Title
Meeting Residential Ventilation Standards Through Dynamic Control of Ventilation Systems

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Publication Date
2011-05-31

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Meeting Residential Ventilation Standards Through Dynamic Control of Ventilation Systems

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April 2011

This work was supported by the California Energy Commission Public Interest Energy Research Program award number CEC-500-07-006 and the Assistant Secretary for Energy Efficiency and Renewable Energy, Building Technologies Program, of the U.S. Department of Energy under contract No. DE-AC02-05CH11231.

Disclaimer

ABSTRACT

Existing ventilation standards, including American Society of Heating, Refrigerating, and Air-conditioning Engineers (ASHRAE) Standard 62.2, specify continuous operation of a defined mechanical ventilation system to provide minimum ventilation, with time-based intermittent operation as an option. This requirement ignores several factors and concerns including: other equipment such as household exhaust fans that might incidentally provide ventilation, negative impacts of ventilation when outdoor pollutant levels are high, the importance of minimizing energy use particularly during times of peak electricity demand, and how the energy used to condition air as part of ventilation system operation changes with outdoor conditions. Dynamic control of ventilation systems can provide ventilation equivalent to or better than what is required by standards while minimizing energy costs and can also add value by shifting load during peak times and reducing intake of outdoor air contaminants. This article describes the logic that enables dynamic control of whole-house ventilation systems to meet the intent of ventilation standards and demonstrates the dynamic ventilation system control concept through simulations and field tests of the Residential Integrated Ventilation-Energy Controller (RIVEC).

Keywords: Mechanical Ventilation, Ventilation Effectiveness, Residential Ventilation

INTRODUCTION

The purpose of ventilation is to dilute or remove indoor air contaminants to which building occupants would otherwise be exposed. In addition to improving indoor air quality (IAQ), ventilation is a critical element for improving the energy efficiency of buildings. IAQ is a result of complex interactions among occupant activities and responses, source emission, and contaminant removal. Of these elements, ventilation and source control are the two for which requirements can most easily be established. Determining these requirements depends on an understanding of the materials and processes typically found in houses and the operational strategies of occupants. Almost every building code has ventilation and IAQ requirements, but most codes do not take an integrated approach to residential IAQ; they do not, for example, consider how outdoor weather or air quality might impact the value of ventilation. This study used the only national consensus standard on residential ventilation and indoor air quality, American National Standards Institute (ANSI)/ American Society of Heating, Refrigerating, and Air-conditioning Engineers (ASHRAE) Standard 62.2-2007, as its reference. The standard in its simplest form requires continuous, mechanical, whole-house ventilation at a rate based on the size and occupancy of the house. Although the standard specifies certain performance conditions for mechanical ventilation, it also allows the use of dual-purpose fans (e.g., bath exhausts) to meet whole-house requirements. It also provides a methodology for using time-varying mechanical ventilation.

ASHRAE Standard 62.2 does not account for the fact that, in a typical occupied house, a variety of activities independent of the whole-house ventilation system will ventilate the home, such as the use of kitchen and bathroom exhaust fans, economizers, and clothes dryers. In addition, the standard does not take into account that it is beneficial, e.g., for energy-efficiency or air quality reasons, to at times temporarily reduce or eliminate mechanical ventilation. This article describes and demonstrates a dynamic ventilation control method that takes full advantage of exogenous mechanical ventilation and shifts the operation of the ventilation system to desirable times of day by controlling the whole-house ventilation fan. The performance of the dynamic ventilation control method was determined using energy simulations and a field demonstration of a prototype device, the Residential Integrated Ventilation Energy Controller (RIVEC). The control method described was developed by Sherman, Walker, and Dickerhoff (2009) during work carried out for the California Energy Commission’s Energy Innovations and Small Grant Program.

DEVELOPMENT OF EQUIVALENCE

To develop a dynamic whole-house mechanical ventilation system control technology that can take into account desired ventilation patterns and exogenous ventilation activities requires that we first establish a clear equivalence between what the ventilation standard would achieve and what our dynamic control would achieve.

The basic requirement of ASHRAE Standard 62.2 is that a whole-house mechanical ventilation system operate continuously at a rate equal to 1 cubic foot per minute (cfm) per hundred square feet of floor area plus 7.5 cfm per person; the number of people in a household is presumed to be equal to one more than the number of bedrooms [5 L/s per 100 square meters plus 3.5 L/s per person]. Infiltration from envelope air leakage contributes to total ventilation; however, the standard does not require modification to the mechanical ventilation rate based on envelope air leakage, so this topic is not considered in our study. Thus, the standard gives us a fixed target whole-house airflow rate that can be used together with an assumed constant pollutant generation rate to calculate an occupant’s exposure to a pollutant. The dynamic controller needs to achieve the same, or lower exposure to demonstrate equivalent indoor air quality.

Standard 62.2 also requires that kitchens and bathrooms be equipped with exhaust fans that can provide ventilation of at least 100 cfm and 50 cfm, respectively. The standard does not specify a minimum operating time for these fans as it does for whole-house mechanical ventilation. The standard contains additional requirements regarding source control, mechanical equipment properties, commissioning, etc. These do not affect our effort to achieve pollutant exposure equivalent to what the standard can achieve, so they are not addressed here. The reader can find additional information on these issues in the standard and its associated user’s guide (ASHRAE 2007).

It is important to note that the standard is very flexible about how one may achieve the minimum ventilation: supply ventilation, exhaust ventilation, balanced ventilation, or appropriate combinations thereof may be used. Systems that incidentally ventilate (such as bath fans, dryers, or economizers) may be counted towards the total if they meet the basic requirements. The dynamic controller used in our study uses this flexibility to reduce the energy required for whole-house ventilation and conditioning of ventilation air.

Intermittent Ventilation

The energy and cost impacts of ventilation vary with weather and time of year. Much of this is seen in diurnal variations. For example, at night in the winter (or afternoon in the summer), the large temperature difference between outdoors and indoors makes ventilation air more energy intensive (and therefore expensive) to condition. In addition, increased cost per unit of energy at times of peak demand may make it advantageous to shift ventilation load to other parts of the day. Furthermore, some regions have frequent, short-term outdoor air quality problems (such as elevated ozone levels) that could be partially mitigated by shutting off mechanical ventilation during those periods. The dynamic ventilation control strategy developed here allows the whole-house mechanical ventilation system to operate intermittently, which reduces energy use and cost and enables less ventilation when outdoor air is of poor quality. To fully address all these possibilities would require measurement of indoor and outdoor weather conditions, knowledge of current utility tariffs (including those that are dynamic themselves), and use of outdoor pollutant sensors. Because accomplishing all of these goals would be complex and expensive, we first sought a simpler solution in which a ventilation system's operation would depend on the time of day. This study follows the philosophy used in ASHRAE 62.2's provisions for the use of intermittent ventilation.

The key equations of intermittent ventilation (Sherman 2006) define the efficacy of ventilation as it relates to the ventilation pattern: temporal ventilation effectiveness, i.e., efficacy (ε), is the ratio between the ventilation one would need if the rate were constant and the actual ventilation; for our simple case, temporal effectiveness links the equivalent (or desired) steady-state ventilation rate (A_{eq}), the actual (or needed) rates of over-ventilation and under-ventilation (A_{high} and A_{low}), and the fraction of time that a space is under-ventilated (f_{low}):

$$\varepsilon = \frac{A_{eq}}{f_{low} A_{low} + (1 - f_{low}) A_{high}} \tag{1}$$

If we have an independent measure of efficacy, we can use it and Equation 1 to determine the range of acceptable design parameters. The solution is expressed in dimensionless terms involving the efficacy and two other parameters (f_{low} and N):

$$\varepsilon = \frac{1 - f_{low}^2 \mathrm{N} \cdot \coth(\mathrm{N}/\varepsilon)}{1 - f_{low}^2} \tag{2}$$

where the *nominal turnover* of air, N, is defined as follows:

$$\mathrm{N} \equiv \frac{(A_{eq} - A_{low}) \cdot T_{cycle}}{2} \tag{3}$$

We address here what we consider will be the most typical use of a household ventilation system, intermittent or "notch ventilation," in which the system is shut off for a period of time during the day when, for example, energy costs or outdoor pollution are high. In this case, we assume that the ventilation system is shut off for four hours per day during the period of peak energy demand and is operating continuously for the other 20 hours. Using the equation above and the rates specified in ASHARE Standard 62.2 and typical housing values, the efficacy in that situation would be 96%. Therefore, we should have a mechanical ventilation system that has 25% more airflow when it is operating in the intermittent mode than if it were operating continuously. This approach assumes that the whole-house mechanical ventilation system has a constant ventilation rate during the 20 hours when it is operating. To capture the effects of non-constant ventilation rates, we must generalize a bit further.

The intermittent ventilation algorithms cited above are based on the work of Sherman and Wilson (1986), which looked at ventilation time variations on a fixed schedule. To generalize to ventilation rates that are not on a fixed schedule, we need to develop an equivalent way to determine IAQ. We do this by following Sherman and Wilson to determine the equivalent exposure to a general but constant (or uncorrelated) contaminant exposure. For such a case, the key parameter is the *turnover time*, τ , of the indoor air:

$$\tau_e(t) = \int_{-\infty}^{t} e^{\int_{t}^{t'} A(t'')dt''} dt' \tag{4}$$

If we have a target constant ventilation rate that will result in appropriate absolute exposure to a contaminant, then the *relative exposure*, R, is the product of the target multiplied by the instantaneous turnover:

$$R(t) = A_{eq}\tau_e(t) \tag{5}$$

The intermittent ventilation equations are based on providing the same steady-state dose of a contaminant over any cycle time of interest. The *relative dose*[1]*, d,* is the average relative exposure over any steady-state cycle, T:

$$d = \frac{1}{T}\int_{0}^{T} R(t)dt = 1/\varepsilon = A_{eq}\bar{\tau} \tag{6}$$

The efficacy used in the intermittent ventilation equations is the inverse of the relative dose and can be related to the average turnover time for the period.

The equations above are useful for continuous unbounded data, but for many purposes it is more useful to use a recursive formula for discrete data. We can rewrite the expression for turnover time as follows:

$$\tau_i = \frac{1-e^{-A_i\Delta t}}{A_i} + \tau_{i-1}e^{-A_i\Delta t} \tag{7}$$

For the intermittent ventilation ("notch") strategy, the relative exposure, calculated from the discrete turnover time, has a minimum value of approximately 0.8 just before the notch time and a value of twice that at the end of the notch time. This value (0.8) is used in the control algorithm to decide when to turn on the whole-house ventilation system.

We can also write an expression for the (recursive) discrete relative dose. This value varies only a few percent from unity for notch ventilation:

$$d_i = A_{eq}\tau_i(1-e^{-\Delta t/24hrs}) + d_{i-1}e^{-\Delta t/24hrs} \tag{8}$$

CONTROL APPROACH

Ideally a dynamic controller would be able to manage any mechanical ventilation system that is installed, meeting whole-house mechanical ventilation requirements at minimum energy cost. The controller can do this by shifting the ventilation load of the whole-house mechanical ventilation system off peak and taking into account exogenous mechanical ventilation by other systems.

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[1] This is would only be the true toxicological dose under special conditions, but the difference is not important for the context in which we are using it.

To accomplish this, the controller must be able to regulate the state of the installed mechanical ventilation system and to sense the status of all significant exogenous mechanical ventilation systems. To prevent rapid cycling or switching of the whole-house ventilation fan, the controller makes decisions at fixed time steps or intervals, e.g., every 10 minutes. To perform the necessary calculations, the controller must be programmed with a variety of specific house and system parameters such as:

- Floor area of house
- Volume of house
- Number of bedrooms
- Target ventilation rate in air changes per hour (ACH) A_{eq}. The default is calculated from above using ASHRAE 62.2
- Peak hours for minimizing ventilation
- Properties of whole-house mechanical ventilation system: airflow capacity (converted to ACH), Type: (exhaust, supply, heat recovery ventilator [HRV], central fan integrated supply [CFIS])
- Properties of each exogenous mechanical ventilation system: airflow capacity (converted to ACH), type: (exhaust, supply, balanced)

Controller Logic

A reasonable control logic to achieve the intent of ASHRAE Standard 62.2 while minimizing energy costs and operation during peak times would be the following set of actions at each time step:

1. Determine current mechanical ventilation rate: The controller monitors the status of the whole-house mechanical system and all exogenous mechanical ventilation systems. It then totals all exhaust flows separately, all supply flows separately, and all balanced flows separately. The current mechanical ventilation rate will be the balanced flow added to the larger of the supply flow or exhaust flow.

2. Estimate current IAQ: Using equations from the IAQ section above, determine the relative exposure, *R*, and relative dose, *d*. These values will be used in the control algorithm.

3. Modify whole-house mechanical ventilation: Based on the control algorithm, the whole-house mechanical ventilation will be turned on or off for the next period.

The control algorithm aims to keep the dose at or below unity and to shut off the whole-house mechanical ventilation system during a designated peak period. To accomplish this, each day is broken up into four periods. There is a (four-hour) peak period during which the whole-house system is off. There are pre-peak and post-peak shoulder periods (4 hours each), and a 12-hour base period, with control logic as follows:

Base Period:

- Minimum Total Mechanical Ventilation: A_{eq}
- Turn on whole-house ventilation: if d is greater than unity or R is greater than 0.8
- Turn off whole-house ventilation: if d is less than unity and R is less than 0.8

Pre-Peak Shoulder Period:

- Minimum whole-house mechanical ventilation: 0
- Turn on whole-house ventilation: if R is greater than 1 but not if the current mechanical ventilation rate is greater than 1.25 A_{eq}
- Turn off whole-house ventilation: if d is less than unity and R is less than 1

Peak Period:

- Minimum whole-house mechanical ventilation: 0
- Turn off whole-house ventilation: always

Post-Peak Shoulder Period:

- Minimum whole-house mechanical ventilation: 0
- Turn on whole-house ventilation: if d is greater than unity or R is greater than 1 but not if the current mechanical ventilation rate is greater than $1.25\,A_{eq}$
- Turn off whole-house ventilation: if d is less than 1 and R is less than 1 or if the current mechanical ventilation rate is greater than $1.25\,A_{eq}$

SIMULATIONS

We simulated the dose and exposure of a generic pollutant with a ventilation system regulated by a dynamic controller of the type described above. We used the simulation tool described by Walker and Sherman (2006), which employs a combined ventilation and heat-transfer model of a home along with a detailed heating, ventilation, and air conditioning (HVAC) model. The ventilation model combines multiple localized and distributed leaks with mechanical ventilation systems using short time steps (one minute in this study) to allow for thorough evaluation of intermittent ventilation systems as well as ventilation systems integrated into heating (natural gas furnace) and cooling equipment. Typical uncertainties for whole house ventilation flow rates (combining both mechanical and natural infiltration) are 5% to 10% and for attic and house temperatures are 1 ºC (Walker et al. (2005)). Uncertainties for energy consumption are about 4% (Siegel et al. (2000)).

The dynamic controller logic described above was programmed into the simulation software to control the whole-house mechanical ventilation system. The simulations tracked the electrical energy used to operate the mechanical ventilation systems; however, the majority of the energy used for ventilation is to condition the air. The total energy use was separated into the natural gas used for heating and the electricity used for cooling and for operating the mechanical ventilation system. The simulations also calculated total airflows and relative pollutant dose and exposure. The exposure and dose calculations assume a constant pollutant emission rate and are normalized relative to the minimum airflow requirements of ASHRAE Standard 62.2 using Equations 5 and 8 above. Simulations were also performed without dynamic control of the whole house ventilation system so that the changes in energy and ventilation rate attributable to dynamic control could be determined.

Three different climates were selected from the climate zones defined in California's State Energy Code to represent a wide range of climate conditions: a mild coastal climate (represented by Oakland CA), a warm inland climate with substantial summer cooling (represented by Fresno CA), and a cold mountain climate (represented by Mt. Shasta CA). The house that was simulated was based on the 1764 ft^2 [164 m^2] two-story Prototype C home used in the California State Energy Code (CEC 2005) – with a Title 24 compliant building envelope (in terms of insulation levels, air leakage, window orientation and performance) and HVAC system. This home is typical of new construction in many US states.

The wide variety of available ventilation systems has been reviewed by Russell et al. (2007). This study concentrated on systems that are typically found in residences and are of interest to the residential ventilation industry:

- Continuous exhaust. Continuous exhaust in this study refers to the whole house mechanical ventilation system used to comply with the whole-house requirements in Section 4 of ASHRAE Standard 62.2. It is not referring to the local exhaust requirements of Section 5 of the standard. An exhaust ventilation system is typically a exhaust fan that is ducted directly outside - usually from a bathroom, kitchen or laundry room. "Continuous Exhaust" is continuous as far as the user is concerned in that it is "on" all the time. It may, however, be temporarily shut off by the controller when the controller determines that system operation is not necessary for a period of time.

- Heat recovery ventilator (HRV). A residential HRV includes both supply and exhaust airflows to the home - each with their own blower. They can be connected to the forced air heating and cooling system for distribution or have their own dedicated duct system. A heat exchanger (usually of crossflow design) is used to temper the incoming supply air by exchanging energy with the exhaust air. Unlike the continuous exhaust or CFIS systems the HRV are balanced - with equal supply and exhaust air flows.

- Continuous exhaust + Central Fan Integrated Supply (CFIS). This system combines a continuous exhaust with a supply system using the central forced air system blower and ducts to distribute ventilation air. The supply ventilation air enters the return side of the duct system via a duct from outside and is supplied to the home via the supply air registers of the heating and cooling system. Often these systems have controllers to ensure a minimum run time for the blower and/or to close the outside duct with a damper if the system operation for heating or cooling run time exceeds that required for ventilation purposes. In the simulations the CFIS system operated for 20 minutes out of each hour.
- Continuous supply. The continuous supply system uses its own ducts to distribute supply ventilation air. The supply air is tempered by mixing (in the ratio of 3:1) house air with the supply air, thus the continuous supply fan moves four times the air flow needed to meet ASHRAE 62.2.

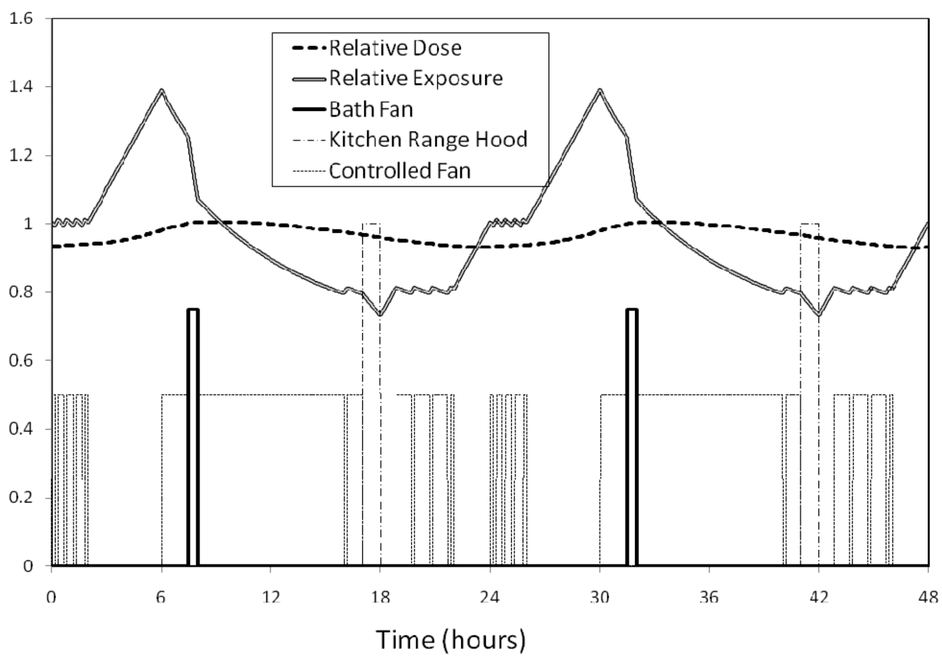

Figure 1. Simulated controlled whole-house ventilation fan (continuous exhaust) and other household fan operation during the winter (hour zero is midnight)

Figure 1 shows the effect on relative dose and exposure of changes in operation of the whole-house controlled fan (in this case a continuous exhaust) as well as other ventilation fans during two days of winter. In this and subsequent figures, the relative exposure and relative dose are shown on the left scale. The indicators for fan operation are in arbitrary units and are simply there to indicate whether the fan was on or off. Figure 1 shows that the whole-house mechanical fan is turned off when other ventilation-inducing equipment is used. In the heating mode, the controller is set to shut off the ventilation during the four-hour pre-dawn period of the day when ventilation is most energy intensive and expensive to operate. After that period, the controller keeps the ventilation system operating until it has brought IAQ back to acceptable levels; after that, the controller cycles the system as needed. The relative exposure changes rapidly and cycles above and below unity in response to the operation of the various exhaust fans. The relative dose changes much more slowly and exhibits much smaller extremes, as expected for this integrated parameter. The most important observation is that the controlled fan responds in the manner prescribed by the control algorithm so that the relative dose stays below unity except during the peak (predawn) period. Even at the end of the peak period, when the controlled fan has been off for four hours, the off-peak operation is sufficient to keep the relative dose within 1% of unity. To see whether these short-term observations were consistent with performance at other times of the year, we calculated the annual average dose and the peak relative dose, as shown in Table 1. These results show that dynamic system operation using the control algorithm results in lower annual dose than a continuously operating system. With regard to the concern that peak exposure levels could rise too high when the ventilation is turned off for four hours at a time, we found that the highest peak values were only 11% above the target, which is a reasonable result.

Table 1. Summary of Simulation Results for Relative Dose using Dynamic Ventilation Controller

Climate	Ventilation System	Annual Average Relative Dose	Peak Relative Dose
Mild	Continuous Exhaust	0.986	1.11
Mild	Continuous Exhaust + CFIS	0.955	1.09
Mild	Continuous Supply	0.986	1.11
Mild	Continuous Exhaust + Economizer	0.923	1.11
Warm	Continuous Exhaust	0.971	1.11
Warm	Continuous Exhaust + CFIS	0.947	1.08
Warm	Continuous Supply	0.971	1.11
Warm	Continuous Exhaust + Economizer	0.860	1.11
Cold	Continuous Exhaust	0.996	1.11
Cold	HRV	0.927	1.08
Cold	Continuous Exhaust + CFIS	0.963	1.09
Cold	Continuous Supply	0.996	1.11

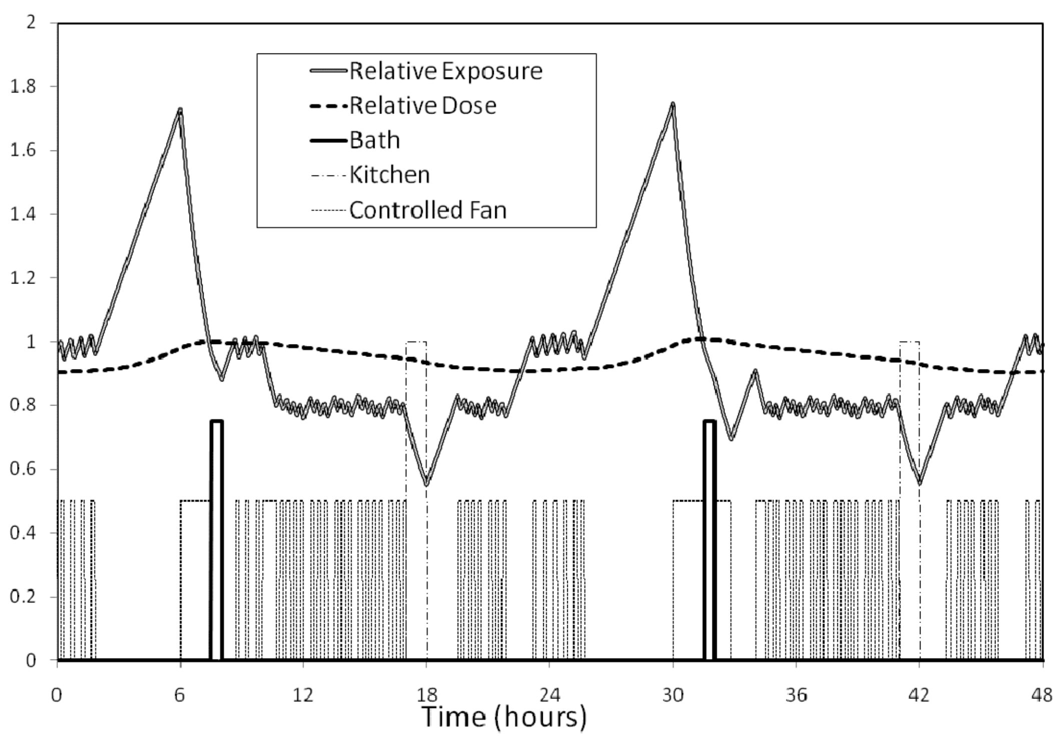

Figure 2. Simulated controller operation for a house with a cycling HRV for a whole-house fan (hour zero is midnight)

Figure 2 shows how the dynamic controller operates with an HRV system. Because most HRVs are oversized for minimum ventilation requirements, the controller cycles them on and off even when no other fans operate. The HRV is integrated into the air handling system so that in the continuously operating base case the HRV would operate 35% of each hour (and require air handler operation as well). Figure 2 shows the cycling behavior of the dose- and exposure-controlled HRV as well as the additional "off" periods determined by the controller when other fans operate and during the 4-hour pre-dawn peak. Because the HRV has such a high airflow, the relative exposure drops more quickly to the target value in the post-peak and base periods compared to exhaust system shown in Figure 1. This is particularly apparent in the base time period (hours 10 to 22 and 34 to 46) where the HRV reaches the relative exposure of 0.8 rapidly and controls to this value, but the continuous exhaust takes several hours to reach this condition. Because the HRV is able to quickly reach the target relative exposure faster it spends more time with a lower exposure and therefore has a lower relative dose compared to the other systems.

Table 2 summarizes the energy savings from the dynamic controller for ventilation systems in the three climate zones. The savings are relative to the same type of ventilation system operating continuously. The annual air changes per hour is reduced by the operation of the controller and that results in an energy savings in both gas and electricity. The fractional savings is the fraction of ventilation related energy savings for dynamic control relative to non-dynamic control. The savings are greater for some systems than others in part because the different ventilation systems produce different air change rates and operate at different times of the day, thus subject to different weather conditions.

Table 2. Summary of Simulation Results for Energy saved by Dynamic Ventilation Control

Climate	Ventilation System	Reduction in Average ACH[h^{-1}]	Electricity Savings (site kWh)	Gas Savings (site kWh) [Therms]	kWh Percent Savings, %	Renormalized kWh Percent Savings %
Mild	Continuous Exhaust	0.017	36	229 [7.8]	22	24
Mild	Continuous Exhaust + CFIS	0.025	41	258 [8.8]	28	34
Mild	Continuous Supply	0.021	254	237 [8.1]	61	64
Mild	Continuous Exhaust + Economizer	0.018	48	223 [7.6]	24	35
Warm	Continuous Exhaust	0.023	81	167 [5.7]	25	29
Warm	Continuous Exhaust + CFIS	0.03	98	179 [6.1]	32	39
Warm	Continuous Supply	0.015	284	275 [9.4]	49	53
Warm	Continuous Exhaust + Economizer	0.034	95	179 [6.1]	26	49
Cold	Continuous Exhaust	0.017	50	316 [10.8]	19	20
Cold	HRV	0.036	367	-111 [-3.8]	11	20
Cold	Continuous Exhaust + CFIS	0.025	71	604 [20.6]	37	42
Cold	Continuous Supply	0.022	265	627 [21.4]	28	28

Energy savings do not vary much between the mild and warm climates; savings are greater in the cold climate mostly because the indoor-outdoor temperature differences are greater for the cold climate (although the fractional savings are not significantly different). The only anomaly is for HRV operation.

The HRV is interlocked to the central forced air system and the fan power when operating is several hundred Watts. This acts to offset the demand for natural gas for heating, such that reducing the operating time of the HRV can result in increased natural gas use for heating. These energy savings are lower than they could be because in each case the dynamically controlled ventilation systems provided better indoor air quality, as shown by the annual average relative dose results in Table 1. In a simplified attempt to account for this effect the last column of Table 2 shows the savings if the energy consumption assuming the dynamic controller was adjusted to provide exactly the same indoor air quality and that the energy use would scale linearly with the annual average dose. Because the systems all provide somewhat better indoor air quality, the renormalized savings are higher. This effect is highest for the HRV and economizers systems because they provide the best indoor air quality. Energy savings could be maximized by modifying the control algorithm to **not** improve indoor air quality.

The savings quoted are in terms of site energy. Monetary savings will depend on the relative cost of gas and electricity as well as rate structures, including various types of utility incentives. For rates currently typical in California, the annual savings without any time-of-use or peak pricing range from $20 to $85.

Savings from dynamic ventilation control depend on climate, the type of whole-house ventilation system used, and how occupants use other ventilation-related equipment such as kitchen and bath fans and clothes dryers. The impact of these factors is evident in the whole-house system run times summarized in Table 3. The table shows the fractional run time of the ventilation system which is dynamically controlled. The “Reduction” column is the percentage reduction in run-time fraction compared to a 62.2 compliant system running 20 hours on and 4 hours off.

Table 3. Simulation Results for Run time for whole-house fans with dynamic control

Climate	Ventilation System	Fractional Run time	Reduction
Mild	Continuous Exhaust	0.65	22%
Mild	Exhaust + CFIS	0.59	29%
Mild	Continuous Supply	0.65	22%
Mild	Economizer	0.57	31%
Warm	Continuous Exhaust	0.65	22%
Warm	Exhaust + CFIS	0.60	28%
Warm	Continuous Supply	0.65	22%
Warm	Economizer	0.49	41%
Cold	Continuous Exhaust	0.65	22%
Cold	HRV	0.30	18%
Cold	Exhaust + CFIS	0.59	29%
Cold	Continuous Supply	0.65	22%

These results show that, for the typical usage of non whole-house ventilation fans, the fractional run time for the most of the whole-house ventilation systems with dynamic control is between 60% and 65% representing a reduction of about 25% of what it would be without dynamic control. For the continuous exhaust, exhaust + CFIS and continuous supply, these results are fairly independent of climate. Because their operation depends on outdoor conditions, reduction in whole-house fan operation for economizers is highly climate dependent as shown by the differences between the mild and warm climates. The same effect can be true for direct evaporative air conditioning. The economizer systems show the highest savings because economizer operation is not considered part of the whole-house ventilation system, so the large economizer airflows give the opportunity to turn off the whole whose system for considerable periods of time. The reduction for the HRV system is lowest because the HRV runs only 35% of the time in the non-dynamic control case.

CONTROLLER FIELD DEMONSTRATION: RIVEC

Sherman, Walker, and Dickerhoff (2009) created a prototype dynamic controller called the Residential Integrated Ventilation-Energy Controller (RIVEC). RIVEC was developed to assess the feasibility of using a residential ventilation controller to: reduce the energy impact of required mechanical ventilation by 20%, maintain or improve IAQ, and provide demand-response benefits. RIVEC is designed to meet the intent of California's 2008 Title 24 requirements for residential ventilation, taking into account outdoor conditions, other ventilation devices (including economizers), peak demand concerns, and occupant preferences. RIVEC is also designed to manage all compliant residential ventilation systems that the California Energy Commission reviewed in developing the Title 24 requirements. The prototype controller was built to all of these specifications, bench tested, and then field tested in an occupied house in Moraga, California during late summer, 2008. The field testing had three weeks of operation using the RIVEC system, six days where the whole house system was forced to be off and 2 days where the whole house system operated 100% of the time with no RIVEC.

The test house was chosen because it had a mechanical ventilation system and an economizer in addition to the usual household appliances. The operation of the whole house mechanical ventilation system and other mechanical ventilation devices (the economizer and bathroom/kitchen fans) were recorded. The airflows through these mechanical ventilation devices were measured using an active flow hood and bag filling techniques (Walker and Wray (2003) and Walker et al. (2001)). The combination of operating time and airflows were used to determine the relative exposure and dose using Equations 5 and 8. Figure 3 shows the RIVEC and economizer operation and the computed relative exposure and dose with an assumed constant source. Table 4 summarizes the measured field test results. Because economizer operation was the biggest factor in the low dose and exposure value results, the results are presented for all data as well as for times when the economizer had been off for at least 6 hours. The results show that RIVEC significantly reduces overall ventilation fan requirements, especially when the economizer is operating. In this case the ventilation fan only operated 45% of the year. RIVEC kept the average relative dose and exposure well below unity even when the economizer had been off for 6 hours, which indicates that use of the controller did not compromise indoor air quality.

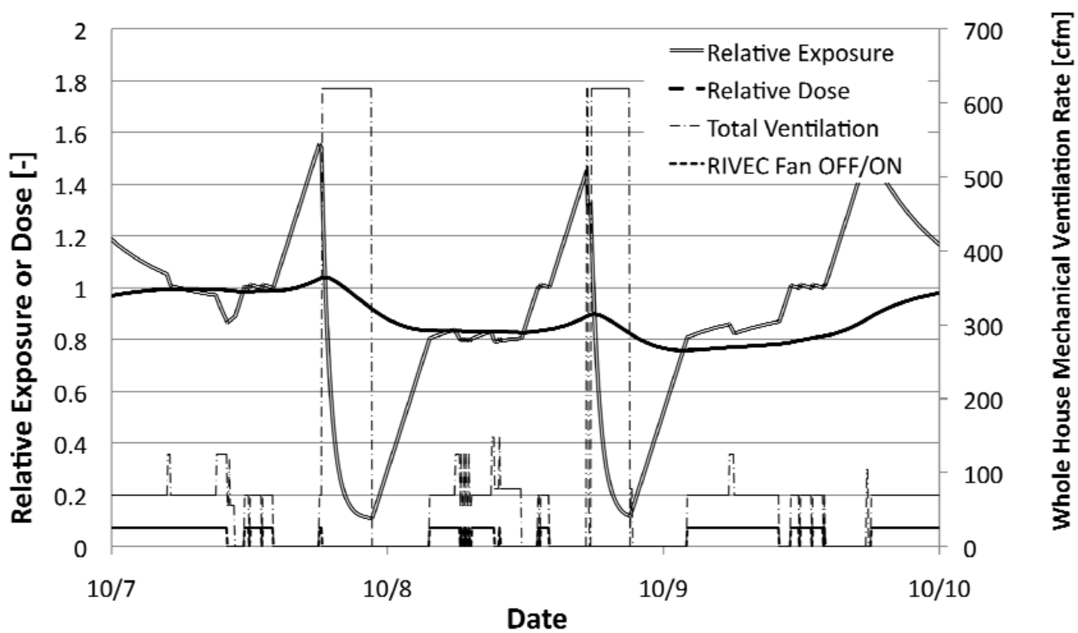

Figure 3. Field Measurements of RIVEC control and economizer operation observed in field tests in Moraga house combined with calculated relative exposure and dose

The energy savings were estimated using simulations because of large difference in weather for the field testing of RIVEC and non-RIVEC (i.e., continuous operation) operating times, which made direct energy comparisons meaningless. As with all relatively short-term field data, the degree to which these results are generally applicable is limited because of weather, specific occupancy patterns, and other normal variations. To overcome those limitations, we used the specific information about the test house, including diagnostic and occupancy pattern data, to make a customized model of the test site. The customized model was used to simulate RIVEC performance over a typical year.

Table 4. Measured RIVEC field test results

	All data			No Economizer for 6 hours		
	RIVEC control	**Always ON**	**Always OFF**	**RIVEC control**	**Always ON**	**Always OFF**
Total Airflow (cfm)	124	282	88	64	92	48
Average Relative Exposure	0.845	0.403	1.353	0.993	0.652	1.518
Average Relative Dose	0.802	0.779	1.258	0.853	0.698	1.215
Outdoor Temperature	60.6	70.4	63.9	60.1	82.3	65.2
Fractional ventilation fan operation	45	100	0	58	100	0

Economizer operation dominates the ventilation of the Moraga home during most of the cooling season. The economizer's large airflows for several hours at a time, which are repeated on an almost daily schedule, result in rapid relative exposure reductions and low relative doses. Figure 4 illustrates the simulated RIVEC-controlled fan operation for two typical summer days with economizer operation. Because economizer operation results in very high ventilation rates with correspondingly low exposure and dose, RIVEC turns off the controlled fan for long periods of the day.

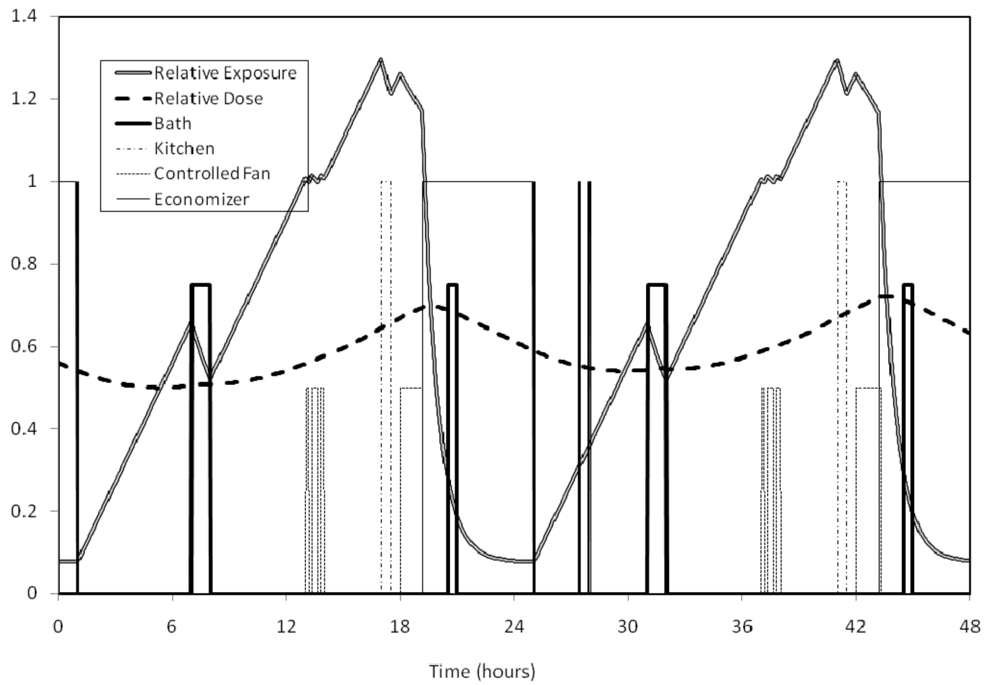

Figure 4. Simulated relative dose and exposure for the Moraga house, with fan schedule showing RIVEC operation for cooling with economizer during two days of the cooling season

The economizer does not operate during the heating season when relative exposure and dose averaged close to unity over a multi-day period, with whole-house ventilation controlled by RIVEC and diurnal cycles corresponding to the operation of kitchen and bath fans as shown in Figure 5.

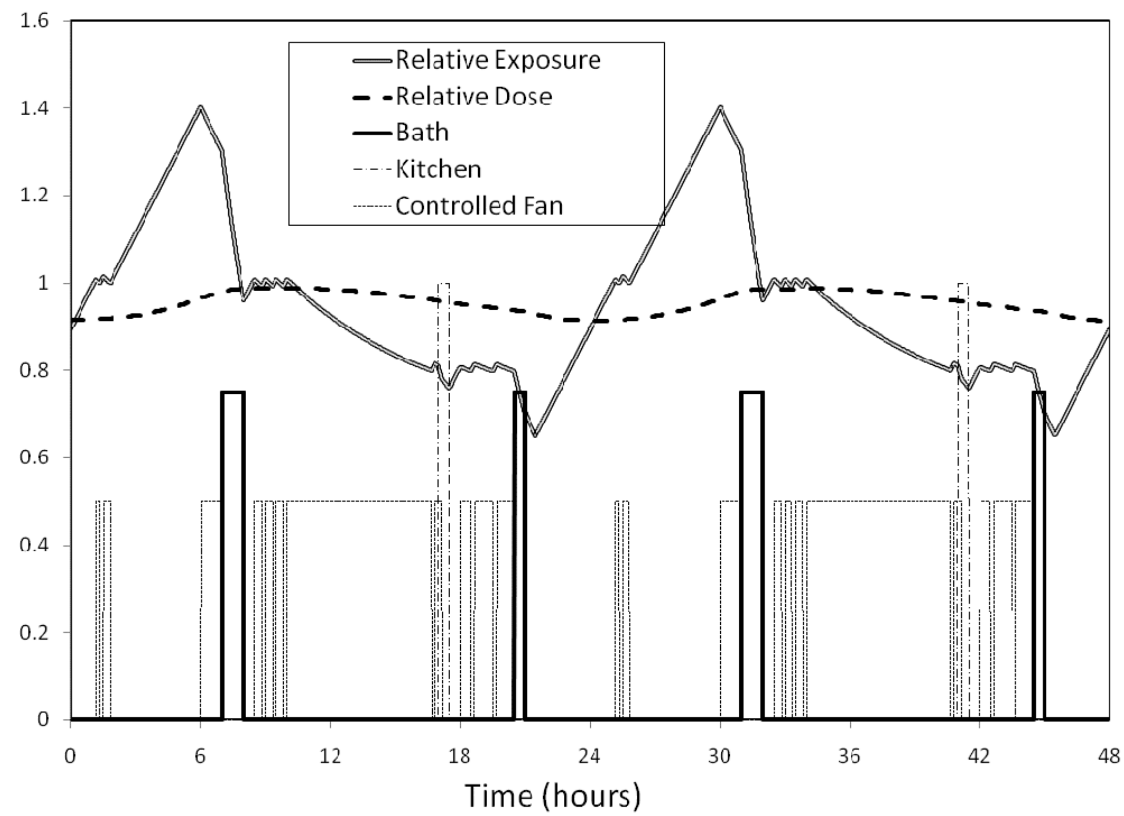

Figure 5. Simulated relative dose and exposure for the Moraga house with fan schedule showing RIVEC operation during two days of the heating season

The simulated energy savings with RIVEC totaled approximately 1,000 kilowatt hours (kWh) (255 kWh of electricity with the remainder in natural gas) for the Moraga field test home. If the observed operation of fans and economizers is extrapolated to a full year, RIVEC reduces the run time of the mechanical ventilation system by 71%, including a 100% reduction during the daily four-hour peak period when the system is shut off.

SUMMARY, CONCLUSIONS, AND RECOMMENDATIONS

We have demonstrated that it is possible to use a dynamic control strategy for residential mechanical ventilation that provides the indoor air quality benefits intended by the ASHRAE standard at substantially reduced energy costs (typically between 20% and 40%). The savings depend on the HVAC system and how occupants use other ventilation-related devices such as exhaust fans, clothes dryers, and economizers. The energy savings also depend on climate - with larger savings in more extreme climates.

Dynamic control can also successfully defer mechanical ventilation to avoid times of peak outdoor contamination or peak electricity demand. The dynamic controller used for our field demonstration, RIVEC, shut off the whole-house mechanical ventilation for a four-hour period daily (to account for peak loads) but was still able to provide contaminant exposure better than or equivalent to that offered by a continuous operation ventilation system.

Dynamic control can efficiently manage oversized mechanical ventilation systems by cycling them to achieve equivalent dose. In the example in this study, dynamic control was simulated for an oversized Heat Recovery Ventilator. With dynamic control, the HRV only needed to run 30% of the time overall to provide the necessary ventilation.

The dynamic control approach discussed in this article was demonstrated as a stand-alone device. However, this approach could be integrated into devices such as home automation systems or smart thermostats.

Undemonstrated Capabilities

The demonstration above was for a continuously occupied home with constant emission rates and a clock-driven notch ventilation pattern that could respond to exogenous ventilation. However, because the approach defined here is based on relative exposure and dose, it could easily be expanded to include unoccupied periods, variable emission sources, measured indoor and/or outdoor contaminant levels, smart electrical grids, and other variables. If, for example, the dynamic controller was coupled with an occupancy sensor, contaminant levels could be allowed to rise during periods when the building was unoccupied to minimize the need for ventilation and associated conditioning, which could save additional energy. The controller could also be programmed to provide demand response by, for example, responding to a utility signal indicating high grid demand; on receipt of the signal, the controller could defer ventilation to the extent that this did not compromise indoor air quality. These types of extensions to the system are currently being investigated.

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