MN-00-10-1

Whole-House Ventilation Strategies to Meet Proposed Standard 62.2: Energy Cost Considerations

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ABSTRACT

ASHRAE Standard 62.2P is being proposed to address residential ventilation issues. As housing, especially new housing, gets more airtight and better insulated, it has become clear that many homes are underventilated. The standard contains requirements that provide minimum ventilation rates and source control measures necessary for acceptable indoor air quality. This paper uses previously reported analytical techniques to compare the energy costs of various ventilation strategies for a wide variety of climates and housing types. For new construction, we conclude that mechanical ventilation is needed. In new houses with gas heating, the cheapest whole-house system is a central exhaust fan. The marginal energy costs to provide such ventilation are on the order of 50¢ per day. However, other systems can be more appropriate when depressurization, filtration, moisture, and more expensive heating issues are considered. For most of the existing housing stock, we conclude that infiltration provides adequate ventilation.

INTRODUCTION

When selecting a whole-house ventilation strategy, one must consider the associated energy costs. These costs are important because ventilation-related energy consumption may account for about a third to a half of the space conditioning energy load in houses (Liddament 1996). We estimate this fraction represents about 1.7 to 2.5 EJ in the U.S. each year, with associated operating costs of $14 to $22 billion (EIA 1999a). This energy is about 15% to 22% of the estimated 11.4 EJ consumed each year by the U.S. residential building sector (EIA 1999b). A good whole-house ventilation strategy will minimize this energy consumption and associated costs while still providing effective ventilation.

Assessing energy consumption and costs associated with ventilation for a particular house appears to be a complex issue that depends on climate, house and equipment characteristics, occupant loading and behavior, and energy prices. Without guidance, it is difficult to determine which ventilation strategy to use.

ASHRAE Standards 62 (1999a) and 62.2P (1999b) both offer guidance on minimum ventilation rates for houses. Standard 62.2P also offers three cursory examples on related energy issues in its Appendix E. Beyond these examples, there is little guidance available to assist system selection and none in the form of measured data that correlate ventilation, energy consumption, and cost. Two recent studies (Sherman and Matson 1997; Matson and Feustel 1998) have used simplified modeling in lieu of these data to examine the energy and cost impacts of complying with the whole-house ventilation requirements in ASHRAE Standard 62 (1989). However, those studies are now outdated because Standard 62.2P contains more elaborate requirements, such as mandatory mechanical ventilation and rates that depend on conditioned floor area and occupancy.

The primary goal of this paper is to estimate and compare the energy cost impacts of some standard approaches for some representative cases that meet the requirements of ASHRAE Standard 62.2P, using the previously reported techniques. A secondary goal is to examine the role that infiltration could play in providing adequate ventilation.

This study does not address issues such as outdoor air quality, specific indoor contaminant concentrations, spatial ventilation effectiveness, time-varying occupancy and pollut-

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ant sources, combustion air requirements, house pressurization and depressurization limits, equipment-generated noise, and thermal comfort. These issues are better dealt with on a house-by-house basis using a licensed design professional.

BACKGROUND

The Need for a Residential Ventilation Standard

Apart from energy issues, ventilation is important because it affects residential air quality, which in turn affects health, comfort, and building serviceability. Historically, a mixture of infiltration from leaky envelopes and occupant-controlled openings (i.e., windows) ventilated our homes. Until the energy crisis of the 1970s, many people thought this ventilation was sufficient without regulatory intervention, and there was an easy feeling about the residential indoor environment.

Today, the Environmental Protection Agency lists poor indoor air quality as the fourth largest environmental threat to our country (ALAM 1999). This decline of the indoor environment is due to several qualitative changes in our homes. One change is that new houses are more energy efficient due to improved thermal control indoors. However, this has led to higher moisture levels, which could cause increases in diseases such as asthma. Asthma is of concern because it is the leading serious chronic illness of children in the U.S. Increased indoor moisture levels are also of concern because moisture-related construction defects and damage are on the increase in new houses. A second change is that appliances, home offices, and manufactured products have increased the kind and amount of indoor emissions. According to the American Lung Association, these elements within our homes are increasingly being recognized as threats to our respiratory health. A third change is that urbanization has increased housing density and reduced building separation. As a result, people today expect their house to isolate them from its surroundings, and they do not open windows as much. Due to these changes, proper ventilation is even more necessary now, and mechanical systems are often installed in attempts to meet this need. In addition, a desire to define levels of acceptability and performance has replaced the past easy feelings about the indoor environment.

Minimum requirements for residential ventilation can improve many of these indoor air quality problems. Several public and private institutions have interests in indoor air quality but little experience in developing ventilation requirements. In contrast, ASHRAE, as the professional society with ventilation as part of its mission for over 100 years, already has experience developing ventilation requirements focused on commercial and institutional buildings. Therefore, it was the logical place to develop a new consensus document focused on residential ventilation.

Current Residential Ventilation Requirements: Standard 62-1999

For residential issues, ASHRAE Standard 62-1999 is substantively identical to its 1989 precursor. Table 2.3 in Standard 62-1999 specifies that the whole-house ventilation rates should be the higher value of 0.35 ach and 15 cfm per person. These rates are determined using the volume of the conditioned living spaces and on the occupancy of those spaces. Occupancy is estimated using occupant loading assumptions related to the number of bedrooms or is based on a higher loading if it is known.

Standard 62-1999 allows combined infiltration and natural ventilation (e.g., operable windows) to meet its whole-house requirements. According to Section 5.1 of this standard, whole-house mechanical ventilation is not required except when the combined infiltration and natural ventilation are insufficient. In this standard, there are also requirements for local ventilation of kitchen, bath, and toilet areas. Such ventilation must be provided either by the installation of minimum amounts of mechanical ventilation or by operable windows. However, this standard offers no real guidance to determine when whole-house or local mechanical ventilation is necessary. Therefore, its residential requirements can be interpreted as being quite onerous or meaning virtually nothing.

Proposed Residential Ventilation Requirements: Standard 62.2P

The version of ASHRAE Standard 62.2P that will be used as the basis of this paper is the one that was completed in September and October 1999. It has some significant differences from the version described earlier by Sherman (1999).

Overview. Standard 62.2P is an attempt to address concerns about deteriorating indoor air quality in low-rise dwellings and to set minimum code-intended requirements that will allow energy efficiency measures to be evaluated (Sherman 1999). The main body of this standard has minimum requirements for whole-house ventilation rates, as well as for the control, operation, and distribution of such ventilation. The standard also has requirements for minimum local ventilation rates and for their control and operation.

Other requirements in the main body of Standard 62.2P specify direct outdoor air intake, air inlet placement, minimum sizes for operable natural ventilation openings, air cleaning and filtration performance, isolation of the dwelling unit from other spaces such as garages, and the need to evaluate combustion appliance venting. The standard requires that air moving equipment adhere to manufacturers' installation specifications and that it meet maximum sound ratings. It also specifies minimum duct diameters and maximum duct lengths that can be used without system performance tests and specifies acceptable configurations for multi-branch exhaust fan and duct systems to prevent cross-contamination. In addition to the requirements in the main body of the standard, its appendices provide user guidance on issues such as operation and

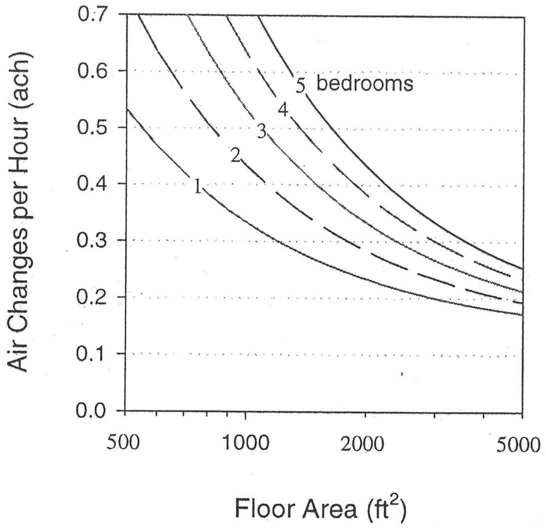

Figure 1 *Minimum whole-house total ventilation rates required by ASHRAE Standard 62.2P.*

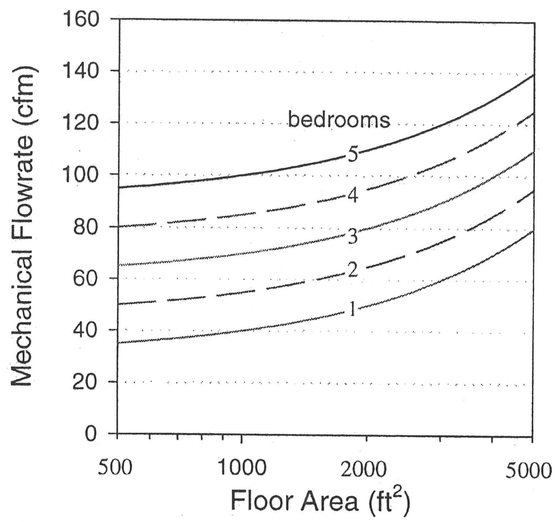

Figure 2 *Minimum whole-house mechanical ventilation rates required by ASHRAE Standard 62.2P.*

maintenance documentation; air filter selection; pollution sources, exposure, and control; and HVAC system design issues related to ventilation.

To meet the whole-house and local requirements, the standard requires that mechanical ventilation be used. It permits alternatives to mechanical ventilation (e.g., trickle-ventilators, passive stacks, and operable windows) but only when approved by a licensed design professional. The following describes in more detail the requirements for whole-house ventilation rates and mechanical ventilation systems.

Whole-House Ventilation Rate Requirements. The minimum whole-house total ventilation rate ($Q_{tot,62.2}$) that is required each hour by Standard 62.2P is 15 cfm per person plus 2 cfm per 100 ft^2 of conditioned floor area. These flows are combined infiltration and supplemental mechanical ventilation. Occupancy is determined in the same manner as in Standard 62-1999.

Figure 1 shows how this total ventilation rate depends on occupancy and house size. For larger houses, the Standard 62.2P total rate is up to 0.15 ach lower than the Standard 62-1999 value, but it is up to 0.15 ach greater for small houses.

Infiltration. Standard 62.2P assumes infiltration is 1 cfm per 100 ft^2 of conditioned floor area. This infiltration credit (Q_{credit}) is based on weather for a presumed critical week and on a very airtight building envelope (normalized leakage of 0.125). The critical week occurs when the weather is extreme enough that occupants no longer open their windows for the remainder of the season, which may vary from climate to climate. Although actual infiltration is dependent on climate and normalized leakage, the magnitude of the credit remains independent of these parameters.

In fact, infiltration may be quite different from this assumption, especially for the housing stock as a whole. The standard takes the conservative position that only a small, fixed infiltration credit is appropriate to avoid "gaming" or providing the counterproductive incentive to poke holes in a tight building envelope. Unfortunately, this approach has the negative impact that a whole-house mechanical system must be installed in older leaky houses, which results in extra costs and energy consumption.

We believe that these extra costs can be avoided by considering actual infiltration. Therefore, all our analyses include it. In particular, a "no-fan" case is included in which ventilation is provided by actual infiltration and intermittent local mechanical ventilation but not by whole-house mechanical ventilation.

Whole-House Mechanical Ventilation Rate. Although our energy and cost analyses consider actual infiltration, the sizing of the whole-house mechanical ventilation systems is based on the presumed infiltration credit in Standard 62.2P. The minimum whole-house *mechanical* ventilation rate (Q_{fan}) that is required each hour by this standard is simply the total requirement ($Q_{tot,62.2}$) minus the infiltration credit (Q_{credit}). Therefore, it is 15 cfm per person plus 1 cfm per 100 ft^2 of conditioned floor area.

Figure 2 shows how this mechanical ventilation rate varies with occupancy and house size. Typically, this "fan size" is expected to be 60 to 100 cfm (Sherman 1999).

Whole-House Mechanical Ventilation System Requirements. Standard 62.2P allows mechanical ventilation systems to operate continuously or intermittently to provide whole-house ventilation. Such systems must have one or more supply or exhaust fans and include associated ducts and override controls. To provide the same effective ventilation rate in each hour, systems that operate intermittently must supply more air than continuously operating systems. The standard requires that such systems operate automatically and for at least one hour out of every twelve. The concept of effective ventilation, which represents the proper temporal ventilation averaging process, is described by Sherman and Wilson (1986) and by Yuill (1986, 1991).

Supply and exhaust flows do not have to be equal (balanced). However, Section 4.5 of Standard 62.2P prohibits unbalanced systems that depressurize the house to any extent in humid climates if the indoor air can be mechanically cooled. Similarly, it prohibits unbalanced systems that pressurize the house to any extent in cold climates. These prohibitions are waived if the building envelope incorporates a moisture-resistant design.

MODELING APPROACH

To determine annual energy consumption and annualized costs associated with providing the minimum whole-house ventilation required by Standard 62.2P, we examined 3168 combinations of three house floor areas, two house heights, two occupancies, three leakage areas, and four ventilation strategies in 22 climates representative of U.S. detached single-family housing. Subsequent sections of this paper describe these parameters in more detail.

For each combination, we used a modified version of the RESVENT computer program developed by Sherman and Matson (1993, 1997) to calculate the hour-by-hour infiltration rates and actual and effective ventilation rates over an entire year. As in our earlier work (Sherman and Matson 1997), the effective ventilation approach embodied in ASHRAE Standard 136 (1993) was used to assess dynamic effects on pollutant concentration caused by time-varying ventilation for typical weather conditions. For each simulation, RESVENT, DOE-2.1E, and ASHRAE Standard 152P (1999c) were used to determine the capacities of the space conditioning system and to characterize part-load thermal performance. RESVENT then calculated the hourly ventilation-related energy loads on the central HVAC system, based on the actual hourly ventilation rates. At the end of each simulation, RESVENT calculated the annual energy consumption and costs associated with this ventilation. Appendix A summarizes the RESVENT calculations. Data generated by these simulations were then used to assess when to use particular strategies.

HOUSE CHARACTERISTICS

The 36 houses considered in this study were based on prototype slab-on-grade gas-heated houses developed by Brown et al. (1998a, 1998b) to characterize common building practices for new production houses in 12 U.S. cities. Three different floor areas for one- and two-story houses were used (1000, 2000, and 4000 ft^2). These areas and house heights were selected to represent a range of typical house sizes in the United States.

Two occupancies were used for each floor area: two or four occupants for 1000 ft^2 (one or three bedrooms), three or five occupants for 2000 ft^2 (two or four bedrooms), and three or six occupants for 4000 ft^2 (two or five bedrooms).

Sherman and Matson (1997) have used measured building envelope leakage areas to determine a representative range of normalized leakage areas (NL) for U.S. housing. For this study, two normalized leakage areas were selected from that range of data: 0.3 and 1.2. The first value (NL_{new}) represents new construction practices, while the latter (NL_{stock}) represents an average for existing U.S. housing stock.

A third normalized leakage area was also used: NL_{base}. This "base" leakage area is calculated to make infiltration sufficient to meet the minimum whole-house total ventilation rate required by Standard 62.2P. It is derived from the annual effective air change rate calculation of ASHRAE Standard 136 (1993) and is given by

$$NL_{base} = \frac{7.32 \cdot Q_{tot,\,62.2}}{(W \cdot A_f)} . \tag{1}$$

This normalized leakage area is climate dependent because the annual factor W varies with location in the United States over a range of about 0.57 to 1.21 (ASHRAE 1993). Occupancy also implicitly influences this leakage because $Q_{tot,62.2}$ depends on occupancy. This "base" leakage can be larger or smaller than the normalized leakage typical in new construction (NL_{new}). We have chosen this base level as an idealized system that just happens to provide exactly the right amount of leakage. It is simply used as a comparison point rather than representing an achievable design or realistic configuration.

To scale wind data from airport measurements to the building site, the infiltration model requires terrain and shielding information. All the houses were assumed to be located in suburban terrain with some scattered obstructions within two house heights.

VENTILATION STRATEGIES

Overview

Three whole-house mechanical ventilation strategies were modeled in this study. Two of these involved systems that operated continuously: a central exhaust-only system and a heat recovery ventilator system. The third strategy involved a forced-air cycler system, which operated intermittently.

These three strategies were compared against a "no fan" strategy that used infiltration to generate an equivalent ventilation. For some of the tightest configurations, this strategy would not meet the intent of 62.2P as we defined it. Those configurations were removed from the analysis. In every strategy, windows were always closed.

Each strategy included local mechanical exhaust ventilation that met the minimum local ventilation requirements of Standard 62.2P: a 100 cfm kitchen fan operating each day from 17:00 to 17:30, a 50 cfm master bathroom fan operating each day from 06:00 to 06:30, and a 50 cfm bathroom fan operating each day from 07:00 to 07:30. A 250 cfm clothes dryer operating once a week from 20:00 to 21:00 was also included.

Whole-House Mechanical Ventilation Characteristics

Central Exhaust-Only System (EXH). This strategy represents a *continuously operated unbalanced* mechanical ventilation system that is separate from the central HVAC system. Makeup air for the exhaust flow was provided through leakage openings in the building envelope.

The exhaust flow was sized equal to the required whole-house mechanical ventilation rate (Q_{fan}). The exhaust fan power was 0.6 W/cfm of its flow (Sherman and Matson 1997). Fan energy corresponding to the ventilation-related thermal load on the central space conditioning system was accounted for assuming the fan power was 0.5 W/cfm of its total supply flow (Phillips 1998), which was calculated based on 400 cfm/ton of peak cooling capacity. Appendix A describes the fan energy calculation in more detail.

Heat Recovery Ventilator System (HRV). This strategy represents a *continuously operated balanced* mechanical ventilation system that is also separate from the central HVAC system. It has supply and exhaust flows, which were sized equal to the required whole-house mechanical ventilation rate (Q_{fan}).

Total fan power for the HRV was 1.1 W/cfm of its balanced flow. This value is an average derived from HRV fan power and airflow data provided by HVI (1997). Fan energy associated with the ventilation-related thermal load on the central space conditioning system was calculated in the same way as for the central exhaust-only system.

HVI (1997) publishes heat recovery effectiveness and efficiency data, which depend on device type. Using these data along with the definitions of effectiveness in CSA Standard C439 (CSA 1988), the average sensible and total heat recovery effectivenesses for sensible and total heat recovery devices were calculated. For the sensible devices, the average sensible and total heat recovery effectivenesses were 77% and 15%, respectively. For the total devices, these values were 79% and 49%, respectively. Heating and sensible cooling load calculations used sensible effectiveness; cooling load calculations with latent loads used total effectiveness. Sensible heat recovery devices were used in nonhumid climates; total heat recovery devices were used in humid climates.

Forced-Air Cycler System (FAC). This strategy represents an *intermittently operated unbalanced* mechanical ventilation system. The system has a duct from outdoors that connects to the return duct of the central forced-air space conditioning system. Outdoor air flows through this duct due to return duct suction whenever the system blower operates. The outdoor air and return air are mixed together within the return duct. A controller turns on the blower whenever thermal demands are insufficient to cause the system to run a set minimum time in any one-hour period. In our study, this minimum run time was 20 minutes per hour.

The intermittent outdoor airflow was sized using the method described by Rudd and Lstiburek (1999). Assuming the blower operates at its minimum run time of 20 minutes per hour, their sizing equation has the form

$$Q_{FAC} = 3 \cdot \left(Q_{tot,\,62.2} - \frac{2}{3} \cdot Q_{credit}\right). \tag{2}$$

This intermittent airflow is equivalent to a continuous mechanical ventilation rate of Q_{fan} over the 60-minute period associated with the 20-minute blower run time.

Each hour's actual outdoor airflow was based on the actual run time of the blower in that hour but at a fraction of the forced-air outdoor airflow "size":

$$Q_{supply,\,i} = Q_{FAC} \cdot Max(t_{frac,\,i}, 20/60) \tag{3}$$

For each house, thermal demands on the space conditioning system were evaluated using the DOE-2.1E hourly energy simulation program. The output from these simulations was a set of 8760 hourly thermal part-load factors. Each part-load factor represents the fraction of an hour ($t_{frac,i}$) that the system had to run to meet the load imposed upon it by the house thermal demands. Fan energy associated with operating the system blower was calculated in the same way as for the central exhaust-only and HRV systems.

CLIMATES

The severity and wetness of United States climates were categorized using infiltration degree-days (ASHRAE 1994) and using typical-year frequency distributions for outdoor wet-bulb temperatures. Severity classes were defined as follows: mild, less than 5400°F days (3000°C days); temperate, 5400°F to 8100°F days (3000°C to 4500°C days); and severe. The severe climates are cold, except for a small hot region near the southern Gulf coast of Texas. A humid climate was defined as one with outdoor wet-bulb temperatures of at least 67°F (19°C) for a minimum of 3500 hours or of at least 73°F (23°C) for a minimum of 1750 hours during the warmest six months of a typical year for that climate.

Figure 3 shows the distribution of these climates in the continental United States and the locations of 20 of the 22 cities we used. The two cities not shown are Honolulu (mild, humid) and Anchorage (severe, dry). Typical-year representative weather data were used in the hourly simulations for each of these 22 cities. For all but one of these cities, these data were in TMY-2 format. Climate Zone 9 data from the California Energy Commission were used for Pasadena. Each weather file included 8760 hours of outdoor temperature, humidity ratio, wind speed, and barometric pressure data.

FUEL PRICES

To determine energy costs associated with providing whole-house ventilation, statewide average residential electricity and natural gas prices for the last complete year of data (1997) were used in our study (EIA 1998, 1999c).

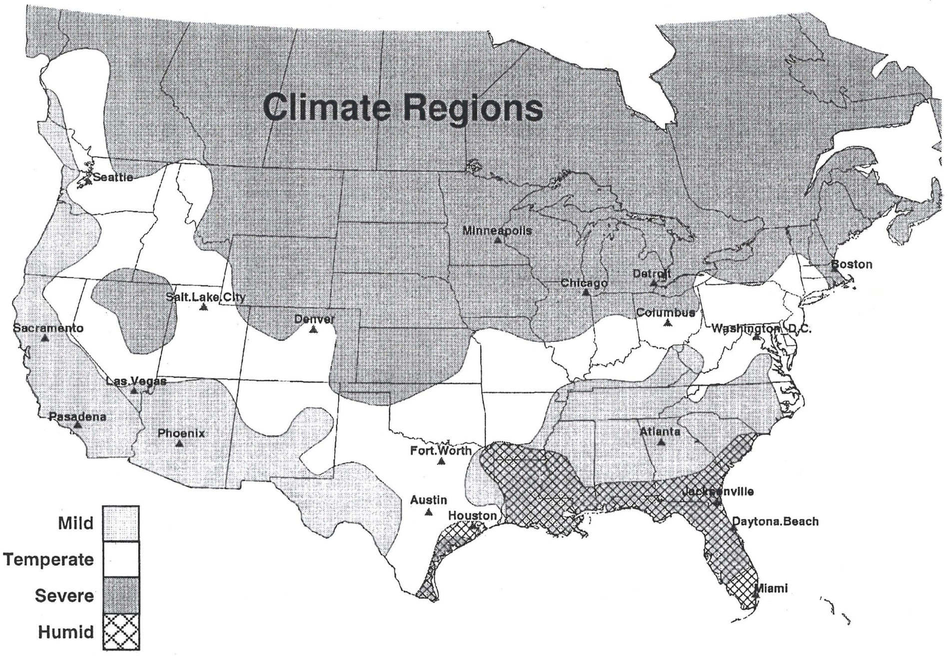

Figure 3 *Continental U.S. climatic regions and locations of 20 selected cities.*

RESULTS

A large number of different configurations were examined. Except for many new houses with infiltration alone (no fan), each configuration met the minimum ventilation requirements of ASHRAE Standard 62.2P. We elected to use ventilation operating cost per unit floor area as the metric to compare the various options. This metric has the advantage that it combines the impact of two fuel types together. Different strategies have different impacts on the heating fuel (gas) compared with the electricity necessary to power fans or air conditioning.

Normalizing by floor area also allows us to compare houses of different sizes. Although the required ventilation depends on the number of bedrooms and on house size, this metric greatly reduces the scatter of the data. Figure 4 shows the mean costs as a function of climate, leakage level, and system type for the configurations that met the intent of Standard 62.2P.

The results vary by climate, but in a relatively predictable way within each climate. Therefore, for each of the 144 configurations in each climate, if we subtract out the no-fan base leakage case, we can then summarize all the data in a single table. Table 1 is the mean difference (and standard deviation) of a particular strategy from our idealized, no-fan base leakage case.

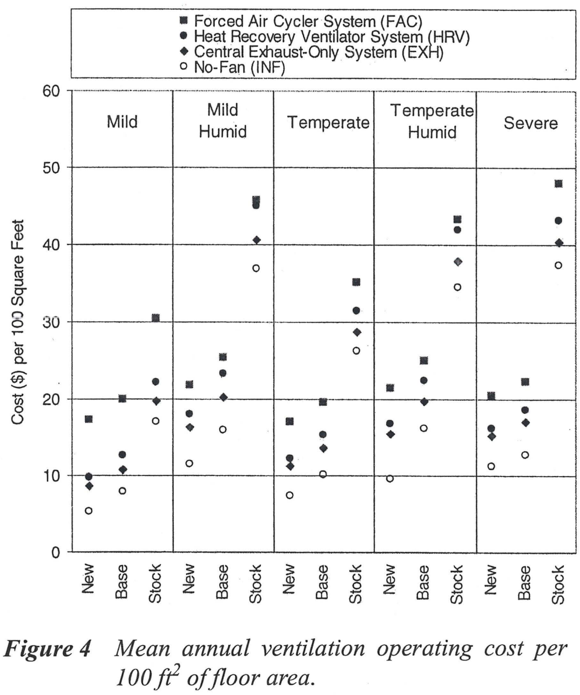

Figure 4 *Mean annual ventilation operating cost per 100 ft^2 of floor area.*

TABLE 1
Incremental Costs from No-Fan Base Leakage Case ($/100 ft^2)*

Leakage	No-Fan (INF)	Exhaust (EXH)	Heat Recovery (HRV)	Forced-Air Cycler (FAC)
Base	0	4.01 (±1.88)	6.01 (±2.90)	10.42 (±3.00)
New	–0.51 (±1.75)	1.42 (±1.57)	2.59 (±1.86)	7.74 (±2.61)
Stock	17.72 (±8.58)	20.61 (±8.38)	23.68 (±8.93)	28.05 (±7.66)

* Many new houses do not meet the intent of the Standard by infiltration alone. Only configurations that met the intent of 62.2 were included in this average.

TABLE 2
Ventilation Operating Costs for No-Fan Base Leakage Case ($/100 ft^2)

Mild	Mild Humid	Temperate	Temperate Humid	Severe
7.37 (±3.09)	15.13 (±6.14)	12.73 (±4.91)	10.03 (±4.24)	14.97 (±5.87)

There are several clear trends in both Figure 4 and Table 1. As expected, the leakier the house, the higher the energy cost. Although the ventilation systems stay in the same order (INF, EXH, HRV, FAC) from lowest to highest leakage in mean operating costs, the standard deviations are large enough that this may not be true for a particular configuration.

As Figure 4 shows, the no-fan base leakage cases vary by climate. To calculate the mean annual operating cost in a particular climate, add the mean incremental values in Table 2 to the respective mean values in Table 1.

DISCUSSION

Our results focus on the energy cost impacts of meeting Standard 62.2P. Since this standard provides a minimum set of requirements, there are additional considerations and value-added features that one might wish to consider. We mention some of them below.

Source of Ventilation Air

Standard 62.2P does not specify the source of air that provides the ventilation nor how the air enters the house. Therefore, in terms of energy performance, we made no distinction based on the source of ventilation air between the systems we analyzed. This means that our operating cost results for a central exhaust-only system also represent those for a central supply-only system.

However, there are some air quality distinctions between the systems we analyzed. Outdoor air is introduced at a single controlled location for forced-air cycler and heat-recovery systems, which allows more control of the air quality. In contrast, for central exhaust systems or for infiltration, outdoor air can enter the house through leakage paths, which do not provide a good opportunity to control the quality of the ventilation air. While pulling air through the building envelope may provide some heat recovery and some filtration, many are concerned about the IAQ impacts of such a system.

System Comparisons

Central Exhaust-Only System. This system proved to be the best overall mechanical ventilation system in terms of operating energy cost. When integrated with a bathroom fan, it can also be among the cheapest first cost options.

There are several considerations that may make exhaust systems less attractive. In hot, humid climates, they can pull moist outdoor air into the building envelope, which can cause condensation problems. With extremely tight envelopes, either an HRV or a supply system might be preferable to avoid depressurization problems. Alternatively, envelope leakage may need to be increased by installing passive inlets to keep excessive depressurization from causing problems. However, due to the fixed infiltration credit in Standard 62.2P, that standard does not recognize the increased infiltration, energy consumption, and costs that will result from adding the passive inlets. As mentioned earlier, exhaust systems pull ventilation air through leaks and the quality of that air cannot be controlled.

Heat Recovery Ventilator System. In terms of operating costs, the HRV system was significantly better than the forced-air cycler system and slightly worse than the central exhaust-only system. This latter ranking can probably be attributed to high electrical consumption of the HRV, caused by using two fan motors and some defrosting. That consumption penalty was offset by reduced ventilation-related thermal losses, but not sufficiently. The latter ranking could be much different in cold climates with more expensive heating, such as with electric heating or higher gas prices.

Given the much higher first cost of the HRV system compared to a central exhaust-only system, an HRV is not likely to be selected based on operating cost considerations alone. For example, it might be used in house configurations for which a central exhaust-only system is inappropriate due to issues such as moisture, filtration, or depressurization.

Forced-Air Cycler System. The forced-air cycler system can be very inexpensive to install but was the worst option in our operating cost comparison. The basic reason was that the air-handling unit supplied excess ventilation air during extreme weather periods and ran frequently to supply ventilation during periods of mild weather.

Whenever the air handler operated, its total flow was from 15 to 40 times larger than the required ventilation rate. When ventilation alone was required, our simulations indicated that the air handler ran from 16% to 22% of the time. Thus, the fan electricity attributable to ventilation was much greater for this system than for a continuously operating ventilation-only system.

This excess ventilation flow during extreme periods could be mitigated by a motorized damper, and excessive fan power could be mitigated by a variable-speed drive. Although these options were not analyzed, we expect the performance of such a system would approach that of the central exhaust-only system. However, its first cost penalty could be prohibitive.

There are non-energy issues to be considered with the forced-air cycler system as well. On the plus side, of the strategies we considered, it was the only one that provided positive air distribution in every three-hour block. On the negative side, in some climates, the ventilation-only cycling of the air handler may cause comfort problems to the occupants. Furthermore, in cold climates, supply systems such as this are not generally recommended because they tend to push moist indoor air into the building envelope, which can cause condensation problems.

No Fan (Infiltration). For any leakage level, it is obvious the energy and first costs of having a mechanical ventilation system will be higher than not having one. Most of the existing building stock is sufficiently leaky that the intent of Standard 62.2P can be met without using a whole-house mechanical system. Failure of the standard to take into account actual infiltration levels can lead to inappropriate system designs.

CONCLUSIONS AND RECOMMENDATIONS

Our conclusions and recommendations are based on the representative configurations we chose and are intended to be generally correct for North America. Specific circumstances may be quite different, especially if cost structures are significantly different.

New Construction

Most new construction is tight enough that infiltration will not provide sufficient ventilation. The marginal energy costs to provide ventilation with a central exhaust-only system in a typical new house would be on the order of 50¢ per day. This can be compared to an infiltration-only cost of $2 per day to condition the air in a typical existing house.

In most cases, the first choice of mechanical systems is a central exhaust-only system. It has low first cost and minimal operating cost. In hot, humid climates or if the envelope is exceedingly tight, either an HRV or a supply system might be preferable to avoid depressurization problems. In cold climates with high heating costs, an HRV might be more cost-effective.

Retrofit Applications

Most of the existing stock is sufficiently leaky that the ventilation rate from infiltration alone would satisfy the intent of Standard 62.2P with respect to whole-house rates. If fan pressurization tests of the building envelope confirm that the leakage is sufficiently above the base level, we do not recommend the addition of a whole-house system. In fact, for much of this stock, the building envelope can be tightened to reduce energy costs and still provide adequate ventilation through infiltration.

Currently, Standard 62.2P does not allow an infiltration credit based on measured airtightness. If this standard is to be applied in retrofits, we believe that it is important to allow this credit so that mechanical systems can be downsized or eliminated.

ACKNOWLEDGMENTS

This work was supported by the Assistant Secretary for Energy Efficiency and Renewable Energy, Office of Building Technology, Building Systems Division of the U.S. Department of Energy under Contract Number DE-AC03-76SF00098.

The authors wish to acknowledge the DOE-2 technical support provided by Joe Huang and Ender Erdem of the Simulation Research Group at Lawrence Berkeley National Laboratory.

NOMENCLATURE

A_f	= conditioned floor area (ft^2)
AFUE	= furnace annual fuel utilization efficiency (0.78)
C_p	= heat capacity of air (0.240 Btu/(lb_m °F))
COP	= cooling system coefficient of performance (3.18)
$Cost_{elec}$	= electricity unit price [$/kWh)
$Cost_{gas}$	= gas unit price ($/therm)
$E_{cooling,annual}$	= annual cooling energy consumption (kWh)
E_{cost}	= annual energy cost associated with ventilation ($)
$E_{fans,annual}$	= annual electrical consumption of fans (kWh)
$E_{heating,annual}$	= annual heating energy consumption (therm)
FH_{frac}	= free heat due to internal and solar gains, as a fraction of annual space heating load (Hanford and Huang 1992)
h_{in}	= indoor enthalpy at 75°F and 50% relative humidity (Btu/lb_m)
$h_{out,i}$	= outdoor enthalpy in hour i, based on outdoor temperature and humidity ratio for that hour (Btu/lb_m)
i	= hour of year

NL = normalized leakage area defined in ASHRAE Standard 119

$Q_{cooling,sensible,i}$ = sensible cooling load for hour i (Btu/h)

$Q_{cooling,total,i}$ = total cooling load for hour i (Btu/h)

Q_{credit} = infiltration credit (cfm)

Q_{fan} = required whole-house mechanical ventilation rate (cfm)

Q_{FAC} = forced-air cycler maximum airflow (cfm)

$Q_{heating,i}$ = sensible heating load for hour i (Btu/h)

$Q_{inf,i}$ = infiltration rate from the LBL infiltration model in hour i (cfm)

$Q_{l,i}$ = larger of total supply and total exhaust airflows in hour i (cfm)

$Q_{s,i}$ = smaller of total supply and total exhaust airflows in hour i (cfm)

$Q_{supply,i}$ = forced-air cycler duct airflow in hour i (cfm)

$Q_{tot,i}$ = actual total ventilation rate in hour i (cfm)

$Q_{tot,62.2}$ = minimum required whole-house total effective airflow rate (cfm)

$t_{frac,i}$ = fractional on-time of space conditioning system in hour i

$t_{run,annual}$ = annual total fractional run time difference of blower

$T_{deadband}$ = temperature deadband between heating and cooling (6.3°F)

$T_{heating}$ = indoor heating setpoint temperature (66°F)

$T_{out,i}$ = outdoor temperature in hour i (°F)

W = weather factor defined in ASHRAE Standard 136 (ach)

$\varepsilon_{recovery}$ = heat recovery effectiveness

ρ = density of air (0.075 lb_m/ft^3)

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APPENDIX A

VENTILATION, ENERGY, AND COST CALCULATIONS

Hourly Ventilation Rate Calculations

In our study, the actual hourly infiltration rates throughout the year were calculated by a modified version of the RESVENT computer program developed by Sherman and Matson (1993, 1997), which uses the LBL infiltration model (Sherman and Modera 1984). RESVENT then superimposed each hour's infiltration rate with the corresponding mechanical whole-house and local ventilation rates to calculate the actual hour-by-hour ventilation rate $Q_{tot,i}$:

$$Q_{tot,\,i} = \sqrt{Q_{inf,\,i}^2 + (Q_{l,\,i} - Q_{s,\,i})^2} + Q_{s,\,i} \tag{A1}$$

Equation A1 differs from the analogous superposition equation that is presented in Section 4.4 of ASHRAE Standard 136 (ASHRAE 1993). The equation used here is assumed to provide better estimates for the combination of infiltration with unbalanced and balanced ventilation flows (Sherman 1992).

The actual hour-by-hour rates $Q_{tot,i}$ were used as input for the effective ventilation model of Sherman and Wilson (1986) to determine the hour-by-hour effective ventilation rates. Each of the ventilation strategies studied always provided adequate ventilation because each strategy was developed to just comply with the minimum whole-house ventilation requirements of Standard 62.2P.

Annual Energy Consumption Calculations

Ventilation-related energy consumption has two components. One is related to the energy loss or gain caused by heating or cooling the air that enters the conditioned spaces. The other is the parasitic energy demand imposed by fan operation, both for the ventilation system and the space conditioning system.

Annual ventilation-related space conditioning loads and energy consumption were calculated using the methodology outlined by Sherman and Matson (1997). In summary, the energy used to heat or cool ventilation air depends on the air mass flow rate, the indoor-outdoor enthalpy difference, and the heat recovery effectiveness. Using these parameters, we first determined the ventilation-related loads on the space conditioning system for each hour of the year. Then, using the sum of these loads over the entire year, the ventilation-related annual space conditioning energy consumption was determined. Equations A2 through A6 describe these calculations.

$$Q_{heating,\,i} = 60\rho Q_{tot,\,i} C_p (T_{heating} - T_{out,\,i})(1 - \varepsilon_{recovery}) \tag{A2}$$

$$Q_{cooling,\,sensible,\,i} = 60\rho Q_{tot,\,i} C_p [T_{out,\,i} - (T_{heating} - T_{deadband})](1 - \varepsilon_{recovery}) \tag{A3}$$

$$Q_{cooling,\,total,\,i} = 60\rho Q_{tot,\,i}(h_{out,\,i} - h_{in})(1 - \varepsilon_{recovery}) \tag{A4}$$

$$E_{heating,\,annual} = \left[\frac{1 - FH_{frac}}{1000 \cdot AFUE}\right] \cdot \sum_{i=1}^{8760} Q_{heating,\,i} \tag{A5}$$

$$E_{cooling,\,annual} = \left[\frac{1.055}{3600 \cdot COP}\right] \cdot \sum_{i=1}^{8760} Max(Q_{cooling,\,sensible,\,i},\, Q_{cooling,\,total,\,i}) \tag{A6}$$

The fan energy directly associated with ventilation can be subdivided into two components. One component is the energy for the ventilation fans that move air across the building envelope. The other component is the fan energy of the forced-air system blower that delivers the ventilation-related space conditioning energy within the house.

For the central exhaust-only and heat recovery ventilator strategies, the fan energy to move the ventilation air across the building envelope was equal to the ventilation fan power multiplied by the total run time over the year (8760 hours) for each of the whole-house and local ventilation system fans. The only fan energy separate from the central forced-air system for the forced-air cycler strategy was due to the operation of local ventilation fans in that strategy.

For all ventilation strategies, the central forced-air system blower operated intermittently rather than continuously. The fan energy associated with operating the central HVAC system blower due to thermal loads caused by ventilation (including infiltration) was equal to the blower fan power multiplied by the total run time associated with those loads over the year. For heating, that run time was estimated as the total ventilation heating load over the year ($Q_{heating,i}$) divided by the heating system capacity; for cooling, the total ventilation cooling load over the year ($Q_{cooling,total,i}$) and peak cooling system capacity were used instead.

For the forced-air cycler strategy, there was also fan energy associated with ventilation (including infiltration) during hours when there was no thermal load because the ventilation system controller sometimes caused the central forced-air system blower to run during these hours. The fan energy associated with operating this blower due to ventilation in these hours was equal to the blower fan power multiplied by the number of these hours, which was calculated as

$$t_{run,\,annual} = 8760 \sum_{i=1}^{8760} \left[Max\left(0, \frac{20}{60} - t_{frac,\,i}\right)\right]. \qquad \text{(A7)}$$

Equation A7 only accumulates differences for hours when house thermal demands, including those related to infiltration and local ventilation, would have caused the blower to run less than 20 minutes per hour in the absence of the forced-air cycler strategy.

Annual Energy Cost Calculations

The energy costs associated with ventilation depend on the amount and price of energy consumed to provide the ventilation:

$$E_{\cos t} = E_{heating,\,annual} \times Cost_{gas} + (E_{cooling,\,annual} + E_{fans,\,annual}) \times Cost_{elec} \qquad \text{(A8)}$$