duty cycle of the central fan was specified (33%), such that, if the heating and cooling system did not operate enough to meet the specified duty cycle, the fan alone would operate to meet the duty cycle requirement. By running the model both ways – with and without the duty cycle specified – a comparison of fan operational hours and the associated impact on heating, cooling, and fan energy use could be made. All fan power was calculated at 0.35 W/cfm (based on a combined fan and motor efficiency of 0.23 and external static pressure of 0.68 inch water column) and 400 cfm per ton of cooling.

The simulations were labeled as Base 1, Base 2, Cen Fan1 and Cen Fan2. The Base 1 case had no mechanical ventilation system. Base 2 had 40 cfm of continuous supply ventilation provided by a separate fan. Two central-fan-integrated cases were simulated, each one had 60 cfm of intermittent supply ventilation for one third of the time (33% duty cycle), while the difference between them was where the air distribution ducts were located. The location of air distribution ducts was the same for Base 1, Base 2, and Cen Fan1. For these three cases, air distribution ducts were located in the attic for Orlando and Las Vegas, and located in the crawl space for Charlotte. No Cen Fan1 case was modeled for Chicago since ducts are commonly located in conditioned space in that area (i.e. basement and interior walls and floors). For Cen Fan2 cases, all air distribution ducts were located inside the conditioned space. No unintentional duct leakage was modeled for any of the cases, only conduction gains and losses. Because duct leakage would likely be found in most conventional residential forced-air systems, savings for the Cen Fan2 case would likely be greater than what is reported here. As a reminder, when using central-fan-integrated supply ventilation, it is important that the air distribution ducts be sealed with fab-glass and mastic or other equal duct closure system.

A model house, previously used for numerous computer simulation studies, was chosen. This 1500 ft^2 house was single-story, with three bedrooms, two bathrooms, and two-car garage. Glazing area was fifteen percent of floor area. The building envelope characteristics shown in Table 3, for each city, were taken from the prescriptive requirements listed in the ASHRAE Standard 90.2-1993 (ASHRAE 1993).

For each of the four cities, a determination of the heating and cooling system size was made using a commercially available software that calculates the system capacity requirement based on a room-by-room Manual J approach (Elite 1994). For some of the cases, the calculated cooling system size was increased by 6000 Btu/h (.5 ton) to avoid peak-day equipment runtime fractions much greater than 1.0 (duty cycle > 100%). The system sizing used in the simulations are shown in Table 3.

Table 3. Building Envelope characteristics and size of heating and cooling equipment used in simulations

	U-value (Btu/h ft² °F)			
Building Component	**Chicago**	**Charlotte**	**Las Vegas**	**Orlando**
Above grade exterior walls	0.063	0.063	0.063	0.063
Above grade adjacent walls	0.095	0.095	0.095	0.095
Basement walls	C=.080	---	---	---
Ceilings	0.036	0.036	0.036	0.036
Floors	---	0.072	R5, 2ft	R4, 2ft
Coors	0.19	0.19	0.19	0.19
Windows	0.36	0.87	0.87	0.87
Window SC	0.7	0.7	0.5	0.5
Heating and cooling system sizes used in simulations				
Heating (Btu/h)	37000	37000	37000	37000
Cooling (ton)				
Base 1	2.0	2.5	3.0	2.5
Base 2	2.0	2.5	3.0	2.5
Cen Fan1	2.5	2.5	3.0	2.5
Cen Fan2	2.0	2.0	2.5	2.0

Air infiltration was handled differently for each case. For the Base 1 case, with no whole-house mechanical ventilation, infiltration was calculated by the DOE model using the Sherman-Grimsrud (S-G) method and a specific leak area of 0.0005 (ELA=108 in^2), corresponding to conventional construction where no extra effort is made to seal the house tightly. The Base 2 case had 40 cfm of continuous supply ventilation with no natural air infiltration. The rationale here was that, for a house constructed tightly with the intention of installing mechanical whole-house ventilation, the continuous supply ventilation would suppress natural pressure forces, minimizing natural infiltration. For the central-fan-integrated cases, it was reasoned that during the time the central fan was operating, and supplying 60 cfm of outside air, that natural pressure forces would be suppressed, minimizing natural infiltration. To account for natural infiltration during periods when the central fan was not operating, natural infiltration was simulated for all hours using the S-G method and a specific leakage area of 0.0002 (ELA=44 in^2), corresponding to construction where extra effort was made to seal the house tightly. With the minimum duty cycle for the fan set at 33%, the 60 cfm of outside air for 33% of the time, combined with the natural air infiltration, provided as much or more outside air as the 40 cfm continuous supply ventilation case.

No interior humidity setpoint was specified or modeled. Interior humidity was controlled only to the extent of moisture removal by the cooling system when it was operating. During periods when outdoor temperature is mild but outdoor humidity remains high, the introduction of ventilation air could cause indoor comfort conditions to be exceeded. In humid climates, cooling equipment with specific humidity controls or separate dehumidification capability may be required to maintain standard comfort conditions. It is planned to make this topic the focus of further study.

Simulation Results

A summary of the simulation results is shown in Table 4. Published electric and gas utility rates for each city were used in the analysis (EIA 1996). Gas rates published in dollars per thousand cubic feet were converted to dollars per kilowatt-hour for unit consistency with electric rates.

Table 4. Summary of Simulation Results for Central-Fan-Integrated Supply Ventilation System

	Chicago		Charlotte		Las Vegas		Orlando	
	kW-h	$ @.11	kW-h	$ @.07	kW-h	$ @.07	kW-h	$ @.08
Fan Energy								
Base 1: no mechanical ventilation	720	79	673	47	779	55	648	52
Base 2: 40 cfm continuous supply ventilation	750	82	755	53	845	59	762	61
Cen Fan1: 60 cfm intermittent supply vent. @ 33%			1284	90	1486	104	1216	97
Cen Fan2: 60 cfm int. supply vent. @ 33%, ducts in cond. space	1004	110	970	68	1252	88	1025	82
Difference, CF1-B1			611	43	707	49	568	45
Difference, CF2-B1	284	31	297	21	473	33	377	30

	Chicago kW-h	$ @.11	Charlotte kW-h	$ @.07	Las Vegas kW-h	$ @.07	Orlando kW-h	$ @.08
Cooling Energy								
Base 1: no mechanical ventilation	2162	238	2740	192	4727	331	5441	435
Base 2: 40 cfm continuous supply ventilation	2218	244	2786	195	4628	324	5338	427
Cen Fan1: 60 cfm intermittent supply vent. @ 33%			2813	197	4728	331	5321	426
Cen Fan2: 60 cfm int. supply vent. @ 33%, ducts in cond. space	2205	243	2665	187	4177	292	4684	375
Difference, CF1-B1			73	5	1	0	-119	-10
Difference, CF2-B1	44	5	-75	-5	-550	-39	-757	-61

Heating Energy	kW-h	$ @.02	kW-h	$ @.03	kW-h	$ @.02		
Base 1: no mechanical ventilation	15357	307	9300	279	6949	139	373	30
Base 2: 40 cfm continuous supply ventilation	11298	226	7533	226	5508	110	240	19
Cen Fan1: 60 cfm intermittent supply vent. @ 33%	14395	288	7807	234	5594	112	258	21
Cen Fan2: 60 cfm int. supply vent. @ 33%, ducts in cond. space	12467	249	7207	216	5090	102	224	18
Difference, CF1-B1			-1494	-45	-1355	-27	-115	-9
Difference, CF2-B1	-2890	-58	-2094	-63	-1859	-37	-148	-12

Net Annual Cost	$	$	$	$
Base 1: no mechanical ventilation	624	518	524	517
Base 2: 40 cfm continuous supply ventilation	552	474	493	507
Cen Fan1: 60 cfm intermittent supply vent. @ 33%	702	521	547	544
Cen Fan2: 60 cfm int. supply vent. @ 33%, ducts in cond. space	602	471	482	475
Difference, CF1-B1		3	22	27
Difference, CF2-B1	-22	-47	-43	-42

Central Fan Operational Hours	hours	hours	hours	hours
Base 1: no mechanical ventilation	2571	1922	1855	1851
Base 2: 40 cfm continuous supply ventilation	2239	1808	1720	1826
Cen Fan1: 60 cfm intermittent supply vent. @ 33%		3668	3538	3474
Cen Fan2: 60 cfm int. supply vent. @ 33%, ducts in cond. space	3587	3463	3576	3662
Difference, CF1-B1		1746	1683	1623
Difference, CF2-B1	1015	1541	1721	1811

Central Fan Average Hourly Duty Cycle	avg duty cycle	avg duty cycle	avg duty cycle	avg duty cycle
Base 1: no mechanical ventilation	29%	22%	21%	21%
Base 2: 40 cfm continuous supply ventilation	26%	21%	20%	21%
Cen Fan1: 60 cfm intermittent supply vent. @ 33%		42%	40%	40%
Cen Fan2: 60 cfm int. supply vent. @ 33%, ducts in cond. space	41%	40%	41%	42%

Central Fan Maximum Hourly Duty Cycle	max duty cycle	max duty cycle	max duty cycle	max duty cycle
Base 1: no mechanical ventilation	101%	104%	106%	89%
Base 2: 40 cfm continuous supply ventilation	101%	104%	92%	87%
Cen Fan1: 60 cfm intermittent supply vent. @ 33%		104%	96%	87%
Cen Fan2: 60 cfm int. supply vent. @ 33%, ducts in cond. space	104%	99%	95%	94%

Net annual heating, cooling and central fan operation costs ranged from $3 to $27 more for the central-fan-integrated system when comparing the Cen Fan1 case (ducts in unconditioned space) to the Base 1 case (no ventilation system). Although there is an energy use cost for the Cen Fan1 case over the

Base 1 case, the argument against relying on random natural infiltration for acceptable ventilation is well known. Over-ventilation in winter, and under-ventilation in summer can be problematic. In addition, the Base 1 case does not include the important advantage of periodic mixing of house air during mild outdoor conditions, to re-average temperature and humidity conditions, to improve occupant comfort, especially in closed rooms.

Net annual heating, cooling and central fan operation costs ranged from \$22 to \$47 less for the central-fan-integrated system when comparing the Cen Fan2 case (ducts in conditioned space) to the Base 1 case. This highlights the benefit of locating the air distribution ducts inside conditioned space, especially when using a central-fan-integrated ventilation system.

Net annual heating, cooling, and central fan operation costs ranged from \$36 to \$54 more for the central-fan-integrated system when comparing the Cen Fan1 case to the Base 2 case (40 cfm continuous supply), and ranged from \$50 more to \$33 less for the central-fan-integrated system when comparing the Cen Fan2 case to the Base 2 case. The estimated added cost of the Base 2, separately-ducted, and fully-distributed supply ventilation system is approximately \$500 installed. Thus, in either case, at best, it would take over nine years to get a simple payback for the separately ducted supply ventilation system. In addition, the Base 2 case does not include the important advantage of periodic mixing of house air during mild outdoor conditions, to re-average temperature and humidity conditions, to improve occupant comfort, especially in closed rooms.

Conclusion and Recommendations

An effective central-fan-integrated supply ventilation system can be achieved using a 5" to 9" diameter insulated duct from outdoors to the return side of a central air distribution fan, with a specialized fan recycling control. As a prerequisite for energy efficiency in any forced air system, the entire air distribution system must be substantially airtight. For outside air duct pressures between -10 and -80 Pa, outside air flows between 40 and 240 ft^3/min can be achieved with 5" to 9" flexible ducts and a 6" wall cap. Outside air should be filtered before it enters the air handling unit, and a balancing damper is advisable to give additional field control of the delivered outside air volume. The advent of a specialized fan recycling control has made this type of ventilation system viable and more energy efficient. The control should be installed to ensure that fresh air will be periodically distributed throughout the house if the central fan has been inactive for a period of time, such as 20 minutes. This fan recycling may also improve occupant comfort by smoothing out temperature, humidity, and stale conditions between rooms and the central area.

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