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EVALUATION OF EXISTING TECHNOLOGIES FOR MEETING RESIDENTIAL VENTILATION REQUIREMENTS

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Preface

The California Energy Commission's Public Interest Energy Research (PIER) Program supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable, and reliable energy services and products to the marketplace.

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Evaluation of Existing Technologies for Meeting Residential Ventilation Requirements is the final report for the Residential Ventilation Requirements and Distribution System Research for 2008 Building Efficiency Standards, Task 3.2, project (contract number 500-04-005) conducted by Lawrence Berkeley National Laboratory. The information from this project contributes to PIER's Buildings End-Use Efficiency Program.

For more information about the PIER Program, please visit the Energy Commission's website at www.energy.ca.gov/pier or contact the Energy Commission at 916-327-1551.

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Abstract

The California Energy Commission is considering a modification to the state energy code, known as Title 24, to require mechanical ventilation based on the requirements of ASHRAE Standard 62.2-2004, *Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings*. These requirements will require installation of mechanical ventilation systems in nearly all new homes but allow for a variety of design solutions. These solutions, however, may have different energy costs and non-energy benefits. The authors have used a detailed simulation model to evaluate the energy effects of common and proposed mechanical ventilation approaches for a variety of climates. These results separate the energy needed to ventilate from the energy needed to condition the ventilation air and the energy needed to distribute and/or temper the ventilation air. The results show that exhaust systems are generally the most energy-efficient method to meet the proposed requirements, but that supply and balanced systems can provide additional non-energy benefits.

Keywords: ventilation, ASHRAE 62.2, indoor air quality, mechanical ventilation, infiltration, energy use, heat recovery, distribution

Executive Summary

Introduction

Home ventilation provides fresh - or at least outdoor - air to increase comfort and ensure healthy indoor air quality by diluting contaminants. In older California homes, natural air leakage and window opening generally provide enough ventilation to meet health and comfort needs, as well as the minimum ventilation requirements of the current California energy code (*2005 Building Energy Efficiency Standards for Residential and Non-Residential Buildings*, often referred to as Title 24). However, new tighter, energy-efficient homes may experience infiltration rates three to four times lower on average than those of existing homes. As a result, new homes often need mechanical ventilation systems to meet current ventilation standards.

California is therefore considering adding mechanical ventilation requirements to the next version of Title 24, due to take effect in 2008. Specifically, California is proposing a requirement for mechanical ventilation that meets the nationally recognized American Society of Heating Refrigerating and Air-Conditioning Engineers Standard 62.2 and mandates an extra 25 cubic feet per minute (cfm) of capacity to ensure ventilation even if mechanical systems are periodically turned off. To evaluate this proposal, the California Energy Commission needed more information on the performance and energy impacts - including energy impacts during hours of peak electricity use - of various mechanical ventilation technologies.

Purpose

This project evaluated the performance of existing residential mechanical ventilation technologies being considered to meet the proposed Title 24 residential ventilation requirements and investigated the energy, peak demand, and indoor air quality impacts of these technologies.

Objectives

- Evaluate the performance of various mechanical ventilation technologies.
- Evaluate the energy and peak energy impacts of various mechanical ventilation technologies.
- Evaluate the indoor air quality and comfort impacts of various mechanical ventilation technologies.
- Identify how current Title 24 energy-use methods and compliance tools in use by the California building industry will need to change to support the proposed mechanical ventilation requirements.
- Interact with software developers who provide compliance tools for the California Building Energy Efficiency Standards.

Approach

This study used simulations to examine the effect of different ventilation strategies on energy use in California houses. The houses simulated complied with Title 24, and the simulations focused on ventilation technologies complaint with American Society of Heating Refrigerating and Air-Conditioning Engineers Standard 62.2:

- Continuous exhaust
- Intermittent exhaust (operating for 20 out of 24 hours per day)
- Heat recovery ventilator
- Continuous exhaust augmented with a central fan-integrated supply
- Continuous supply with dedicated air distribution

Extra simulations were performed for some mechanical ventilation systems commonly used in California that do not meet American Society of Heating Refrigerating and Air-Conditioning Engineers Standard 62.2 minimum requirements:

- An unvented house with no mechanical ventilation, typical of new California construction
- A vented house assuming that windows are opened to maintain a minimum ventilation rate using the additional 0.35 air changes per hour currently used in the Alternative Calculation Method (a method of determining building code compliance in California)
- Central fan-integrated systems at 7 percent of fan flow for 10 out of every 30 minutes
- Central fan-integrated systems at one-third of the American Society of Heating Refrigerating and Air-Conditioning Engineers Standard 62.2 required air flow rate of 10 out of every 30 minutes

Energy use in the simulations was calculated in time-dependant value units, which account for variations in energy cost related to time of day, seasons, geography, and fuel type.

Project Outcomes

Energy Use Impacts

- American Society of Heating Refrigerating and Air-Conditioning Engineers Standard 62.2 - compliant ventilation systems add significantly to ventilation rates and reduce indoor pollutant concentrations but use more energy than does the unvented house.
- The extra energy required for ventilation was dominated by natural gas use for heating in most climate zones. The exception was the desert region of Southern California (Climate Zone 15), where more electric energy was used for cooling.
- A minimally American Society of Heating Refrigerating and Air-Conditioning Engineers Standard 62.2 - compliant continuous exhaust system uses about 10 percent more time-dependant value energy than does the unvented house. For comparison, 10 percent time-dependant value is roughly equivalent to the energy needed to meet the additional 0.35 air changes per hour ventilation currently required in Title 24 to offset low infiltration rates.

- The ventilation fan power requirements for continuous exhaust fans were equivalent to half of the energy required to condition the ventilation air.
- Relative to this continuous exhaust system, the intermittent exhaust and heat recovery ventilator systems reduced time-dependant value energy use by 1 percent to 5 percent, respectively.
- Heat recovery ventilator energy use was dominated by the energy needed to operate the heat recovery ventilator fans. Because heat recovery ventilator benefits are greatest when there are large indoor-outdoor temperature differences, year-round operation, including when temperature differences are small, results in the fan energy offsetting the space conditioning energy savings.
- Relative to this continuous exhaust system, the central fan–integrated and continuous supply systems used on average 22 percent and 13 percent more energy, respectively.
- Exhaust systems were found to be the most energy efficient of all the systems simulated, mostly because they had a lower impact on the total air exchange, reducing space conditioning needs accordingly. In addition, they also had the lowest fan energy consumption.

Peak Energy Use Impacts

- Air flows generated by operating a typical nighttime economizer for 6–8 hours can provide adequate ventilation for 10–12 subsequent hours without additional mechanical ventilation. This strategy could therefore reduce home mechanical ventilation and associated air conditioning and fan power electricity consumption throughout most of the day—including the afternoon peak electricity periods.

Health and Comfort Impacts

- Central fan–integrated systems provide distribution and mixing of air—providing an extra service beyond the basic requirements of American Society of Heating Refrigerating and Air-Conditioning Engineers Standard 62.2. In developing code requirements, the Energy Commission must decide how to account for this benefit. It could be argued that the energy of operating the central fan–integrated systems should not be included in the ventilation estimates because the central fan–integrated system is providing additional service.
- Intermittent ventilation systems and strategies can be used to significantly reduce the effects of outdoor air pollutants. Intermittent exhaust can reduce the ozone delivered to the house by 50 percent at peak outdoor ozone concentration and still maintain the ventilation rates required by the proposed standard.
- Compared to other systems, the dedicated supply ventilation system is the most energy-efficient way to distribute ventilation air throughout the house. If distributing air for comfort or filtration is important, then one of the compliant central fan–integrated supply systems should be selected. The energy used to operate central fan–integrated systems could be reduced by employing variable speed brushless permanent magnet motors. These motors were not considered in this study because of the high cost associated with the premium systems containing these motors and their low market penetration.

Conclusions

This project provided a wealth of information to help ensure that the revision of Title 24 meets the goals of ensuring comfort and indoor air quality, while minimizing any energy penalty. All the American Society of Heating Refrigerating and Air-Conditioning Engineers Standard 62.2–compliant technologies studied here are recommended for use in California with the following caveats:

- Intermittent exhaust allows flexibility of operation, energy savings, and the ability to reduce the effects of outdoor pollutants but must still be sized and operated to meet American Society of Heating Refrigerating and Air-Conditioning Engineers Standard 62.2.
- Heat recovery ventilator energy use could be optimized by ensuring that air flow rates match American Society of Heating Refrigerating and Air-Conditioning Engineers Standard 62.2 minimum requirements. Because most heat recovery ventilator systems provide more air flow than required, intermittent operation meets the requirements. Thus, a heat recovery ventilator with three times the American Society of Heating Refrigerating and Air-Conditioning Engineers Standard 62.2 minimum air flow rate could operate for only 20 minutes out of each hour.
- Supply systems (either dedicated continuous supplies or intermittent central fan–integrated systems) move much more air than do other systems to temper the incoming air as required for comfort, and thus require more fan power. Some specification of this fan power requirement should be incorporated in Title 24 compliance calculations.
- Significant ventilation credit can be given in the new revision of Title 24 for systems that produce large ventilation air flows because such systems can reduce overall ventilation energy use and peak energy use.

Recommendations

During the course of this project, the following research needs were identified for ventilation and energy use in California homes:

- A broad-based field study to determine envelope and duct air leakage of current new construction and how commonly used mechanical ventilation systems perform with such leakage.
- A study to determine how contaminants are transported from garages and other buffer zones and to determine if carbon monoxide alarms are necessary.
- Evaluation of the need to have air distribution systems to provide acceptable indoor air quality, including an examination of the performance of variable speed fans that can provide low flows with reduced energy consumption.
- Testing that would allow additional ventilation credit to be given in Title 24 economizers, direct evaporative coolers, and other systems that provide ventilative cooling.
- Evaluation of variable speed blower motors for ventilation air distribution.

Benefits to California

This work produced information that can help the State of California craft mechanical ventilation requirements that will protect health and comfort at the lowest energy and cost penalty. Notably, it revealed that all of the American Society of Heating Refrigerating and Air-Conditioning Engineers Standard 62.2–complaint systems studied here are suitable for use in California, with specific caveats.

Further, the study developed findings that can help achieve specific goals:

- Use of more efficient fans could decrease the electricity energy needed to provide mechanical ventilation.
- Use of intermittent exhaust could reduce the amount of the undesirable pollutant ozone entering into homes by as much as 50 percent during times of peak outdoor ozone concentration and still maintain healthy ventilation rates.
- Use of a nighttime economizer to increase airflow could reduce mechanical ventilation during the day, including peak electricity periods.

1.0 Introduction

1.1. Background

Because it affects, health, comfort, and serviceability, indoor air quality (IAQ) in homes is an increasing concern to many people. According to the American Lung Association, many factors within homes are increasingly recognized as threats to respiratory health, and the Environmental Protection Agency lists poor IAQ as the fourth largest environmental threat to our country. Asthma is the leading serious chronic illness of children in the United States, and construction defect litigation and damage are on the increase in new houses, with some of this increase related to such IAZ problems as moisture.

Residential ventilation can address many IAQ problems. Ventilation provides fresh (or at least outdoor) air to increase occupant comfort and ensure healthy IAQ by diluting contaminants. Historically, people have ventilated buildings to provide source control for both combustion byproducts and objectionable odors (Sherman 2004). A wide range of ventilation technologies are available to ventilate dwellings, including both mechanical systems and sustainable technologies. Most of the existing U.S. housing stock uses infiltration combined with window opening to provide ventilation, sometimes resulting in over-ventilation with subsequent energy loss and sometimes resulting in under-ventilation and poor indoor air quality (IAQ).

Traditionally, residential ventilation was not a major concern because policy makers believed that between operable windows and envelope leakage, people were getting enough outdoor air. However, in the three decades since the first oil shock, houses have become much more energy efficient. Based on previous work (Sherman and Dickerhoff 1994), a 2002 study (Sherman and Matson 2002) showed that recent residential construction has resulted in tighter, energy-saving building envelopes that create a potential for under-ventilation. Infiltration rates in these new homes are on average 3 to 4 times lower than rates in existing stock. As a result, new homes often need mechanical ventilation systems to meet current state and federal ventilation standards. At the same time, the types of materials used in furniture, appliances, and building materials in houses have changed—in some cases introducing new pollutants into homes. People have also become more environmentally conscious about the resources they are consuming and the environment they inhabit.

All of these factors have contributed to an increasing level of public concern about residential IAQ and ventilation, leading to a desire to define levels of acceptability and performance. Because states and other jurisdictions are responsible for protecting the health and safety of their populace, more and more of them are considering changes to their codes and regulations.

1.2. Ventilation in California's Energy Code Revision

The State of California has always included minimum ventilation requirements in its energy code (currently, the *2005 Building Energy Efficiency Standards for Residential and Non-Residential Buildings*, referred to as Title 24) to protect the health and safety of occupants. In the current and previous versions of this code, minimum ventilation in homes could be met by a combination of infiltration and natural ventilation (such as window opening).

For the next round of that code, due to take effect in 2008, California is considering mechanical ventilation requirements. The proposed requirements calls for mechanical ventilation that meets the American Society of Heating Refrigerating and Air-Conditioning Engineers (ASHRAE) Standard 62.2 plus an extra 25 cubic feet per minute (cfm) of capacity to allow for periodic turning off of the system (whether as part of a controlled ventilation system or by occupant intervention). The ASHRAE 62.2 minimum mechanical ventilation is sized as follows:

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

where N is the number of bedrooms in the house.

Ventilation can consume energy while providing acceptable indoor air quality. There is no fixed "right" amount of ventilation in the same sense that there is no fixed "right" size of furnace or air conditioner for all houses and climates. To find the minimum requirements for thermal conditioning, one must determine the thermal load and the desired thermal conditions to be met. Similarly, to determine how much ventilation is necessary, one must look at the sources and emissions of concern to indoor air quality (e.g., moisture levels and pollutant loads) and the desired level of indoor air quality. Based on these two factors, the choice of a ventilation level must be made by trading off various costs (e.g., energy costs, first cost, risks) with the benefits associated with the building service (e.g., health and comfort).

This said, determining appropriate minimum ventilation rates is much more complex than determining thermal insulation levels. Thermal loads are well studied and can be robustly estimated from internal gains and weather conditions. In contrast, pollutant sources tend to vary considerably among different households and are quite dynamic. Thermal comfort can be predicted quite reliably from just a few environmental parameters (e.g., temperature, air speed, humidity) whereas acceptable indoor air quality depends on numerous environmental and exposure parameters, many without established value ranges or acceptability criteria.

Because of the complexities of IAQ, ventilation standards and guidelines have followed a much more subjective route than have thermal standards and guidelines. Extant ventilation requirements are based on evaluations of what has or has not worked in the past and thus incorporate the experience of experts in the field. As the first and only American National Standards Institute– (ANSI-) certified residential ventilation standard in the country, ASHRAE Standard 62.2-2004 (ASHRAE 2004), summarized below, serves as the starting point for this project to evaluate currently available technologies and their energy impacts on meeting the proposed requirements.

1.3. Overview of ASHRAE Standard 62.2-2004

In developing this standard, ASHRAE recognized that there were many different kinds of houses, many different climates, and many different construction methods. To accommodate these differences, the major requirements were designed with several alternate paths to allow flexibility. Some requirements are performance based, with specific prescriptive alternatives.

The standard recognizes that there are several different ways to achieve a specified ventilation rate and allows both mechanical and natural methods.

There are three primary sets of requirements in the standard and a host of secondary ones. The three primary sets are as follows:

- Whole-house ventilation, to dilute the contaminant emissions from people, materials, and background processes
- Local exhaust, to remove contaminants from specific rooms where production of sources is expected (primarily kitchens and bathrooms)
- Source control, to mitigate sources that can reasonably be anticipated and controlled

The secondary requirements focus on properties of specific items—sound and flow ratings for fans and labeling requirements for controls—needed to achieve the main objectives of the standard. Some of the secondary requirements, as well as the guidance in the appendices, help keep the building system design from failing because ventilation systems were installed. For example, ventilation systems that excessively push moist air into the building envelope can lead to material damage unless the design of the envelope is moisture tolerant.

A summary of the ASHRAE 62.2 requirements for whole-house ventilation, local exhaust ventilation, source control, and rating follows:

- The whole-house ventilation rate is 3 cfm/100 square feet (ft^2) plus 7.5 cfm/person. It is assumed that 2 cfm/100 ft^2 can be supplied through infiltration. For most of the country, the difference must be supplied by mechanical ventilation. For most of California operable windows may be used because the climate meets infiltration degree day requirement (a requirement that accounts for the relationship between climate and infiltration) of less than 4500°F-day. Also, the local authority having jurisdiction may determine that window operation is a locally permissible method of providing ventilation.
- Local mechanical exhaust is required in kitchens and bathrooms. Kitchens must have the capacity to exhaust at least 100 cfm through a range hood or provide 5 kitchen air changes per hour (ACH). Bathrooms must have the capacity to exhaust 50 cfm or have 20 cfm of exhaust continuously.
- There are numerous requirements for source control: dryers must be vented to outdoors; naturally aspirated combustion appliances may not be inside under some conditions; filtration is required on air handling systems; and leaky ductwork is not permitted in garages, to name a few.
- Air moving equipment must be third party–rated and certified to meet its intended use, as well as meet certain noise and airflow specifications.

1.4. California Energy and Ventilation Concerns

This study was part of a project to provide input to the California Energy Commission (Energy Commission) to support its efforts to update the state's energy code. A recent study

(McWilliams and Sherman 2005) examined the relationship of ventilation to health by comparing ASHRAE/ANSI 62.2-2004 to the current Title 24 requirements and other relevant codes. The current study identified needed changes to the current Title 24 energy use algorithms and compliance tools applied by the California building industry to support the proposed ventilation requirements. The project team collaborated with software developers who provide compliance tools for the Title 24 Standards to provide these new algorithms.

Title 24 focuses on cost-effective ways to minimize the energy-related impacts of providing building services, such as thermal conditioning, lighting and water heating. Ventilation provides another such energy consuming service, ensuring acceptable IAQ. The building ventilation requirements in the current Title 24 standard are primarily engineering based and, as a result, technical feasibility is likely to remain a key driver for Title 24 provisions. This project evaluated existing ventilation strategies and technologies for their use in meeting the proposed Tile 24 ventilation requirements.

Specifically, the project examined the following engineering issues related to providing acceptable indoor air quality without incurring large energy penalties:

- The applicability of occupant control, particularly window opening
- Distribution and mixing of fresh air throughout a house
- The role of unusual sources and source control
- The role of air cleaning and particle filtration
- Effects of poor outdoor air quality (e.g., particulates near busy roads or in rural communities)

The current version of Title 24 has ventilation requirements that go beyond what most states require. Other states including Florida, Minnesota and Washington have also adopted minimum ventilation requirements. Other countries, such as Canada, France, Sweden, the United Kingdom and Denmark, also have specific ventilation requirements. To evaluate changes to Title 24, it is important to look at what other codes have specified, as well as what are considered best professional practice (ASHRAE 62.2).

1.5. Overview of Residential Ventilation Requirements in Title 24

Mechanical ventilation is required by Title 24 only if the building has a specific leakage area (SLA) less than 3.0, which corresponds (in typical California climates) to approximately 0.26 ACH provided by infiltration. Of course, the actual ventilation provided by infiltration would vary with seasonal temperature differences and local wind conditions. When mechanical ventilation is specified, Title 24 requires installation of a whole house ventilation system with the capacity to provide 0.047 cfm/ft^2.

If the SLA is less than 1.5, Title 24 requires provision of supply ventilation with enough capacity to maintain a house pressure greater than −5 Pascals (Pa) relative to outside when the other exhaust fans are running. This requirement is intended to avoid potential backdraft problems with combustion appliances, particularly fireplaces, and provide some protection in case of

future installation of combustion appliances. The standard also requires an air inlet and glass doors for fireplaces and wood, pellet, and gas stoves.

When performing energy calculations, Title 24 also assumes that windows will be opened by the occupants whenever the ventilation rate drops below 0.35 ACH.

Title 24 also includes provisions for ventilative cooling, a form of natural ventilation. Title 24 assumes that under certain circumstances, occupants will open their windows to maximize free cooling. This ventilative cooling reduces air conditioning and exhausts indoor contaminants.

1.6. Next Generation of Residential Ventilation in Title 24

Several changes have been recommended for Title 24 and form the basis for the simulations conducted in this project. These recommendations fall into two groups: those for IAQ-related requirements and those for energy-related requirements, as shown below.

1.6.1. IAQ-Related Proposed Recommendations

- ASHRAE Standard 62.2-2004 should be adopted by reference except where overridden by any of the other recommendations contained in this report.
- Air leakage and window operation shall be deemed to provide the default infiltration of 62.2 and also meet the window opening requirements of the standard, but may not be used to otherwise meet the whole-building ventilation or local exhaust provisions of the standard.
- The minimum mechanical ventilation rates of the standard shall be increased by 25 cfm.
- When ducts pass through buffer zones (e.g., a garage, attic or crawlspace), the total leakage shall be limited to 5% of the central forced air blower air flow.

1.6.2. Energy-Related Proposed Recommendations

- SLA=4 shall be the default value for use in energy calculations, but measured air tightness may be used if known. Current restrictions on tightness below SLA=1.5 should be removed.
- Designs with suitable control systems (e.g., a programmable timer) can take energy credit for turning off the ventilation system up to four hours per day if the four hours chosen are times of peak indoor-outdoor temperature difference (in this study, peak was 3–7 p.m. for cooling and 1–5:00 a.m. for heating).

1.7. Project Objectives

To summarize, the objectives of this project were to evaluate the performance of existing residential ventilation technologies for meeting the proposed Title 24 residential ventilation requirements and to investigate the energy and peak demand impacts of the various technologies.

2.0 Project Approach

To evaluate the energy impacts of the proposed requirements, an extensive simulation protocol was executed using the REGCAP simulation tool. REGCAP has been used in several previous studies of heating, ventilating, and air conditioning (HVAC) system performance[1]. REGCAP includes a detailed air flow network model that calculates the air flow through building components as they change with weather conditions and HVAC system operation. The pressure difference and airflow calculations include the effects of weather, leak location, and HVAC system flows on house and attic air pressures. These dynamic air pressure and air flow interactions are particularly important because the air flows associated with ventilation systems (including duct leakage) significantly affect natural infiltration in houses.

In this project, REGCAP performed minute-by-minute ventilation, heat, and moisture calculations to estimate the dynamic performance of buildings and HVAC components. The small timesteps, while computationally and analytically intensive, allowed for direct simulation of temporally complex ventilation controls. The simulations, performed for a full year, examined approximately 100 different combinations of house size, climate, and ventilation technologies.

The following sections contain details on the simulation plan—which was reviewed by Energy Commission staff and the Project Advisory Committee (PAC). To determine the energy used to provide mechanical ventilation, the 2005 *Certified Home Ventilation Products Directory* (Home Ventilating Institute [HVI] 2005) was used to obtain fan power for fans that met the air flow requirements proposed for Title 24 and the sound requirements of ASHRAE Standard 62.2. The specific fan manufacturers and model numbers are given in brackets [ ] for each system in the plan below.

More details of the tool and of the simulations can be found in Appendix A.

2.1. Houses Simulated

Three house sizes were simulated to examine the implicit effect of occupant density in the ASHRAE 62.2 requirements:

- Small (1000 ft^2) one-story two-bedroom house
- Mid-size (1761 ft^2) two-story, three-bedroom house[2]
- Large (4000 ft^2) two-story, five-bedroom house.

Because the number of occupants does not scale with the size of a house, larger houses tend to have lower occupant densities. For most of the mechanical ventilation simulations, the medium-sized house was used, and for selected cases, the smaller and larger houses were used.

[1] See REGCAP Bibliography at the end of the Appendix.

[2] Based on the 2005 T24 ACM prototype C

Envelope leakage for each house was fixed at an SLA of 4. This figure was recommended by the PAC as a reasonable value for new California construction. The corresponding leakage values are summarized in Table 1.

The house and duct insulation used to determine the non-ventilation building load, and duct system performance varied by climate as shown in Table 2[3]. The insulation was degraded according to the *Residential Alternative Calculation Method (ACM) Approval Manual for the 2005 Building Energy Efficiency Standards for California*[4].

The exterior surface area for wall insulation scaled with floor area and number of stories. Wall-to-floor area ratios and window -to-floor area ratios were developed from measured data from several thousand new homes[5] and from the simplified box prototype C in the ACM manual. The wall area was assumed to be 1.54 times the floor area for a two-story home and 1.22 times the floor area for a one-story home. Window area was 20% of floor area with windows equally distributed on the four exterior walls. The solar heat gain coefficient (SHGC) varied by climate zone between 0.4 and 0.65. Values specified in ACM Table 151-C, p.133 were used. In climate zones where a minimum SHGC was not required, ACM Tables 116-A and 116-B, p.56 were used. The required U-value (the measure of air-to-air heat transmission due to thermal conductance and the difference in indoor and outdoor temperatures) was taken from ACM Table 116-A, and the SHGC corresponding to the same window from table 116-B was used. Clear glazing was assumed as was an exterior shading of 50%.

Table 1. Envelope leakage

Floor Area (ft^2)	SLA	ELA4 (in^2)	m^3/(sPan)	cfm/Pan
1000	4	58	0.038	81
1761	4	101	0.067	143
4000	4	230	0.152	325

[3] Based on CA T24 2005 Package D requirements including degradation factors.

[4] California Energy Commission. 2005.

[5] Based on BSC/Building America data.

Table 2. House insulation levels

Climate Zone	Ceiling			Wall		Ducts outside conditioned space
		Heating Degraded	Cooling Degraded		Degraded	
1	R38	21.6	31.9	R21	17.6	R6
2	R30	18.8	26.1	R13	10.9	R6
3	R30	18.8	26.1	R13	10.9	R6
4	R30	18.8	26.1	R13	10.9	R6
5	R30	18.8	26.1	R13	10.9	R6
6	R30	18.8	26.1	R13	10.9	R4.2
7	R30	18.8	26.1	R13	10.9	R4.2
8	R30	18.8	26.1	R13	10.9	R4.2
9	R30	18.8	26.1	R13	10.9	R6
10	R30	18.8	26.1	R13	10.9	R6
11	R38	21.6	31.9	R19	10.9	R6
12	R38	21.6	31.9	R19	10.9	R6
13	R38	21.6	31.9	R19	10.9	R6
14	R38	21.6	31.9	R21	17.6	R8
15	R38	21.6	31.9	R21	17.6	R8
16	R38	21.6	31.9	R21	17.6	R8

2.2. Proposed Title 24 Requirements

The ASHRAE 62.2 required minimum mechanical ventilation air flows for the three house sizes simulate are as follows:

- 1000 ft^2 and 2 bedrooms (3 occupants) ⇒ 33 cfm
- 1761 ft^2 and 3 bedrooms (4 occupants) ⇒ 48 cfm
- 4000 ft2 and 5 bedrooms (6 occupants) ⇒ 85 cfm

Adding the extra 25 cfm—as proposed for the new California standards—results in:

- 1000 ft^2 and 2 bedrooms (3 occupants) ⇒ 58 cfm
- 1761 ft^2 and 3 bedrooms (4 occupants) ⇒ 73 cfm
- 4000 ft2 and 5 bedrooms (6 occupants) ⇒ 110 cfm

2.3. Intermittent Operation

Intermittent exhaust was simulated as a peak demand reduction technique (and possibly outdoor pollutant control). The system consisted of a bathroom fan that is on for 20 hours and off for 4 hours during peak (3–7 p.m. for cooling and 1–5:00 a.m. for heating). The relationships given in an *ASHRAE Transactions* article (Sherman 2005) and ASHRAE 62.2 show that intermittently under-ventilating for 4 hours out of 24 (given the background natural infiltration

and extra 25 cfm capacity of the continuous exhaust minimum flow required by 62.2) yielded acceptable, effective ventilation rates that met 62.2 requirements.

2.4. Additional ASHRAE 62.2 requirements

All the fans used to provide mechanical ventilation were selected to meet the sound and installation requirements of ASHRAE 62.2. Fans that meet the 1.0 sone (a unit of perceived loudness) requirement for continuous operation and 3 sones for intermittent operation tend to be energy efficient fans that also have power ratings in the HVI directory.

2.5. Weather

Weather data were taken from Title 24 compliance hourly data files converted to minute-by-minute format by linear interpolation. The simulations also used location data (altitude and latitude) in solar and air density calculations. The required weather data for the simulations were as follows::

- Direct solar radiation (watts per meter squared [W/m^2])
- Total horizontal solar radiation (W/m^2)
- Outdoor air dry-bulb temperature(°C)
- Outdoor air humidity ratio
- Wind speed (meters per second [m/s])
- Wind direction (degrees)
- Barometric pressure (kPa)
- Cloud cover index

2.6. Heating and Cooling Equipment

The simulations used the detailed equipment models discussed in more detail in Appendix A. Equipment sizing was based on a combination of Air Conditioning Contractors of America (ACCA) *Residential Load Calculation Procedure*—also know as Manual J—calculations and the results of the field survey of new California homes undertaken by Rick Chitwood[6]. Equipment sizing was most important for systems that use the central furnace blower to distribute ventilation air because the outside air is usually supplied as a fraction of total furnace blower flow and the energy used to distribute the air depends on the size of the blower motor (Appendix E summarizes the heating/cooling equipment capacities and associated blower power consumption). For all these simulations, the correct furnace blower flow and refrigerant charge were used, so air conditioner capacity and energy efficiency ratio (EER) only depended on the return air and outdoor air temperatures.

The heating was supplied by an 80% annual fuel utilization efficiency (AFUE) natural gas furnace. For cooling, a seasonal energy efficiency ratio (SEER) 13 split-system air conditioner with a thermostatic expansion valve (TXV) refrigerant flow control was used.

[6] PIER 08 *Residential Furnace blower Survey* – see results summary in Appendix C

The duct leakage to outside was 5%, split with 2.5% supply leakage and 2.5% return leakage for most of the simulations. A few cases were examined with higher duct leakage: 11% supply and 11% return[7].

Determination of heating or cooling operation was based on the Title 24 seven-day running average technique. When the seven-day running average outdoor temperature was greater than 60°F, cooling was assumed, and when it was less than 60°F, heating was assumed. However, in most climates this resulted in multiple switches between heating and cooling, which was unrealistic. Therefore, for each climate zone (CZ), one day was selected for the heating to cooling mode switch and one day for the cooling to heating mode switch, based on the seven-day running average technique. A list of the switching days is given in Table 3.

Table 3. Days to switch heating and cooling modes

CZ	Day of year to switch to cooling	Day of year to switch to heating
1	No cooling	Always in heating mode
2	134	289
3	152	283
4	152	284
5	185	286
6	144	310
7	115	310
8	108	313
9	112	313
10	113	313
11	117	282
12	117	278
13	103	300
14	133	289
15	64	317
16	160	247

Operation of the heating and cooling equipment used the following set-up and set-back thermostat settings taken from the ACM and shown in Table 4.

[7] Title 24 default for new construction

Table 4. Thermostat settings for ventilation simulations (°F)

Hour	Heating	Cooling
1	65	78
2	65	78
3	65	78
4	65	78
5	65	78
6	65	78
7	65	78
8	68	83
9	68	83
10	68	83
11	68	83
12	68	83
13	68	83
14	68	82
15	68	81
16	68	80
17	68	79
18	68	78
19	68	78
20	68	78
21	68	78
22	68	78
23	68	78
24	65	78

2.7. Simulated Ventilation Technologies

2.7.1. Case 1. Unvented House

This case represented a California home built to comply with 2005 Title 24 building and energy codes, but that did not comply with ASHRAE Standard 62.2 and did not have the ventilation adder used in Title 24 (this was considered in separate simulations). All 16 CZs were simulated for the medium house. The envelope leakage was the same as the mechanically ventilated homes.

2.7.2. Case 2. Continuous Exhaust

The air flow requirements were met using envelope infiltration and continuous exhaust through a bathroom fan. The medium sized house was simulated in 16 CZs. The small and large house were simulated in five CZs (3, 13, 16, 15, &10)—chosen because they contain the majority of new construction in the state.

The ASHRAE 62.2 requirements follow:

- 1000 ft^2 and 2 bedrooms (3 occupants) $\Rightarrow$ 58 cfm

- 1761 ft^2 and 3 bedrooms (4 occupants) ⇒ 73 cfm
- 4000 ft2 and 5 bedrooms (6 occupants) ⇒ 110 cfm

Using the nearest size greater than the minimum using specific directory entries gave the following for fan power use:

- 1000 ft^2 and 2 bedrooms (3 occupants) ⇒ 60 cfm [0.028 m^3/s] 13.7 W [Panasonic FV-05VQ2]
- 1761 ft^2 and 3 bedrooms (4 occupants) ⇒ 73 cfm [0.034 m^3/s] 20.1 W [Panasonic FV-08VQ2]
- 4000 ft2 and 5 bedrooms (6 occupants) ⇒ 60 cfm [0.028 m^3/s] 13.7 W [Panasonic FV-05VQ2] + 50 cfm [0.0236 m^3/s] 13.5 W [Panasonic FV05VF1] (Total of 27.2 W)

The baseline for comparing ventilation technologies was the medium-sized 1761 ft^2 house with continuously operating exhaust.

2.7.3. Case 3. Intermittent Exhaust

The intermittent exhaust operation outlined in Section 3.3 was used in simulations performed for all three house sizes in a heating dominated (CZ 16), in a cooling dominated (CZ 13) climate, and in a temperate climate (CZ3). The fan flow and power requirements were the same as for case 2. Note that the air flow rates and equipment for case 2 were used in this case because 25 cfm was already added to the ASHRAE 62.2 minimum for case 2.

2.7.4. Case 4 (and 4X). Heat Recovery Ventilator (HRV)

The HRVs were operated for half an hour then were turned off for half an hour. The HRV air flows in the HVI directory were much larger than the minimum AHSRAE 62.2 requirements. The selected HRV was one of the lowest air flow HRVs, with an air flow of 130 cfm. This was about 35% more flow than simply doubling the AHSRAE 62.2 minimum requirement of 48 cfm for this house that would be mandated for its 50% duty cycle.

An HRV was simulated for the medium-sized house in cold climates (CZ 16 and CZ 1). The HVI directory listed recovery efficiencies were applied to the air flow through the HRV when calculating the energy use. For these simulations, the apparent sensible effectiveness (ASE) was used to determine the temperature of air supplied to the space ($T_{tospace}$). It was assumed that the HRV has its own duct system that does not leak and is located entirely within the conditioned envelope of the house.

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

The HRV selected from the HVI directory [Broan Guardian HRV 100H] was assumed to be installed correctly and operating at the rated pressure drop. With these assumptions, this HRV uses 124 W at 138 cfm [0.0652 m^3/s] net airflow at the 0.44 inches of water [110 Pa] external static pressure of the standard HVI rating point and has:

- Apparent sensible effectiveness = 70%

- Sensible recovery efficiency = 62%

It was assumed that the supply and return fans used the same amount of power: 62 W. For the supply fan, 55 W of heat was added to the indoor air (based on 7 W of required air power).

An additional set of simulations was performed at a reduced operating schedule (35% less operating time) such that the mean ventilation rate was the same as the ASHRAE 62.2 minimum requirements. Two cases were developed to distinguish between the two HRV schedules:

- Case 4: A mean rate matching the ASHRAE 62.2 minimum
- Case 4X: 50% duty cycle

2.7.5. Case 5. Central Fan Integrated (CFI) Supply with Air Inlet in Return and Continuously Operating Exhaust

CFI and continuous exhaust was simulated for all three houses in CZs 3,13,16,15,and 10. The continuously operating exhaust performance was the same as case 2. This case was augmented with a CFI supply that uses the furnace blower to intentionally draw outdoor air through a duct into the return and distribute it throughout the house using the heating/cooling supply ducts. The outdoor air duct was only open to outdoors during furnace blower operation and has a damper that closes when the furnace blower is off. This damper was assumed to have zero leakage when closed.

The furnace fan power requirements were determined based on the space conditioning equipment capacity determined by Manual J load calculations and a nominal 2 cfm/W (been found to be typical in numerous field studies). Because the CFI systems used the forced air heating and cooling ducts, the same air leakage and heat transfer was applied to the ducts for CFI operation as for heating and cooling operation. For this study, it is assumed that ducts are in the attic. The waste heat from the furnace blower and heat exchange between the ducts and their surroundings were included in the calculations. The fraction of outside air (OA) entering the system is fixed so that it balances the exhaust flow and makes this system switch from exhaust ventilation to balanced ventilation. The CFI supply system operated for at least 20 minutes per hour if the heating and cooling systems operate for less than this time to satisfy thermostat calls for heating or cooling.

To examine sensitivity to duct leakage, the project team simulated the medium sized house in a heating dominated (CZ 16), cooling dominated (CZ 13) and temperate climate (CZ3) with 11% supply an 11% return leakage.

2.7.6. Case 6. Continuous Supply

The project team simulated the medium house in CZs 3, 13, 16, 15, and 10. The continuous supply system uses a fan to supply filtered air from outside that then distributes the air throughout the house without using the furnace blower or the forced air heating and cooling ducts. Therefore, the continuous supply air is not associated with any duct leakage or heat transfer effects. For continuous supply, the supply air is mixed with indoor air for tempering purposes. A mixing ratio of 3:1 was used for indoor to supply air. The supply fan will therefore be sized to be four times the case 2 requirements (292 cfm [0.138 m^3/s]) for the medium sized

house. A selected model [Greentek MTF 150P] provides this flow at a power consumption of 133 W, of which 14 W is air power and 119 W is heat.

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

2.7.7. Case 7. CFI with 7% OA, Without Continuous Exhaust – Not ASHRAE 62.2 Compliant

These simulations were performed for the medium house in CZs 3, 10, 13, 15 and 16. Unlike the case 5 simulations, there was no continuous exhaust. The CFI system was on for 10 minutes, then off for 20 minutes. This system did not account for furnace blower operation for heating or cooling; the OA supply duct was open and the blower was on for the first 10 minutes out of every 30 minutes regardless of the space conditioning system operating mode. Because the air flows from outside were limited by tempering issues, they are the same as for case 5. Due to reduced operating time, the net flows were therefore not ASHRAE 62.2 compliant. The OA flow was based on the total furnace blower flow and was set at 7% of fan flow. Because this was achieved by a fixed damper setting, rather than by damper modulation to achieve a fixed flow, this air flow was a fixed 7% of the furnace blower flow. These assumptions resulted in 7% of heating fan flow during heating, 7% of cooling fan flow during cooling, and 7% of cooling fan flow when ventilating only. A damper closed the OA vent when the CFI was not operating (i.e., for 20 minutes out of every 30 minutes).

2.7.8. Case 8. CFI with 1/3 of ASHRAE 62.2 Flow, Without Continuous Exhaust – Not ASHRAE 62.2 Compliant

These simulations were the same as case 7, but with the air flow adjusted to be the 62.2 air flow rate rather than 7% of blower flow. Because the CFI operated one third of the time it provided one third of the ASHRAE 62.2 required air flow.

2.7.9. Case 9. Minimum Ventilation from ACM

These simulations were for the unvented house of case 1, but with the minimum ventilation rate adder of 0.35 ACH used when air change rates fall below 0.35 ACH. This mimicked the ventilation added currently used in the Title 24 ACM.

2.7.10. Source Control Ventilation

In addition to incorporating specific technologies that meet ASHRAE 62.2, intermittent operation of kitchen and bathroom fans was included for source control.

Intermittent bathroom fans operated for half an hour every morning from 7:30 a.m. to 8:00 a.m. These bathroom fans were sized to meet the ASHRAE 62.2 requirements for intermittent bathroom fans. From Table 5.1 in ASHRAE 62.2, this is 50 cfm (25 liters per second [L/s]) per bathroom. For houses with multiple bathrooms, the bathroom fans operated at the same time, so the 1761 ft^2 house had a total of 100 cfm (50 L/s) and the 4000 ft^2 house had a total of 150 cfm (75 L/s). Power requirements for these fans were 0.9 cfm/W based on the Chitwood field survey data (i.e. 55W for each 50 cfm fan).

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

2.8. Ventilation Options Not Simulated

Two ventilation options that were initially considered for simulating were not simulated:

- *Open windows*. Based on recent survey results, this method of providing ventilation is not sufficiently reliable due to uncontrollable variations in occupant behavior.
- *Passive vents*. Although popular in Europe, this technology is not available in the California market.

This reduction in scope allowed for simulation of additional cases: indoor concentrations at low ventilation rates and the effects of the low ventilation rate added in Title 24 ACM (Case 9).

Other ventilation related options not included follow:

- *Complex control strategies for any kind of ventilation system*. Such strategies are generally not appropriate for a minimum performance standard, and it was too difficult to ensure that the actual operating characteristics are the same as those claimed in compliance calculations. They are also very complex to handle with compliance software.
- *Proprietary Systems, such as Nightbreeze*. Because proprietary control and operation algorithms were unavailable, proprietary systems were not simulated.

3.0 Results of Ventilation Simulations

The following results are for the medium house except where noted.

3.1. Air Change Rates

The air flows were converted into air change rates by dividing by the house volume. Mean annual air change rates were calculated for each simulation and are summarized in Table 5.

Effective air change rates were calculated using the Sherman and Wilson (Sherman and Wilson 1986) turnover time approach—which accounts for temporal variation in air change to calculate the effective air change that would give the same internal exposure to pollutants—and are summarized in Table 6. In general, these rates were lower than the mean air change rates.

However, as the efficacy (ratio of effective to mean ACH) values in Table 7 show, the mechanical ventilation systems were about 5% more effective than the unvented house. This greater effectiveness made the differences in air change rates larger between the unvented and mechanically vented house.

Significant results follow:

- In the unvented house (case 1), sensitivity to weather conditions resulted in a large range of average ventilation rates—from 0.19 to 0.32 ACH—depending on climate. This variability was mostly driven by cold winter weather, which results in higher stack pressures and envelope air flows. In contrast, use of mechanical systems results in less variation in mean air change rates from climate zone to climate zone. Further, all of the mean ventilation rates for the unvented house were lower than for any of the ASHRAE 62.2–compliant cases (cases 2, 3, 4, 4X, 5, and 6), which had mean air change rates of 0.35 or higher.
- Use of continuous exhaust increased the mean effective ventilation rate about 65% over that of the unvented house, with ACH increases ranging from 0.11 ACH in CZ16 to 0.18 ACH in CZ15. Such changes were smaller in houses in the colder climates, because these houses had more natural infiltration. For example, in CZ16, ACH rates were more than 0.5 ACH for the unvented house in the winter. In contrast, in CZ 8, the continuous exhaust increased ventilation rates by more than 85% over that of the unvented house, due to low natural infiltration driving forces, even in winter.
- Use of intermittent exhaust decreased the average ventilation rate about 7% compared to use of continuous exhaust, due to the 17% reduction in operating hours for the exhaust fan. However, the ventilation effectiveness of intermittent exhaust decreased by less than 1% (and is still high at around 97% to 98%). This high effectiveness was achieved because infiltration still occurred when the mechanical ventilation is off. This result indicates that the off period of four hours is not too long and the 24-hour cycling period is short enough.
- Use of HRV increased the average ventilation rate about 45% over use of continuous exhaust. This increase occurred because the HRV flow was about 35% higher than that

required to meet ASHRAE 62.2 even when HRV operated for only 30 minutes out of each hour. Reducing HRV runtime to 20 minutes out of each hour (only one-third the operating time of continuous exhaust) resulted in a mean effective ventilation rate of 5% less than that of continuous exhaust. This very small decrease was achieved because the balanced ventilation of the HRV added directly to natural infiltration, whereas the exhaust-only fan only added about half of its flow to the effective ventilation rate.

- Use of CFI supply with air inlet in return and continuously operating exhaust increased the average ventilation rate by 22% over continuous exhaust, indicating that the added OA supply through the return inlet (with same flow as exhaust) is effective at increasing effective ventilation. This result occurred because when both the CFI and exhaust were operating, the system was balanced. Unlike exhaust only systems, balanced systems add directly to the natural infiltration.
- Continuous supply increases the average ventilation rate by 17% over continuous exhaust, indicating the greater effectiveness of the supply fan. This result was due to a combination of two factors. First, there were periods of balanced mechanical ventilation when the kitchen and bath exhausts operate (when the exhaust ventilation fan provided even more unbalanced exhaust ventilation). Second, under normal natural ventilation conditions, the leakage distribution and wind and stack effect pressures tended to slightly depressurize the house, allowing supply systems to be slightly more effective in the their interaction with the building envelope.
- Two non AHSRAE 62.2–compliant technologies—CFI systems that operated for 20 minutes out of each hour—were evaluated because they are currently used as mechanical ventilation systems in new California houses. Although their air flow rates for OA were close to or equal to the ASHRAE 62.2–specified continuous air flow rates, their fractional runtime made them non ASHRAE–62.2 compliant.

  The first of these systems, case 7, had 7% OA. This system's average ventilation rate was higher than that of continuous exhaust in CZs 13, 15, and 16 (by 0.015 to 0.036 ACH) but lower in CZs 3 and 10 (by 0.02 and 0.7 ACH, respectively). Duct leakage during non-heating or cooling operation contributed significantly to the overall ventilation rate because it led to about 50 cfm of balanced leakage—or up to 0.2 ACH for CZ15. In CZs with higher furnace blower air flows, the 7% OA operating mode led to supply flow rates close to ASHRAE 62.2 requirements (e.g., 140 cfm in CZ15 compared to 150 cfm that would be required to meet 62.2 for one-third time operation). In addition, the supply air flow interacted with the other building leakage and envelope pressures, such that the total ventilation rate was higher than for an exhaust fan of the same air flow.

  The second system, case 8, had an outdoor air flow rate set equal to the minimum ASHRAE 62.2 air flow rate. This setting resulted in lower OA flows than for the 7% OA case. The resulting yearly average effective ventilation rates were less than that of continuous exhaust by 0.017 to 0.074 ACH, depending on climate.

Table 5. Mean annual air changes per hour for different simulation cases

	Simulation Cases										
CZ	**1 Unvented House - *not 62.2 compliant***	**2 Cont. Ex.**	**3 Int. Ex.**	**4 HRV 62.2 match air flow**	**4X HRV 50% ontime**	**5 CFI with Cont. Ex.**	**6 Cont. Supply**	**7 CFI 7% OA - *not 62.2 compliant***	**8 CFI 62.2 33% runtime - *not 62.2 compliant***	**9 Unvented House with 0.35 ACH adder - *not 62.2 compliant***	**Cont. Ex. with 0.35 ACH adder - *not 62.2 compliant***
1	0.26	0.38		0.36	0.55						
2	0.23	0.37									
3	0.24	0.37	0.35			0.44	0.46	0.30	0.31		
4	0.24	0.38									
5	0.23	0.37									
6	0.19	0.35									
7	0.21	0.36								0.53	0.63
8	0.19	0.36									
9	0.21	0.37									
10	0.20	0.37				0.45	0.42	0.35	0.29		
11	0.28	0.42									
12	0.27	0.40									
13	0.23	0.38	0.35			0.47	0.44	0.39	0.33		
14	0.26	0.41									
15	0.24	0.42	0.39			0.51	0.45	0.45	0.35	0.52	0.59
16	0.32	0.43	0.41	0.42	0.61	0.53	0.55	0.47	0.41	0.54	0.56

Table 6 Effective annual air changes per hour for different simulation cases

	Simulation										
CZ	**1 Unvented House - *not 62.2 compliant***	**2 Cont. Ex.**	**3 Int. Ex.**	**4 HRV 62.2 match air flow**	**4X HRV 50% ontime**	**5 CFI with Cont. Ex.**	**6 Cont. Supply**	**7 CFI 7% OA - *not 62.2 compliant***	**8 CFI 62.2 33% runtime - *not 62.2 compliant***	**9 Unvented House with 0.35 ACH adder - *not 62.2 compliant***	**Cont. Ex. with 0.35 ACH adder - *not 62.2 compliant***
1	0.24	0.37		0.34	0.54						
2	0.22	0.37									
3	0.22	0.36	0.34			0.43	0.45	0.29	0.30		
4	0.22	0.37									
5	0.22	0.36									
6	0.18	0.35									
7	0.20	0.35								0.52	0.63
8	0.18	0.35									
9	0.19	0.36									
10	0.19	0.36				0.44	0.40	0.34	0.28		
11	0.25	0.40									
12	0.24	0.38									
13	0.21	0.37	0.34			0.46	0.43	0.38	0.31		
14	0.24	0.39									
15	0.21	0.40	0.37			0.49	0.43	0.44	0.33	0.52	0.58
16	0.29	0.41	0.39	0.39	0.59	0.52	0.52	0.44	0.39	0.54	0.55

Table 7. Ventilation efficacy[8] by case

	Simulation Cases										
CZ	**1 Unvented House - *not 62.2 compliant***	**2 Cont. Ex.**	**3 Int. Ex.**	**4 HRV 62.2 match air flow**	**4X HRV 50% ontime**	**5 CFI with Cont. Ex.**	**6 Cont. Supply**	**7 CFI 7%OA - *not 62.2 compliant***	**8 CFI 62.2 33% runtime - *not 62.2 compliant***	**9 Unvented House with 0.35 ACH adder - *not 62.2 compliant***	**Cont. Ex. with 0.35 ACH adder - *not 62.2 compliant***
1	0.94	0.98		0.96	0.98						
2	0.94	0.98									
3	0.93	0.98	0.97			0.98	0.98	0.95	0.95		
4	0.94	0.98									
5	0.94	0.98									
6	0.95	0.98									
7	0.95	0.98								0.99	0.99
8	0.94	0.98									
9	0.94	0.98									
10	0.92	0.97				0.98	0.97	0.97	0.96		
11	0.90	0.96									
12	0.91	0.97									
13	0.92	0.97	0.98			0.98	0.97	0.97	0.96		
14	0.90	0.96									
15	0.90	0.95	0.96			0.97	0.96	0.97	0.94	0.99	0.98
16	0.89	0.96	0.96	0.93	0.96	0.97	0.95	0.95	0.94	0.99	0.98

[8] Ratio of effective ACH to mean ACH

3.2. Energy Use

The energy use results were all expressed in time dependent value (TDV) in thousands of British thermal units (kBtu) using conversions from kilowatt-hour (kWh) and natural gas therms provided by the California Energy Commission. These calculations provided a different value to electricity energy use for each hour of the year for each climate zone. The results were all expressed in terms of site energy. The results of the continuous exhaust case (Case 2) are expressed relative to the unvented house to show the effect of the commonest simple mechanical ventilation system. The other mechanical ventilation systems results are expressed relative to the simple exhaust case. The figures illustrating the energy use for each ventilation case break down the total into four parts:

1. Furnace blower electricity use
2. Natural gas consumption for heating
3. Compressor electricity use for cooling
4. Electricity use by mechanical ventilation fans

This allowed for observation of the dominant energy consuming category.

3.2.1. Case 1. Unvented House

In most CZs, natural gas for heating dominated energy use, as shown in Figure 1. However, in CZ 15 compressor power for cooling dominated. The climates near the coast (CZs 1 through 7) and mountain climate (CZ 16) had very small cooling energy use (<5000 TDVkBtu) and CZ15 had twice the cooling energy use of the next CZ. The only significant effect of using TDV rather than energy use was that CZ13 used more energy in kWh but less TDV when compared to CZ14.

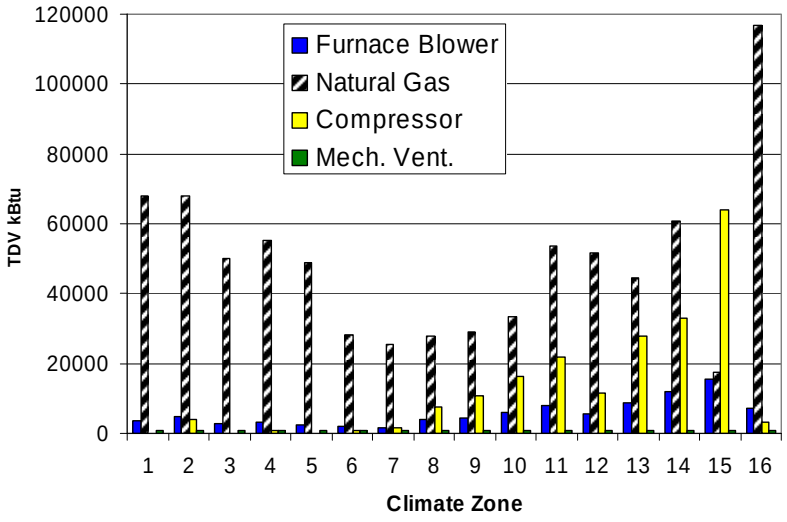

Figure 1. TDV Energy use in medium-sized unvented homes. Mechanical ventilation is intermittently operated kitchen and bath fans only.

3.2.2. Continuous Exhaust vs. Unvented House (Case 2 vs. Case 1)

The continuous exhaust mechanical ventilation exceeds the minimum required by ASHRAE 62.2 by 25 cfm to allow for periodic turning off of the system (whether as part of a controlled ventilation system or by occupant intervention). The average increase in TDV energy over all the CZs of about 10% (or about 6,500 kBtu) was dominated by increased natural gas use for winter heating (except for CZ 15 where electricity use dominated), as illustrated in Figure 2. For this reason, the relative energy cost of adding mechanical ventilation was greatest in climates with the greatest heating requirements: CZ 16 and CZ 1. On a percentage basis, the impact was greater for CZs 6 and 7, with their low baseline heating requirements. CZs 6, 7, and 8 showed reductions in cooling energy use because the extra ventilation in these relatively mild cooling climates led to increased ventilation cooling. In the mild climates (6 through 9) the energy to run the mechanical ventilation fans was close to that used to condition the air. In other climates the energy used to condition the air dominated.

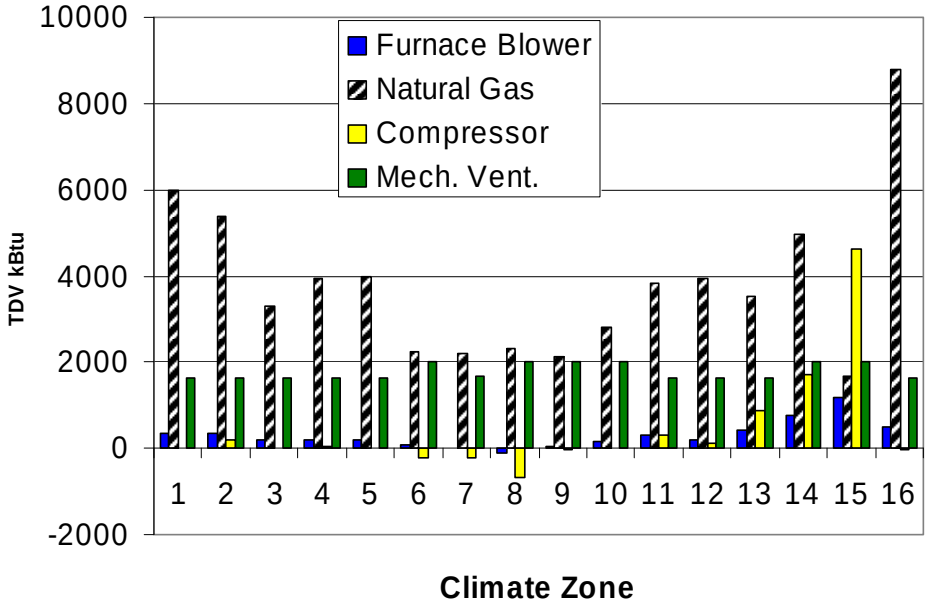

Figure 2. Change in TDV energy use for continuous exhaust compared to an unvented house

3.2.3. Intermittent Exhaust vs. Continuous Exhaust (Case 3 vs. Case 2)

The average change in TDV energy due to the 17% reduction in bathroom fan operating hours was a decrease of about 2500 kBtu or about 2.5%. This decrease was mostly due to a combination of reduced cooling in CZs 13 and 15, reduced heating in 16, and a 13% reduction in mechanical ventilation fan power use in all 5 CZs examined. Figure 3 shows the breakdown of the energy use for the five CZs in which intermittent exhaust was simulated.

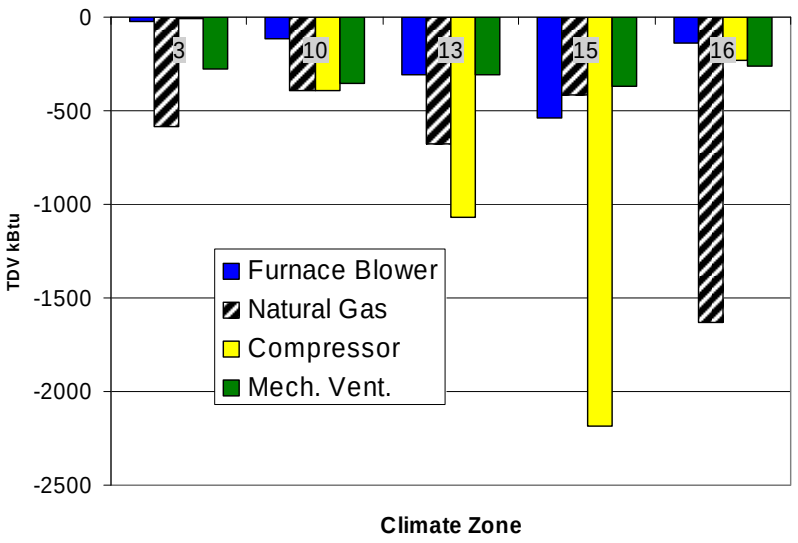

Figure 3. Change in energy use for intermittent exhaust compared to continuous exhaust reference

3.2.4. HRV Matching 62.2 Airflow vs. Continuous Exhaust (Case 4 vs. Case 2)

The average change in TDV energy was a decrease of about 6000 kBtu or about 5% and was dominated by a reduction in the heating load and the consequent reduction in natural gas use. Because the HRV only operated for 10 minutes out of every hour, the mechanical ventilation power requirements were almost identical to those for continuous exhaust. Note that the energy savings shown in Figure 4 were for the two coldest California CZs; these results cannot be extrapolated to milder climates with less heating and less opportunity to reduce the heating requirements of the ventilation air.

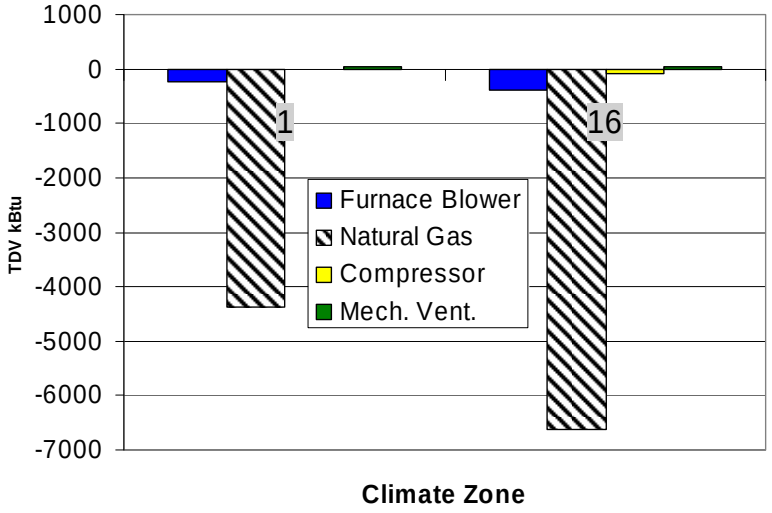

Figure 4. Change in energy use for HRV at 62.2 airflow compared to continuous exhaust reference

3.2.5. HRV at 50% Ontime vs. Continuous Exhaust (Case 4X vs. Case 2)

The average change in TDV energy was an increase of about 1000 kBtu or about 1%, and there was a balance between the extra power requirements of the HRV fan and the reduction in heating requirements. It should be possible to increase the energy savings of the HRV by operating it only in winter, when the natural gas savings are realized. For the rest of the year, there were no savings to offset the HRV fan power use. Comparing Figure 5 to Figure 4 shows the extra energy used to run the fan for 30, rather than 10, minutes of the hour and the reduction in natural gas savings due to longer HRV operating time during climate conditions where there was little to be gained from heat recovery.

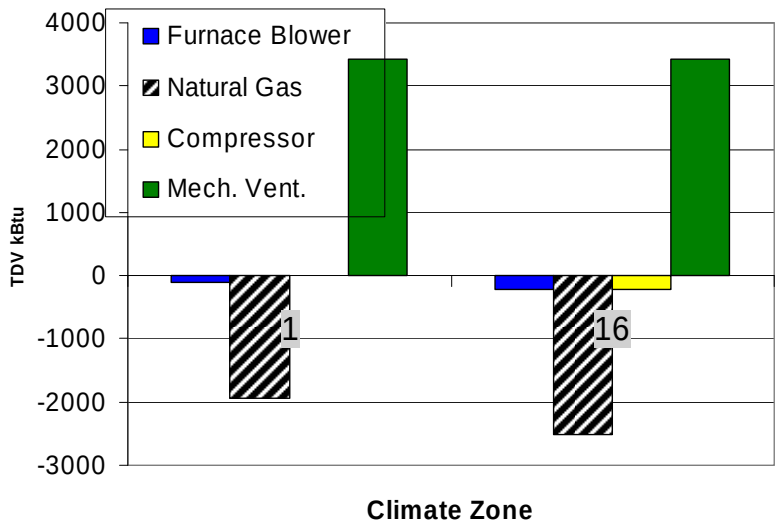

Figure 5. Change in energy use for HRV at 50% ontime compared to continuous exhaust reference

3.2.6. CFI with Continuous Exhaust vs. Continuous Exhaust (Case 5 vs. Case 2)

The average change in TDV energy was an increase of about 19,000 kBtu or about 22% and was dominated by power requirements of the furnace blower used to distribute ventilation air. The potential exists to reduce furnace blower energy use by using ducts, filters, and coils with lower air flow resistance and using electric motors that offer increased efficiency at the lower resulting external static pressures. The use of the central blower increased the mechanical ventilation fan power use considerably, as can be seen by comparing Figure 6 to Figures 3-5.

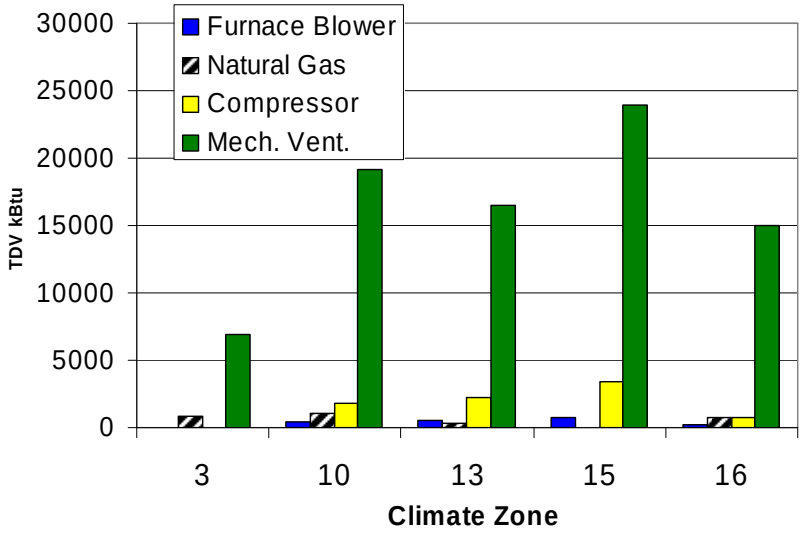

Figure 6. Change in energy use for CFI with continuous exhaust compared to continuous exhaust reference

3.2.7. Continuous Supply vs. Continuous Exhaust (Case 6 vs. Case 2)

The average change in TDV energy was an increase of about 11,000 TDVkBtu or about 13% and was dominated by the power requirements of the supply fan, which has to move four times as much air as continuous exhaust to allow for tempering of outside air. Figure 7 shows that the mechanical ventilation fan energy use dominated the continuous supply system performance. This indicates that there may be scope for improving continuous supply performance using more efficient fans.

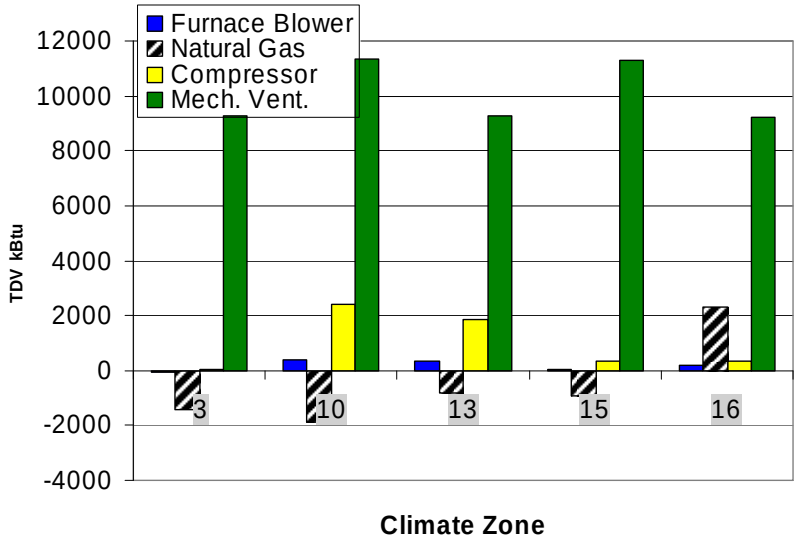

Figure 7. Change in energy use for continuous supply compared to continuous exhaust reference

3.2.8. CFI with 7% Outside Air vs. continuous exhaust – Not ASHRAE 62.2 Compliant (Case 7 vs. Case 2)

The average change in TDV energy was an increase of about 16,000 kBtu or about 18%. Figure 8 shows that, as with all the CFI systems, the electricity used to operate the furnace blower dominated the energy used by this system. The natural gas savings were due to non ASHRAE 62.2–compliant ventilation, as well as the heating effect of the mechanical ventilation fans.

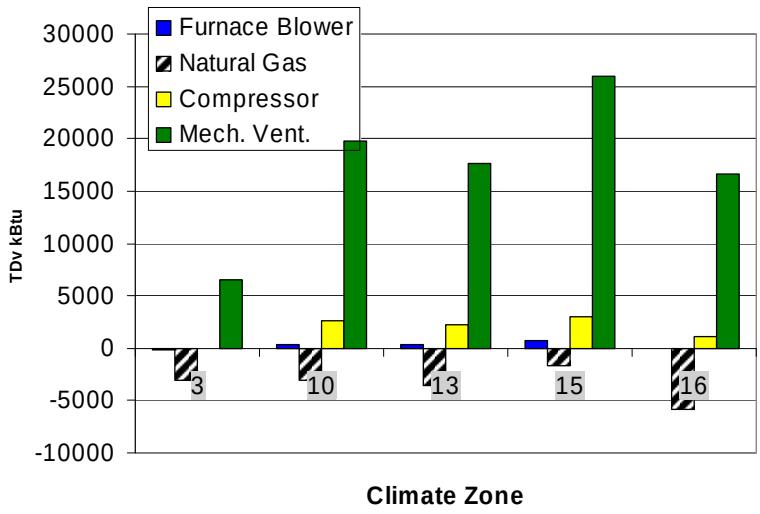

Figure 8. Change in energy use for CFI with 7% OA compared to continuous exhaust reference

3.2.9. CFI with 1/3 of ASHRAE 62.2 Flow vs. Continuous Exhaust – Not ASHRAE 62.2 Compliant (Case 8 vs. Case 2)

The average change in TDV energy was an increase of about 17,000 kBtu or about 20%. As with all the CFI systems, the electricity used to operate the furnace blower dominated the energy used by this system, as shown in Figure 9. The natural gas savings were due to non ASHRAE 62.2–compliant ventilation, as well as the heating effect of the mechanical ventilation fans.

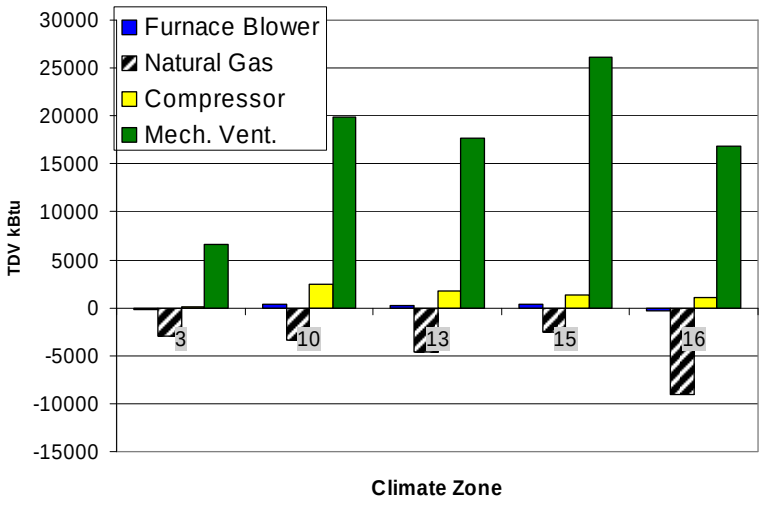

Figure 9. Change in energy use for CFI with 1/3 of ASHRAE 62.2 flow compared to continuous exhaust reference

3.2.10. Low-Ventilation Rate Adder (Case 9 vs. Case 2)

The low-ventilation rate added was used for both the unvented house and the continuous exhaust case.

For the unvented house, the addition of 0.35 ACH when the ventilation rate falls below 0.35 ACH made a significant difference. It added 0.32 ACH to the mean ventilation rates in CZ 7, 0.23 ACH in CZ 16, and 8 to 15% to HVAC energy use. AS shown in Figure 10, in CZ 7 and CZ 16, the low ventilation rate adder reduced air conditioning use by 20% and 12%, respectively, because these CZs had cool nights during the cooling season and the added 0.35 increased ventilation cooling. In CZ15, the consistently high outdoor temperatures precluded any ventilation cooling, and 6% more cooling energy was required. For heating, 11% to 20% more natural gas was used, with more energy used in colder climates.

For cases with continuous exhaust, the effect of the added ventilation was 0.28 ACH for CZ 7 and 0.13 ACH in CZ 16. This was less than for the unvented house because the added ventilation was invoked less often. Overall the low-ventilation rate adder increased continuous exhaust energy use by 3% to 11%.

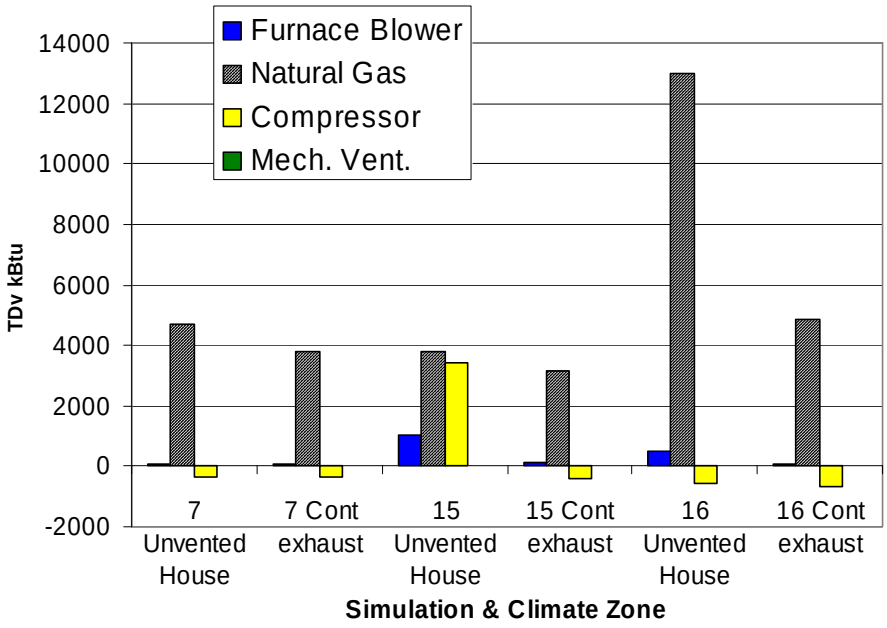

Figure 10. Effects of ventilation adder for unvented house and continuous exhaust

3.2.11. Peak Day Extra Ventilation Effects

A key issue to be resolved in this study was the contribution of extra ventilation to power consumption on TDV peak days. Table 8 summarizes the energy use and average ventilation for the TDV electricity peak days in climates with more than 12 kWh of energy use for the TDV electricity peak day. These results were for the medium-sized house with ASHRAE 62.2 compliant systems (except for the unvented house that was included as the base case). Because these results were for a single day, they do not necessarily represent overall system performance. Nonetheless, they gave useful indications of the effects of ventilation systems on peak TDV electricity cost days.

These results showed the following:

- Compared to the unvented house, the continuous exhaust systems added substantially to the ventilation rate on the peak day (by about 0.2 ACH), but the energy penalty was smaller than the change in ventilation rate, which ranged from an 8% reduction in energy use in CZ 8 to an 8% increase in energy use in CZ 14.
- Compared to the unvented house, the intermittent exhaust in CZs 13 and 15 reduced ventilation at the peak time and reduced energy use by 5%. In CZ 13, intermittent exhaust used less energy than the unvented house.
- Compared to the unvented house, continuous supply system had greater fan power requirements, which led to energy use increases of 12% to 26% in CZs 10, 13, and 15.
- The CFI systems that use the furnace blower used the most energy, with increases from 19% to 40% over that of the unvented house.

Table 8. Results from TDV peak days for energy use

CZ	Ventilation System	Average Indoor-Outdoor temperature difference C	Energy kWh	Average ACH	Energy relative to unvented house, kWh	ACH relative to unvented house	Energy relative to unvented house %	ACH relative to unvented house %
8	Unvented House	0.4	13.1	0.16	-	-	-	-
	Cont. Ex.	0.2	12.0	0.37	-1.1	0.21	-8	126
9	Unvented House	-2.4	19.8	0.20	-	-	-	-
	Cont. Ex.	-2.5	19.4	0.41	-0.4	0.22	-2	109
10	Unvented House	-3.0	20.0	0.20	-	-	-	-
	Cont. Ex.	-3.2	21.7	0.42	1.7	0.21	8	105
	CFI + Cont. Ex.	-3.1	28.0	0.48	8.0	0.28	40	137
	Cont. Sup.	-2.9	25.1	0.38	5.1	0.18	26	87
12	Unvented House	-2.4	19.8	0.21	-	-	-	-
	Cont. Ex.	-2.7	20.3	0.42	0.5	0.21	3	99
13	Unvented House	-4.4	28.1	0.20	-	-	-	-
	Cont. Ex.	-4.3	29.3	0.43	1.2	0.23	4	54
	Int. Ex.	-4.5	27.8	0.39	-0.3	0.19	-1	44
	CFI + Cont. Ex.	-4.6	33.5	0.49	5.4	0.29	19	68
	Cont. Sup.	-4.2	32.3	0.38	4.2	0.18	15	42
14	Unvented House	-4.5	27.5	0.20	-	-	-	-
	Cont. Ex.	-4.5	29.7	0.43	2.2	0.23	8	82
15	Unvented House	-7.9	43.8	0.26	-	-	-	-
	Cont. Ex.	-7.9	46.8	0.49	3.0	0.24	7	55
	Int. Ex.	-7.9	44.8	0.45	1.0	0.19	2	45
	CFI + Cont. Ex.	-8.0	53.0	0.55	9.1	0.29	21	68
	Cont. Sup.	-8.0	48.9	0.41	5.1	0.15	12	36

Table 9. Results from TDV peak days in TDV$

CZ	Ventilation System	Average Indoor-Outdoor temperature difference C	TDV$	Average ACH	TDV$ relative to unvented house	ACH relative to unvented house	TDV change %	ACH change as %
8	Unvented House	0.4	166.72	0.16				
	Cont. Ex.	0.2	146.27	0.37	-20.45	0.21	-12	126
9	Unvented House	-2.4	216.37	0.20				
	Cont. Ex.	-2.5	205.94	0.41	-10.43	0.22	-5	109
10	Unvented House	-3.0	150.28	0.20				
	Cont. Ex.	-3.2	152.77	0.42	2.49	0.21	2	105
	CFI + Cont. Ex.	-3.1	177.57	0.48	27.30	0.28	18	137
	Cont. Sup.	-2.9	186.13	0.38	35.85	0.18	24	87
12	Unvented House	-2.4	205.70	0.21				
	Cont. Ex.	-2.7	204.71	0.42	-0.99	0.21	0	99
13	Unvented House	-4.4	135.47	0.20				
	Cont. Ex.	-4.3	140.83	0.43	5.35	0.23	4	54
	Int. Ex.	-4.5	133.96	0.39	-1.51	0.19	-1	44
	CFI + Cont. Ex.	-4.6	149.91	0.49	14.44	0.29	11	68
	Cont. Sup.	-4.2	149.33	0.38	13.86	0.18	10	42
14	Unvented House	-4.5	213.69	0.20				
	Cont. Ex.	-4.5	229.98	0.43	16.29	0.23	8	82
15	Unvented House	-7.9	412.56	0.26				
	Cont. Ex.	-7.9	426.07	0.49	13.51	0.24	3	55
	Int. Ex.	-7.9	417.10	0.45	4.54	0.19	1	45
	CFI + Cont. Ex.	-8.0	456.51	0.55	43.95	0.29	11	68
	Cont. Sup.	-8.0	435.05	0.41	22.49	0.15	5	36

Table 9 shows the same results as in Table 8, but with TDV$ instead of energy. In this table, the TDV hourly energy weightings and a conversion from energy to dollars were used to provide a number of TDV$ for this peak day. In general, the fractional (percentage) differences were smaller in TDV$ terms than energy terms. This finding implied that much of the energy differences must be at off-peak conditions. Figure 11 illustrates this by comparing the energy use, TDV$, and hourly TDV$ rate for the unvented house and CFI plus continuous exhaust in CZ15 (revealing a large difference in percentage changes between Tables 8 and 9).

Figure 11 shows that when the TDV$ multiplier was high, the differences in energy use were small. It further shows that when the TDV$ multiplier is relatively low, the differences in energy use between the two cases were predominantly at off peak conditions. The small differences at peak are because the air conditioner was operating continuously at peak conditions no matter how the house was ventilated. The only difference in on-peak energy use was use of electricity to power the ventilation systems that were independent of the furnace blower. The power consumption of these fans was insignificant compared to the power consumption of the air conditioner. In addition, the reduction in thermostat setpoint between hours 14 and 18 also tended to make the air conditioner operate continuously. This time period was coincident with the TDV$ peak. This combination of operating characteristics led to the counter-intuitive result that applying peak TDV multipliers to electricity resulted in reducing the differences between ventilation systems.

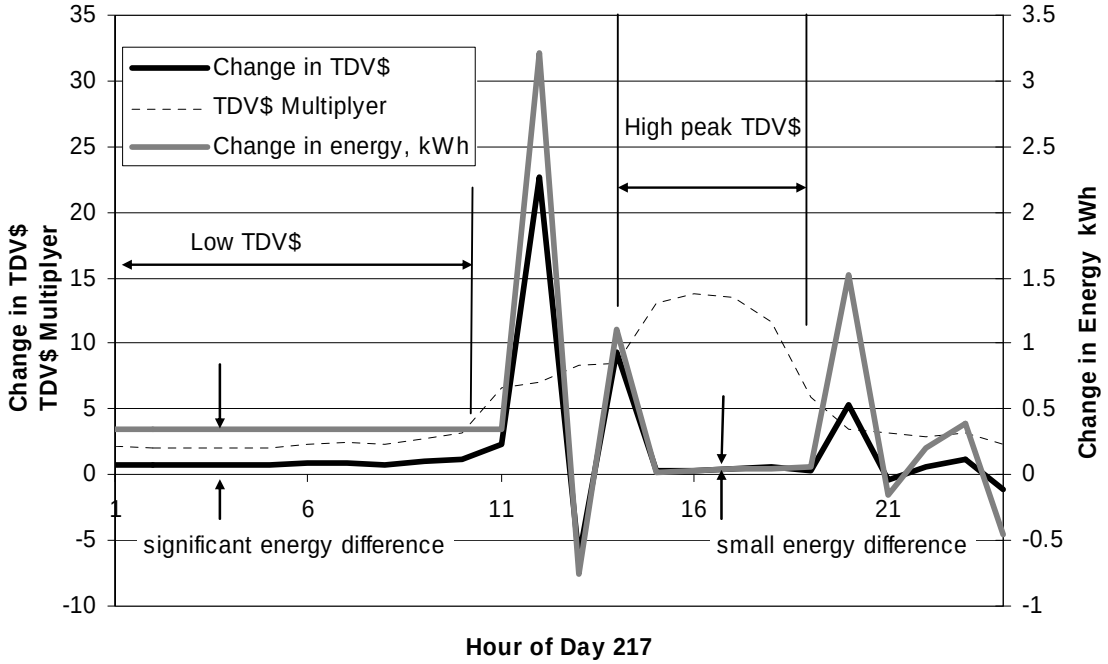

Figure 11. Change in time of energy use in CZ 15 of a CFI plus continuous exhaust relative to the baseline unvented house

3.3. Indoor Concentrations of Pollutants at Low Ventilation Rates

For indoor air quality issues, a key area of concern is periods of low ventilation, when indoor concentrations of contaminants are highest. To compare the different systems, hours when the unvented house had low ventilation rates were selected (hours where the mean ACH was less than 0.1). The concentrations were calculated using the minute-by-minute air flows and assuming a constant indoor emission rate. The results were normalized by comparing to the indoor concentration that would occur for the same house constantly ventilated to the ASHRAE 62.2 rate below, which is close to 0.3 ACH:

0.03×floor Area (ft2) + 7.5 cfm/person

The results (summarized in Appendix D) showed that the unvented house often has indoor concentrations two to three times higher than those of the mechanically ventilated cases that meet ASHRAE 62.2.

Figure 12 shows the difference in air concentrations between a simple continuous exhaust system and the unvented house for each hour of the year during which the unvented house had less than 0.1 ACH. The continuous exhaust results were much more uniform, and generally two to three times lower, than those for the unvented house.

Figure 13 is more complex and includes the many systems examined in CZ10. However, the delineation is still clear between the unvented house and mechanically ventilated houses, even for those systems that do not meet AHSRAE 62.2 (case 7, CFI 7% OA, and case 8, CFI 1/3 62.2).

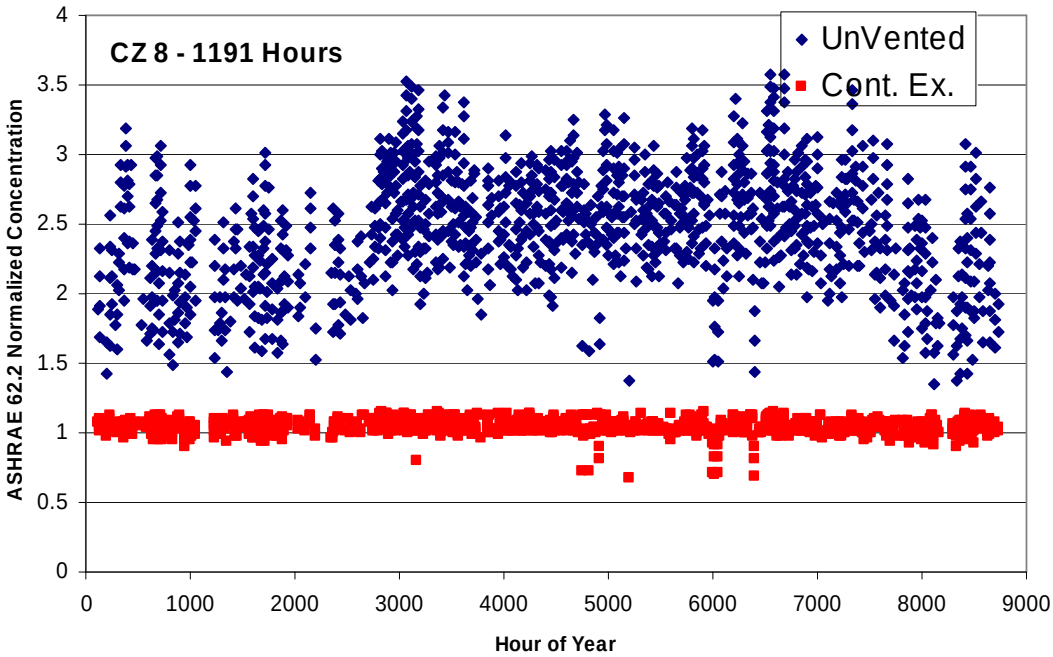

Figure 12. Comparison of indoor air concentrations for a home with continuous exhaust and an unvented home in CZ 8 for periods of low air change rate in the unvented house

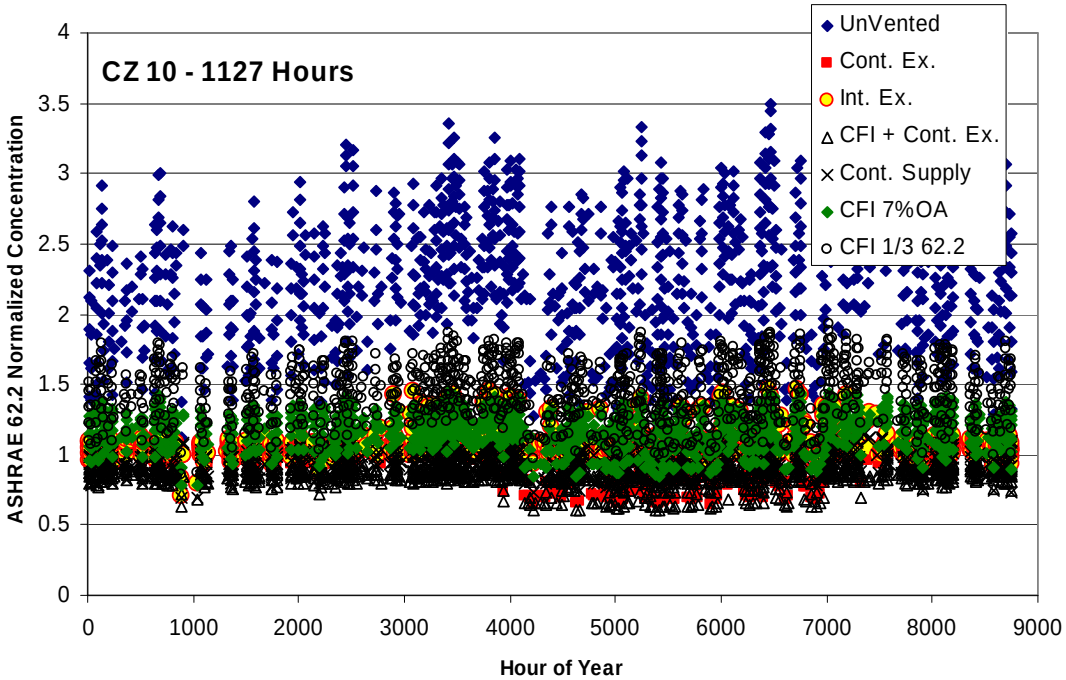

Figure 13. Comparison of indoor concentrations in CZ 10 during periods of low air change rates in the unvented house

3.4. Delivered Air Temperatures for Supply Ventilation

Delivered air temperatures are important for supply air systems because if air is supplied at too high or low a temperature, then occupant comfort will be compromised. For the HRV, continuous supply, and the CFI with continuous exhaust, the supply air temperatures were calculated for every minute of the year. These data were then binned by temperature to see how often a particular delivered air temperature occurs during the year.

For the HRV, the delivered air temperature (T_{del}) was determined from the apparent sensible effectiveness (0.7) and from the indoor (T_{in}) and outdoor (T_{out}) temperatures:

$$T_{del} = 0.7T_{in} + 0.3T_{out}$$

The HRV was operated in two modes: either 10 minutes out of each hour (10/60) or 30 minutes out of each hour (30/60).

Similarly, it was assumed that the continuous supply mixed indoor air with outdoor air in a ratio of 3:1, such that:

$$T_{del} = 0.75T_{in} + 0.25T_{out}$$

Lastly, the CFI system mixed outdoor air with circulating air at a ratio of about 1:15. In addition, heat transfer in and out of the duct system occurred, which can change the delivered air temperature. The CFI temperatures were split into five categories:

1. CFI with no heating or cooling
2. CFI when heat is also on
3. CFI when cooling is also on
4. Heating only – CFI duct closed
5. Cooling only – CFI duct closed

These five categories showed much CFI operation changed delivered air temperatures when the system was heating and cooling, as well as when the CFI was operating for ventilation only (Category 1).

For climate zone 1, only the HRV was simulated. The results in Figure 14 show that for a few minutes of the year (210 minutes for 10/60 operation and 660 minutes for 30/60) the delivery temperatures got as low as 11°C (52°F) but the majority of operation was in the 15/16°C range (59-61°F). This finding suggests that supply vents should be carefully placed so as not to blow directly on occupants.

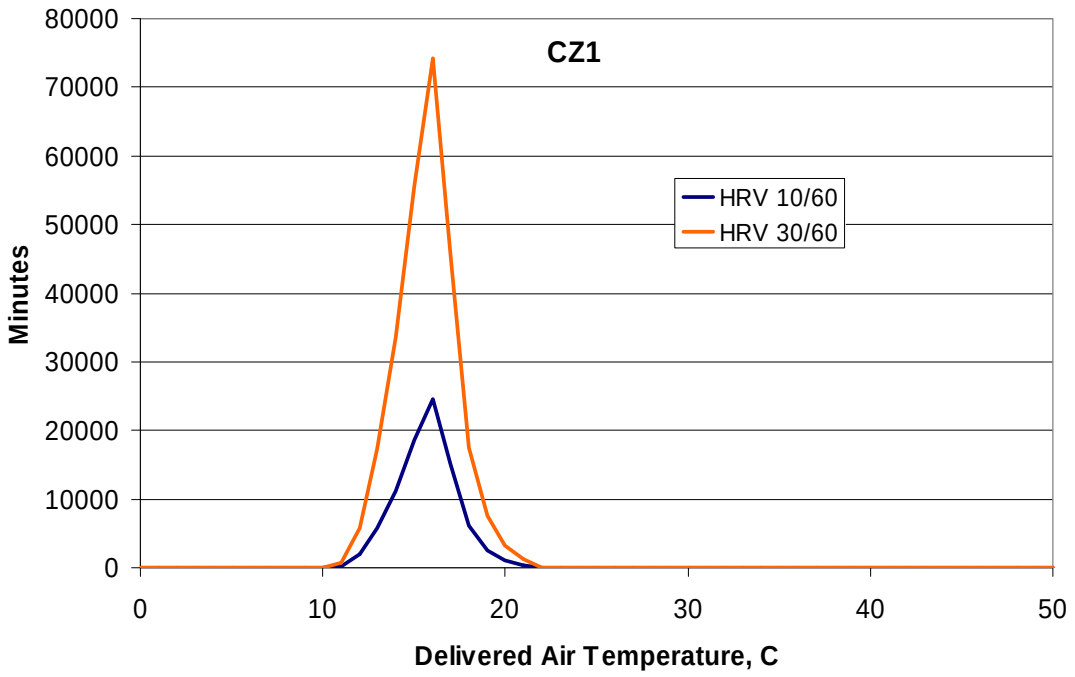

Figure 14. Distribution of delivered air temperatures for HRVs operating 10 and 30 minutes per hour in CZ1

The CFI and continuous supply were simulated in CZ3. Figure 15 shows that, when there was no heating or cooling, the CFI delivered air at less extreme temperatures than did the continuous supply because the CFI mixed outdoor air with more indoor air. For both systems, supply air temperatures were sufficiently low to require care to avoid cold drafts for occupants. Another option to preserve comfort is to use of controls to turn off the systems when outdoor

temperatures are low (e.g., below 0°C (32°F)); at these low ambient temperatures, the natural infiltration through the envelope is high enough that turning off the mechanical ventilation did not result in unacceptably low ventilation rates. Heating with the CFI duct open resulted in a wider range of delivered air temperatures compared to heating when the CFI duct was closed. This shifted the median delivered temperature from 48°C (118°F) to 46°C (115°F) but never delivered air below 36°C (97°F). CFI operation tended to spread out the delivered air temperatures when operating in conjunction with heating and cooling. In CZ 3, there were a few minutes (only 424 minutes, or about 7 hours, for the whole year) of cooling but not enough to be directly visible in the figure.

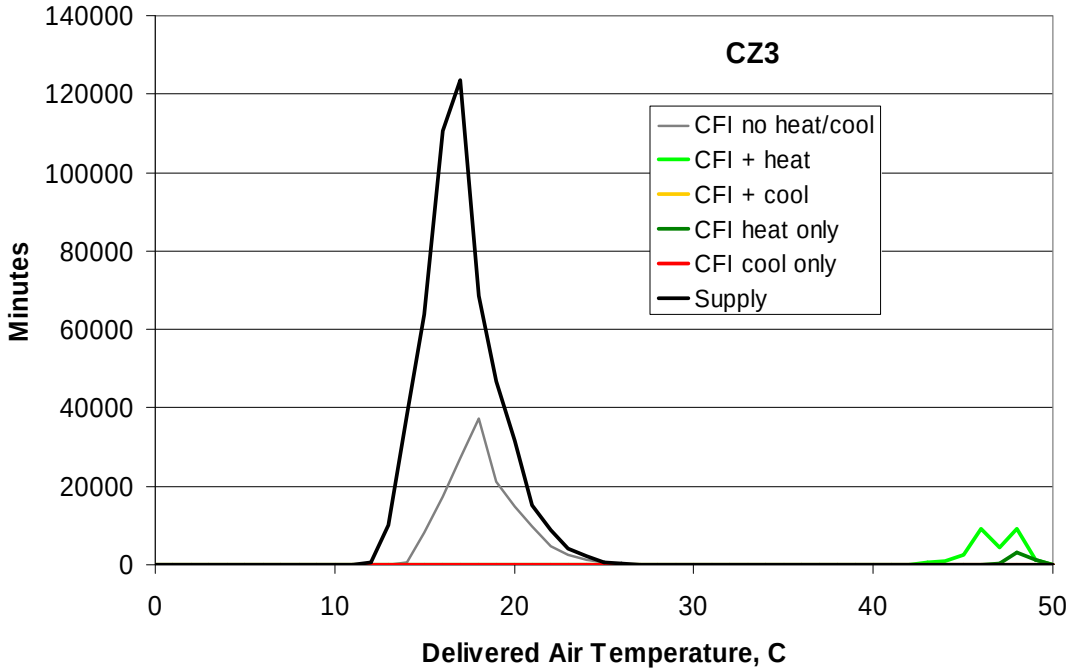

Figure 15. Distribution of delivered air temperatures in CZ3 for CFI and continuous supply

For Climate Zone 10, 13 and 15 the results were similar: the CFI consistently delivered air in a narrower range of temperatures than did continuous supply. This was most apparent in CZ16, where air supplied by the CFI systems did not dip below 14°C (57°F), but air supplied by the continuous supply fell as low as to 9°C (43°F). CZ16 also included HRVs; as in CZ3, HRVs tended to produce the lowest delivered air temperatures. Note that the HRVs (and continuous supplies) in these simulations were not linked to heating or cooling system operation, and they could be installed (together with the appropriate controls) to synchronize heating and cooling with ventilation to provide tempering of extreme delivery temperatures.

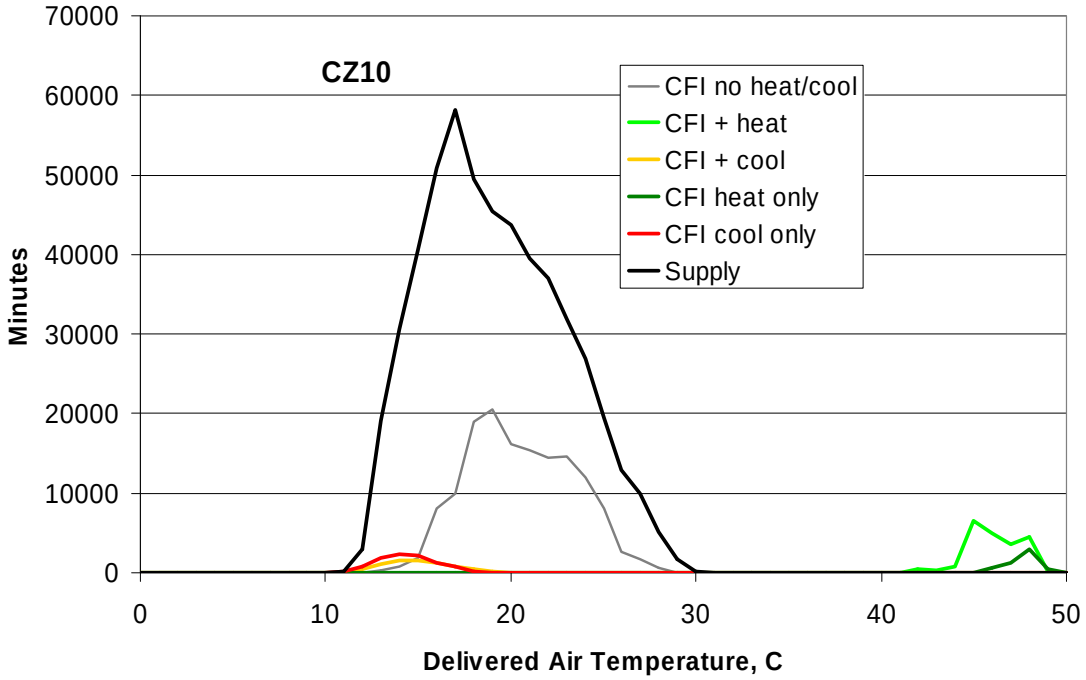

Figure 16. Distribution of delivered air temperatures in CZ10 for CFI and continuous supply

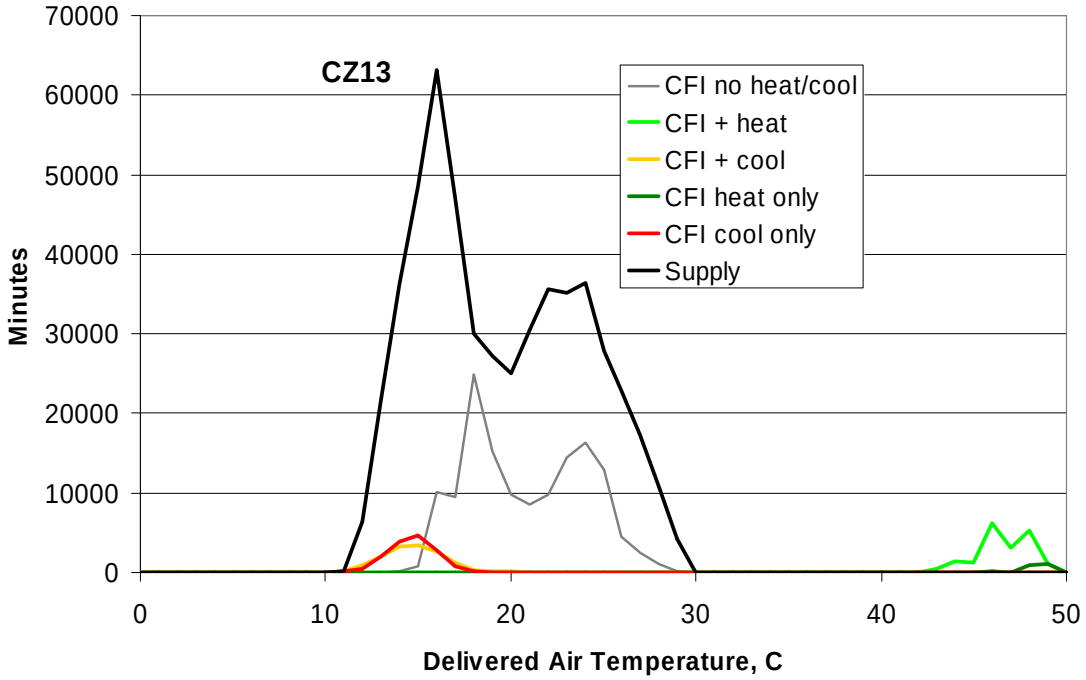

Figure 17. Distribution of delivered air temperatures for CZ13 for CFI and continuous supply

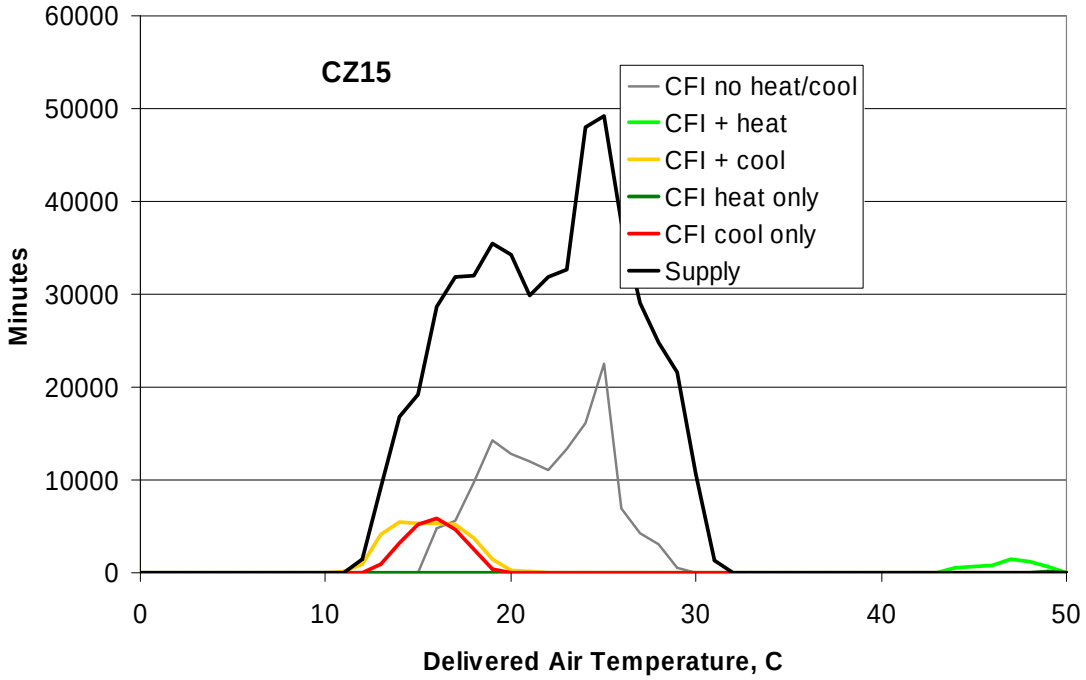

Figure 18. Distribution of delivered air temperatures in CZ15 for CFI and continuous supply

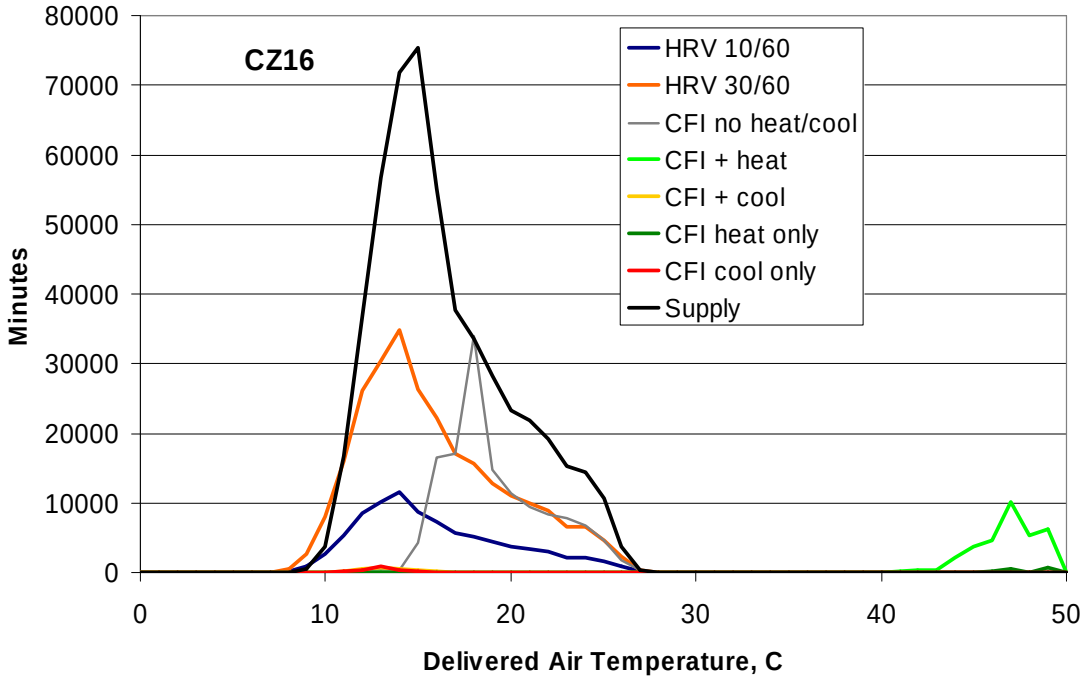

Figure 19. Distribution of delivered air temperatures in CZ16 for CFI and continuous supply

3.5. Effects of Leaky Ducts on CFI performance

To examine the effect of leaky ducts on CFI performance, simulations were performed for CZs 3, 13, and 16 with duct leakage increased from 5% to 11%. As shown in Figure 20, increased duct leakage increased TDV significantly (9% on average for these three climates), indicating that the CFI system should only be used with tight ducts. Because more time is spent heating and cooling (leading to increases in heating and cooling energy), a slight decrease in TDV observed was attributed to mechanical ventilation because the CFI operated for less time without heating or cooling.

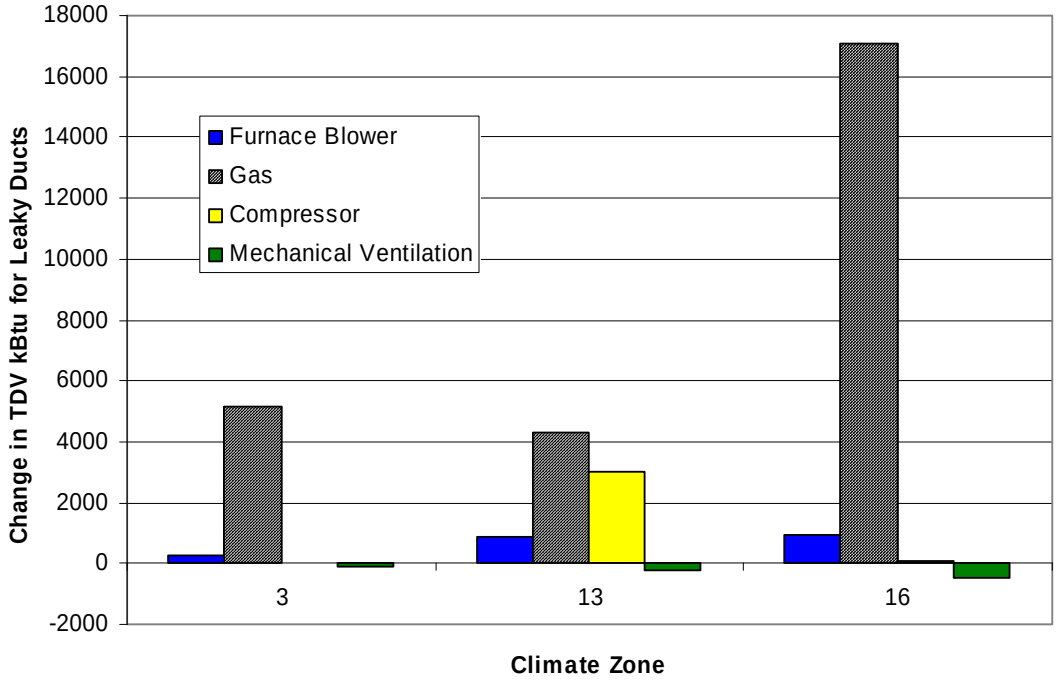

Figure 20. Effect of increasing duct leakage from 5% to 22% on CFI system performance

3.6. House Size

Three different house sizes were simulated to examine the effect on overall energy use (used for peak demand and consumer cost) and energy use per square foot of floor area. House size effects were examined for continuous exhaust, intermittent exhaust, and CFI with continuous exhaust. In the figures below, the L, M and S after each climate zone indicate large, medium or small house.

For the continuous exhaust case, CZs 3, 10, 13, 15, and 16 were simulated for all three house sizes. The results in Figure 21 show the variability with house size and clearly demonstrate the large energy savings for smaller houses.

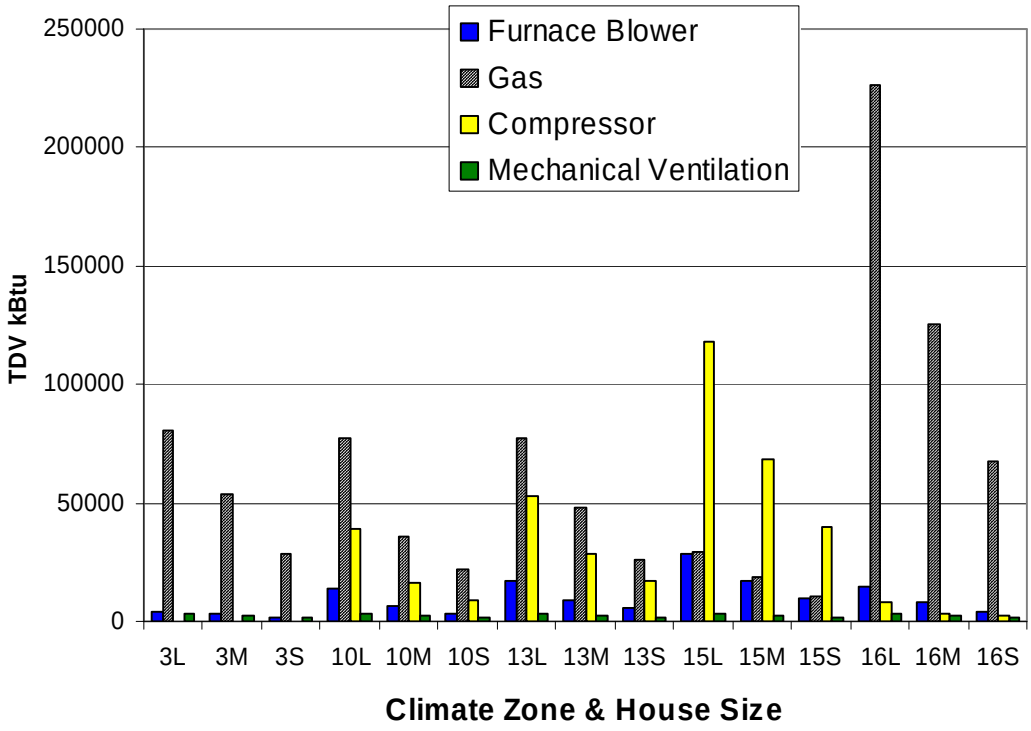

Figure 21. TDV energy use for continuous exhaust for different house sizes

In Title 24, energy use is normalized by house floor area. Figure 22 shows the same results as Figure 21 normalized by floor area. The results showed that the large house, which uses the most energy, had the lowest rating when normalized by floor area. The medium house used more energy per square foot in most cases. The exception was cooling for 15S and heating for 10S, which represent the largest energy use.

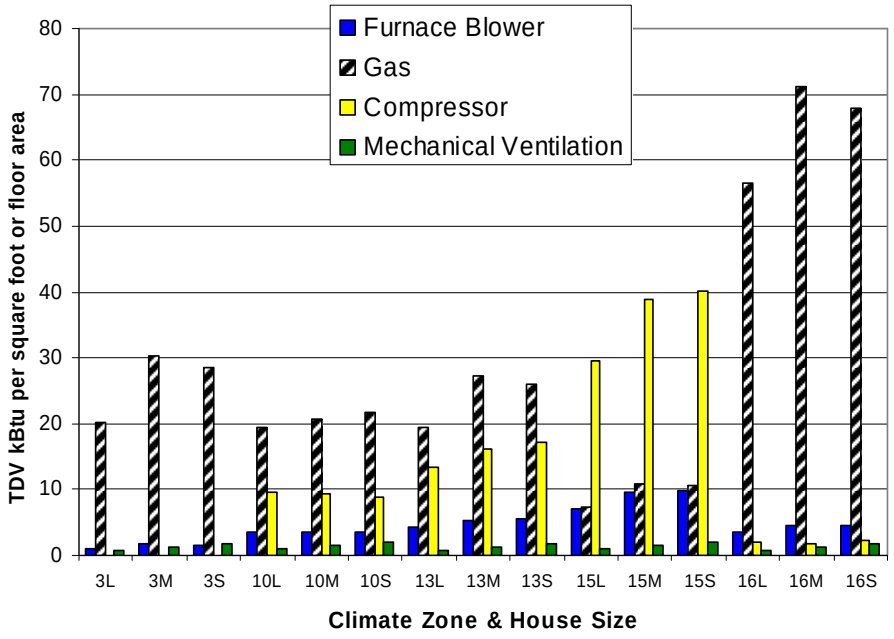

Figure 22. TDV energy use for continuous exhaust for different house sizes normalized by floor area

The air change rates in Figure 23 show that larger houses had fewer ACH. This result was expected because the sizing algorithm from ASHRAE 62.2 and the assumed occupancy did not scale directly with floor area.

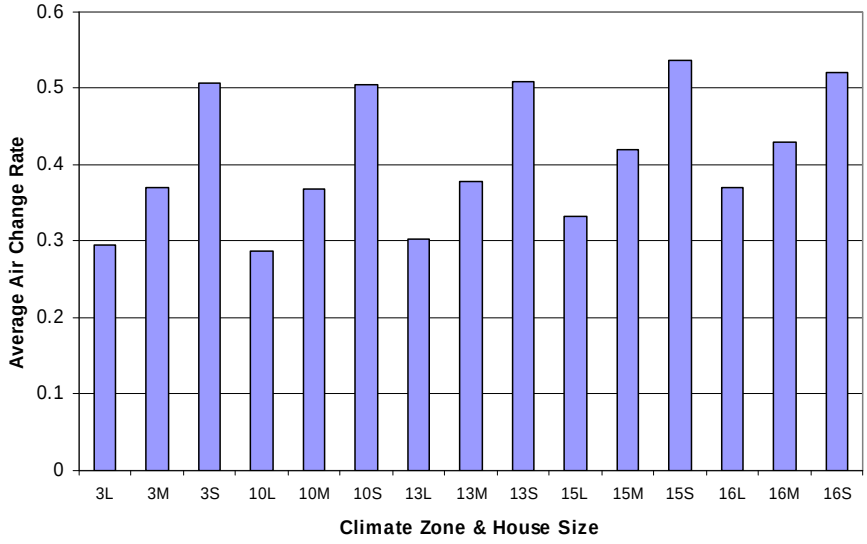

Figure 23. Average air change rate for continuous exhaust for different house sizes

The effect of house size with intermittent exhaust was investigated for climate zone 16 and shown in Figure 24. These results showed the expected scaling with house size. Figure 25 shows the same results normalized by floor area. As with the continuous exhaust case above, the medium house tended to have the highest energy use per square foot in this case.

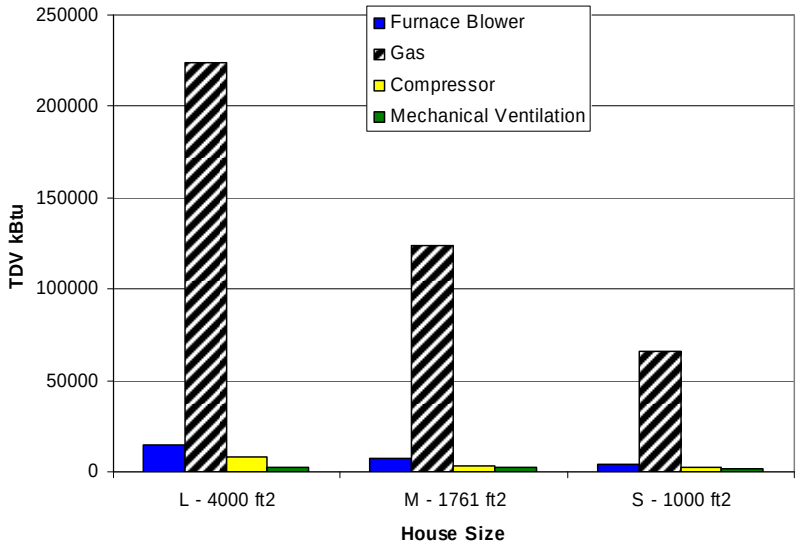

Figure 24. Effects of house size on TDV energy for intermittent exhaust in CZ 16

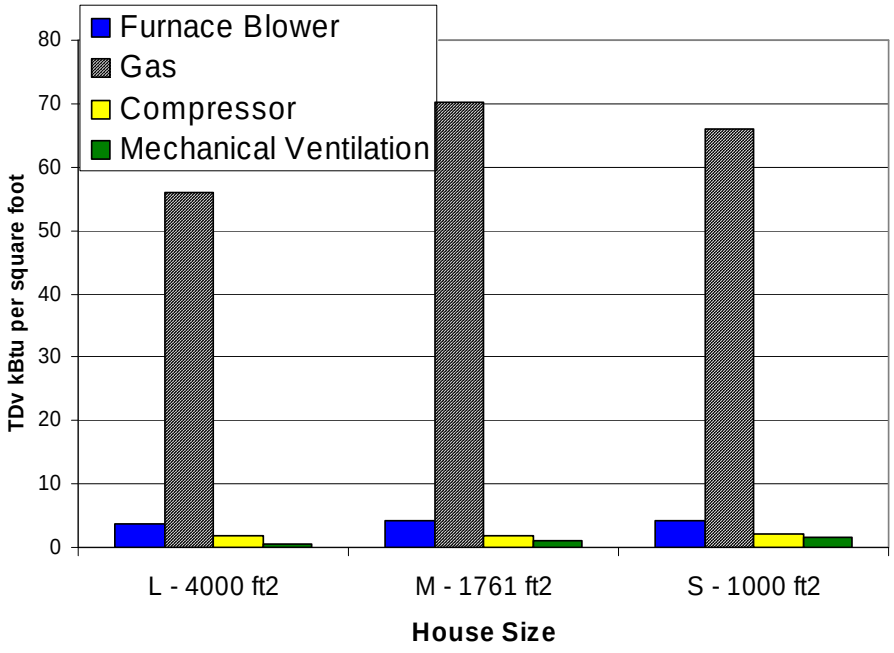

Figure 25. Effects of house size on TDV/ft^2 for intermittent exhaust in CZ 16.

CFI (case 5) was examined in CZ 10 for three house sizes. CZ 10 was less dominated by heating (gas consumption), as shown in Figure 26. This shows how the large house consumed much more energy than the other houses. Figure 27 shows these results normalized by floor area in Figure 27. The differences between the two data sets were less marked than in the comparisons above.

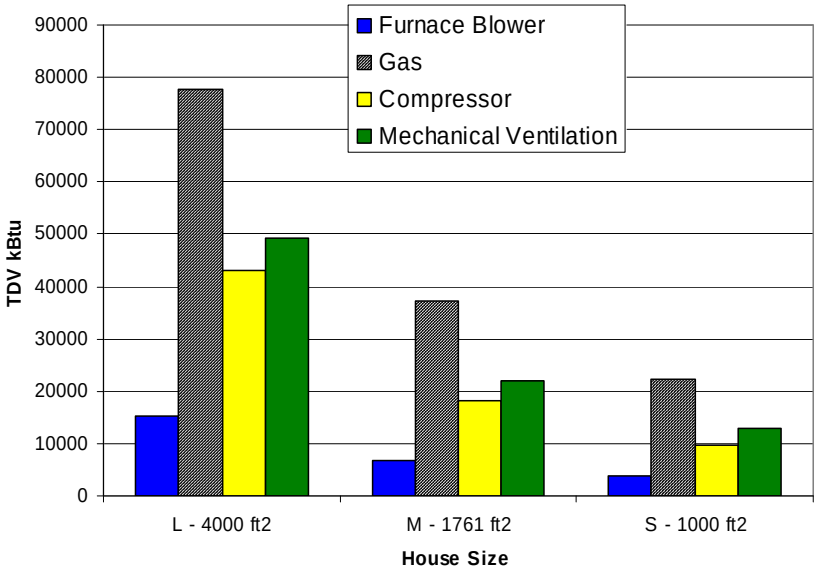

Figure 26. Effects of house size on TDV for CFI in CZ 10

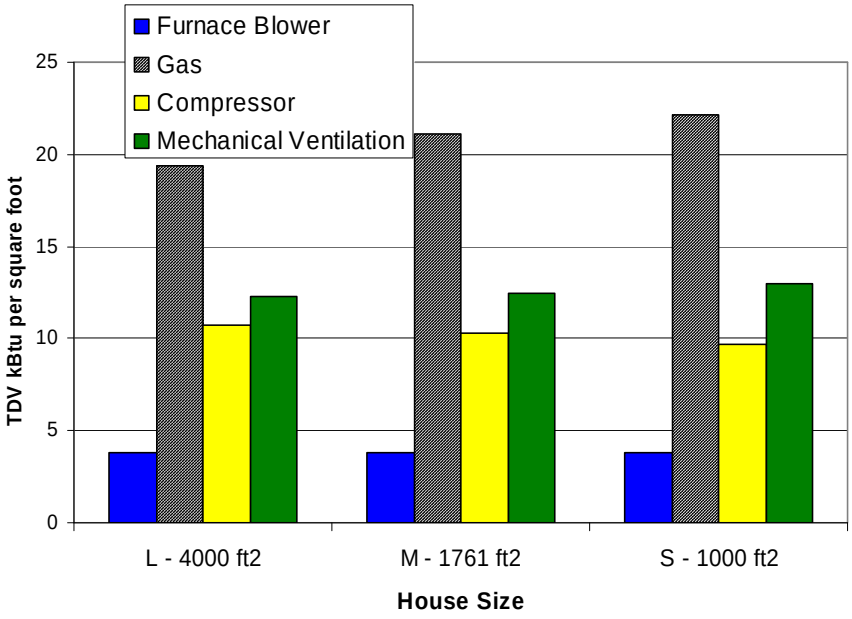

Figure 27. TDV normalized by house size for CFI in CZ 10

3.7. Effects of Intermittent Ventilation on Indoor Ozone

Usually the focus of ventilation is to provide fresh outdoor air and remove stale indoor air because the source of air pollutants is from inside the house. In some cases, however, the outdoor air may contain pollutants that occupants do not want to bring into the house. One important outdoor pollutant in California is ozone. When developing the list of acceptable technologies for ventilating California houses, the authors considered the ability of technologies to operate intermittently–thus giving the flexibility to ventilate less at peak load or if there are undesirable outdoor pollutants. The simplest example of intermittent operation is the use of an intermittent exhaust fan. The effect of an intermittent exhaust fan was evaluated by comparing the air flow rates and resulting transport of ozone into a house for both continuous and intermittent exhaust.

Data for hourly outdoor ozone concentrations were obtained from the California Air Resources Board website for Riverside, CA (in CZ 10). The peak ozone concentration day selected for this analysis was August 14th, 2006. Because the typical meteorological year (TMY) weather data and the ozone concentration data were not taken at the same time (they are for completely different years), ventilation rates taken from a typical day that was nearest to design temperature conditions were used (in this case, for CZ10, September 3, in the TMY data. The corresponding hourly averaged ventilation rates were then used to calculate the amount of ozone entering the house each hour. Indoor concentrations were not calculated because they depend on many things that were not modeled for this study, such as deposition in the envelope of the building and interaction with indoor surfaces. Instead the calculations provide a relative measure of the potential for reducing ventilation-related ozone entry.

Figure 28 illustrates the effects of the intermittent and continuous ventilation strategies on the quantity of ozone that entered the house envelope by ventilation. The differences occurred in the four afternoon/evening hours when the intermittent ventilation was off. Because the reduced ventilation was coincident with peak ozone concentration, the effect on the quantity of ozone delivered was significant. In this example, the reduction in total ozone for the day was almost 20%. For the four hours of reduced ventilation, the average difference was 40% and the greatest reduction for a single hour was 50%.

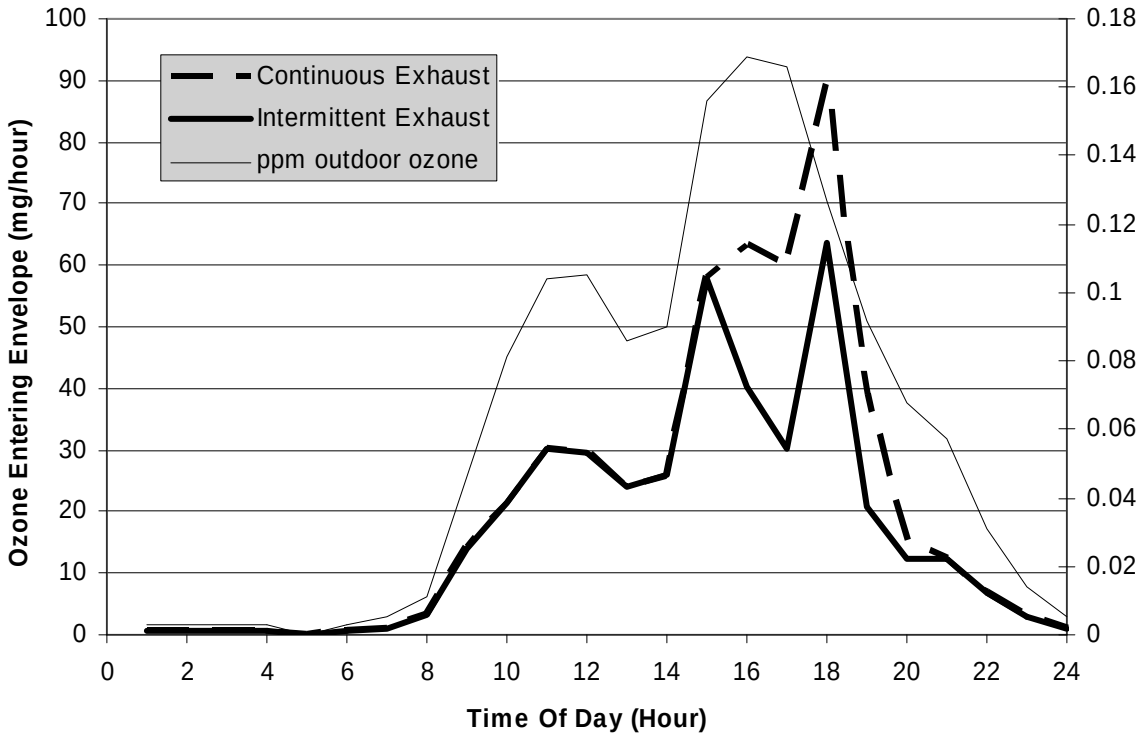

Figure 28. Outdoor concentration profile and ozone delivery rates for Riverside, CA, (CZ10) with continuous and intermittent exhaust

3.8. Economizers with Intermittent Ventilation

The principles of intermittent ventilation can allow the reduction of ventilation during some periods and greater ventilation than the minimum during the remaining periods. This flexibility is most useful when ventilation is substantially increased for some reason independent of supplying minimum ventilation, such as for ventilative cooling. The Title 24 revision could include credits to recognize the energy savings of the strategy. In many climates, the use of air exchange (i.e., flushing) to remove internally generated heat can be an energy savings strategy. Whether air exchange is achieved by opening windows or though a mechanical system, such as an economizer, the impact on indoor air quality is the same: substantially greater flushing of internally-generated contaminants.

Once the flushing is over, it is possible to delay any mechanical ventilation for a period and still achieve equivalent exposures to that assumed to be provided by a constant ventilation rate. Figure 29 was generated using intermittent ventilation equations (Sherman 2006) applied to a situation in which a large amount of flushing (at least 10 times the rate in ASHRAE 62.2) was used for a number of hours.

The bold solid line indicates the number of hours the mechanical ventilation system can be shut-off (hours of credit) following a known length of flushing. For example, after 8 hours of

flushing time (typical for nighttime economizer operation), the system can be shut off for about 11.5 hours. The dashed line is the peak concentration relative to the steady-state value that would result from 8 hours of flushing and 11.5 hours with the system off. As the dashed line shows, the peak concentration is about four times the steady-state value using this practice. If the contaminants of concern have non-linear dose-response curves (e.g., threshold values), this peak level could be important.

The thin solid line is the curve representing a single day (i.e., the number on the x-axis and the number on the y-axis add to 24). The crossing of the thin and bold lines means that no additional mechanical ventilation is needed that day to meet minimum ventilation requirements. Thus, if an economizer is running for at least 12 hours a day, no other mechanical ventilation is required that day. This practice could save substantial energy (and peak power) in hot, dry climates where flushing is commonly done at night, but air conditioning is required during the day.

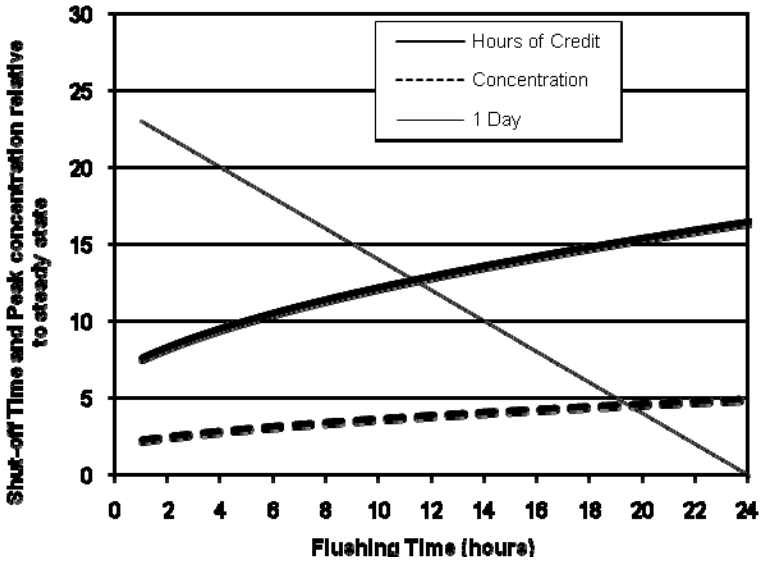

Figure 29. Hours of credit and peak concentration effects for high air flow flushing ventilation

4.0 Conclusions and Recommendations

4.1. Conclusions

This study used simulations to examine the effect of different ventilation strategies on energy use in California houses. The houses were California Building Energy Code–compliant, but not all the ventilation-related energy calculations used in the ACM were implemented (e.g., maximized ventilation cooling). The simulations focused on ventilation technologies that were complaint with ASHRAE Standard 62.2. Extra simulations were performed for some mechanical ventilation systems commonly used in California, but that do not meet ASHRAE 62.2 minimum requirements.

The simulation results showed that ASHRAE Standard 62.2–compliant ventilation systems add significantly to ventilation rates and reduce indoor pollutant concentrations. However, there was a cost associated with this extra ventilation. For a minimally compliant continuous exhaust system, the extra TDV energy use was about 10%. For perspective, this represents about the same TDV energy change as the 0.35 ACH ventilation adder used at low ventilation rates in the current ACM. Relative to the minimally ASHRAE 62.2-compliant exhaust fan, the intermittent exhaust and HRV systems reduced TDV energy use by 1%–5%, respectively. The CFI and supply systems averaged 22% and 13% more energy use, respectively, than did the continuous exhaust.

In terms of TDV, the energy required for ventilation was dominated by natural gas use for heating in most CZs, (except CZ15, where there was more electric energy used for cooling). The ventilation fan power requirements for continuous exhaust fans were about half the extra space conditioning extra load on average. In mild climates (CZ 6–9), the fan energy was about the same as the conditioning energy, but in other climates the conditioning energy dominates.

HRV energy use was dominated by the energy used to operate the HRV fans. Because HRVs give the greatest benefit at high temperature differences, operating all year when temperature differences are small causes the fan energy use to offset the space conditioning benefits. Also, due to their limited air flow range, available HRVs provided airflows that significantly exceeded the ASHRAE 62.2 minimums if operated continuously. In this study, the HRVs were operated in two modes: at 50% duty cycle and at a duty cycle that matches the ASHRAE 62.2 minimum air flow requirements.

CFI systems also provide distribution and mixing of air, an extra service beyond the basic requirements of ASHRAE 62.2. In developing code requirements, the Energy Commission needs to decide how to deal with this benefit. The ASHRAE 62.2–compliant CFI system studied used a continuously operating exhaust to meet ASHRAE 62.2 and the CFI provided extra ventilation when operating (due to the change to balanced ventilation from exhaust). It could be argued that the cost of operating the CFI should not be included in the ventilation estimates because it is providing another separate, although complementary, service.

Two non ASHRAE 62.2–compliant CFI systems currently used in California construction were also investigated. The amount of ventilation provided by these CFI systems depends on the size

of opening to outside and the air flow through the HVAC duct system. Therefore, the amount of ventilation depends on system capacity for a given fraction of outside air. Lastly, the CFI systems all have additional ventilation from duct leakage when operating. To prevent excess ventilation, their duct systems need to be as tight as possible. Even the 5% total leakage used here (2.5% each for supply and return) leads to ventilation flows that are significant (typically half of the 62.2 minimum rate).

Intermittent ventilation systems and strategies can be used to significantly reduce the effects of outdoor air pollutants. Intermittent exhaust can reduce the ozone delivered to the house by 50% at peak outdoor ozone concentration.

The Title 24 revision can provide significant credit (in terms of reduced mechanical ventilation operation) for large ventilation air flows. Typical nighttime economizer operation for 6–8 hours would allow for 10–12 hours of no mechanical ventilation requirements. This would allow for reduced mechanical ventilation (and associated air conditioning and fan power electricity consumption) through most of the day—including during the afternoon electricity peak.

In every climate, the intermittent exhaust system proved to be the most energy efficient system for meeting the proposed requirements, followed by the continuous exhaust system. Both of these systems were even more energy efficient than the non ASHRAE 62.2–compliant systems.

Exhaust systems were found to be more energy efficient for several reasons. Among these is the fact that ASHRAE 62.2 does not correctly account for the addition of infiltration and mechanical ventilation. Because the exhaust system will have a lower impact on the total air exchange, its space conditioning impact will be smaller and it will be the more cost-effective approach—but by producing fewer air changes. The issue of the role of infiltration and superposition and their application to ventilation standards requires more research.

Another conclusion is that CFI systems use substantially more energy than other systems, principally because of the enhanced air distribution provided by CFI systems. While required by neither Title 24 nor ASHRAE 62.2, air distribution can presumably have non-energy benefits in the form of increased comfort and improved indoor air quality.

Air distribution systems are used to distribute heating and cooling; distribute fresh air; and filter, clean and recirculate indoor air. As such, they can be separated from the rest of the HVAC system and should be treated separately both in regulation and design. Treating distribution as a separate function offers the advantage of allowing efficiency advances in distribution to be made independently from, for example, cooling systems. Further research is needed to determine the value of air distribution in reducing sizing of the rest of the HVAC system.

Simulation results indicate that the best way to distribute outdoor air, if deemed important, is to use a dedicated supply ventilation system. If redistributing air for comfort or filtration is important, then one of the compliant systems with CFI should be selected. Not considered in this study, however, are high-efficient variable speed motors that could be used with the central fan to distribute low volumes of air energy efficiently. Such evaluations might suggest valuable alternative systems.

HRVs have the least cost of conditioning air in extreme climates, but inclusion of the HRV fan energy cost makes the total cost about the same as for an exhaust-only system. The HRVs studied here did not, however, have any air distribution associated with them, and so at times low supply air temperatures might cause discomfort depending on the design. CFI HRV designs are also in use. Although not evaluated in this study, such a design should have an air distribution cost in line with the other CFI systems. As HRV/ERV systems become more common, evaluations of these options would be appropriate.

4.2. Recommendations

4.2.1. Future Research

Based on the results of this study, the following research needs exist:

- A broad-based field study to determine envelope and duct air leakage of current new construction and how commonly used mechanical ventilation systems perform with such leakage.
- A study to determine how contaminants are transported from garages and other buffer zones and to determine if carbon monoxide alarms are necessary.
- Evaluation of the need to have air distribution systems to provide acceptable indoor air quality.
- Testing that would allow additional ventilation credit to be given to economizers, direct evaporative coolers, and other systems that provide ventilative cooling.

4.2.2. Recommended Technologies

All the ASHRAE 62.2–compliant technologies studied here are recommended for use in California, with the following caveats:

- All selected ventilation fans should use as little energy as possible. The low sone requirement of ASHRAE 62.2 effectively biases selections toward high efficiency models already.
- Intermittent exhaust allows flexibility of operation, energy savings, and the ability to reduce the effects of outdoor pollutants, but must still be sized and operated to meet ASHRAE 62.2.
- HRV use could be optimized by using air flow rates (or by specifying a time of use that provides the ASHRAE 62.2 air flow required each hour; in the simulation presented here, that time this was 20 minutes operation per hour).
- Supply systems (either dedicated continuous supplies or intermittent CFI systems) move much more air than other systems to temper the incoming air (required for comfort). This means that more fan power is required. Some specification of this fan power requirement should be used in Title 24 compliance calculations.

4.3. Benefits to California

This work produced information that can help the State of California craft mechanical ventilation requirements that will protect health and comfort, at the lowest energy and cost penalty. Notably, it revealed that all of the ASHRAE 62.2–complaint systems are suitable for use in California, with specific caveats.

Further, the study developed findings that can help achieve specific goals:

- Use of more-efficient fans could decrease the electricity energy needed to provide mechanical ventilation.
- Use of intermittent exhaust could reduce the amount of the undesirable pollutant ozone entering into homes by as much as 50% during times of peak outdoor ozone concentration, and still maintain healthy ventilation rates.
- Use of a nighttime economizer to increase airflow could reduce mechanical ventilation during the day, including peak electricity periods.

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6.0 Glossary

ACCA	Air Conditioning Contractors of America
ACH	air changes per hour
ACM	Alternative Calculation Method
AFUE	annual fuel utilization efficiency
ANSI	American National Standards Institute
ASE	apparent sensible effectiveness
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
CFI	central fan integrated
cfm	cubic feet per minute
CZ	climate zone
EER	energy efficiency ratio
Energy Commission	California Energy Commission
ft^2	square foot
HRV	heat recovery ventilator
HVAC	heating, ventilating, and air conditioning
HVI	Home Ventilating Institute
IAQ	indoor air quality
kBtu	thousands of British thermal units
kPa	thousand Pascals
kWh	kilowatt-hour
L/s	liters per second
m	meter
m^2	meters square
OA	outside air
Pa	Pascals
PAC	Project Advisory Committee

PIER	Public Interest Energy Research
RD	research, development, and demonstration
s	second
SEER	seasonal energy efficiency ratio
SHGC	solar heat gain coefficient
SLA	specific leakage area
TDV	time dependent valuation
TMY	typical meteorological year
TXV	thermostatic expansion valve
W	watt

Appendix A: REGCAP Model Outline

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

The ventilation model developed here is a two zone model, in which the two zones are the attic and the house below it and they interact through the ceiling flow. Both zones use the same type of flow equations and solution method. The total building and attic leakage is separated into components and a flow equation is developed for each leakage site. The envelope flow components are illustrated in Figure 1.

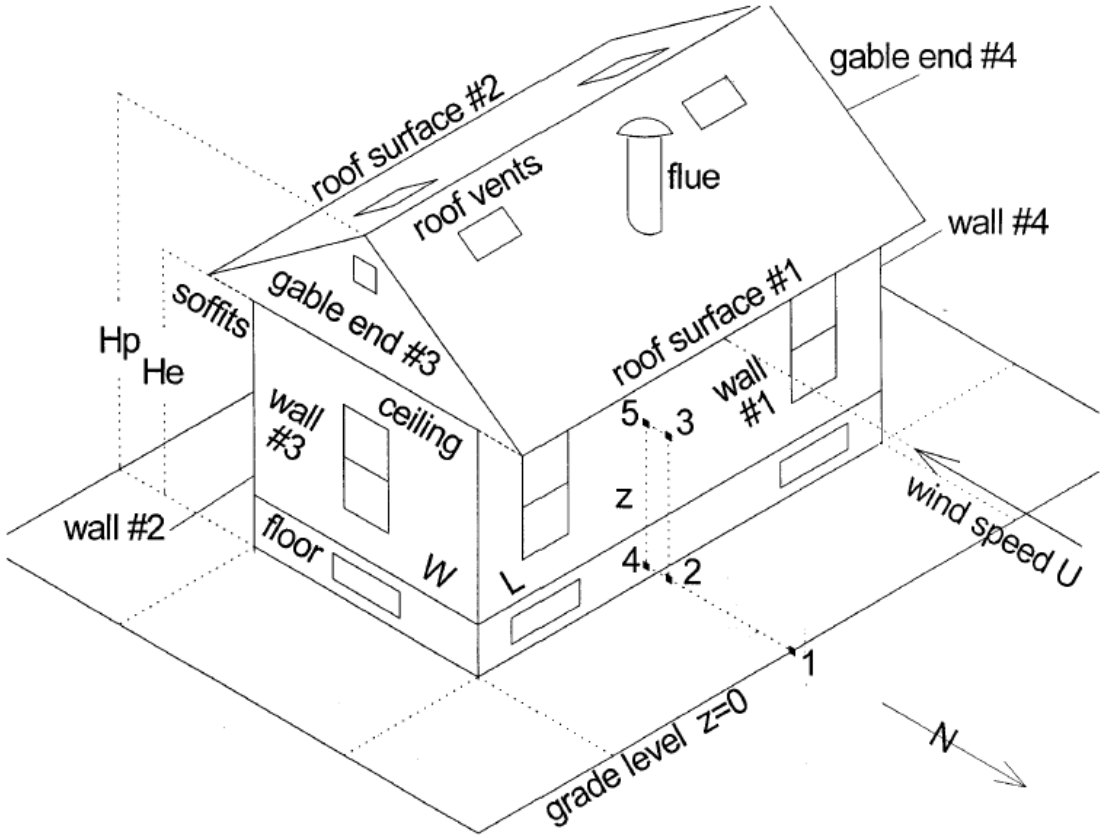

Figure 1. Illustration of house and attic air flow components

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

a user specified length, width and height. The attic has the same floor plan as the house and a pitched roof with soffits and gable ends.

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

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

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

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

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

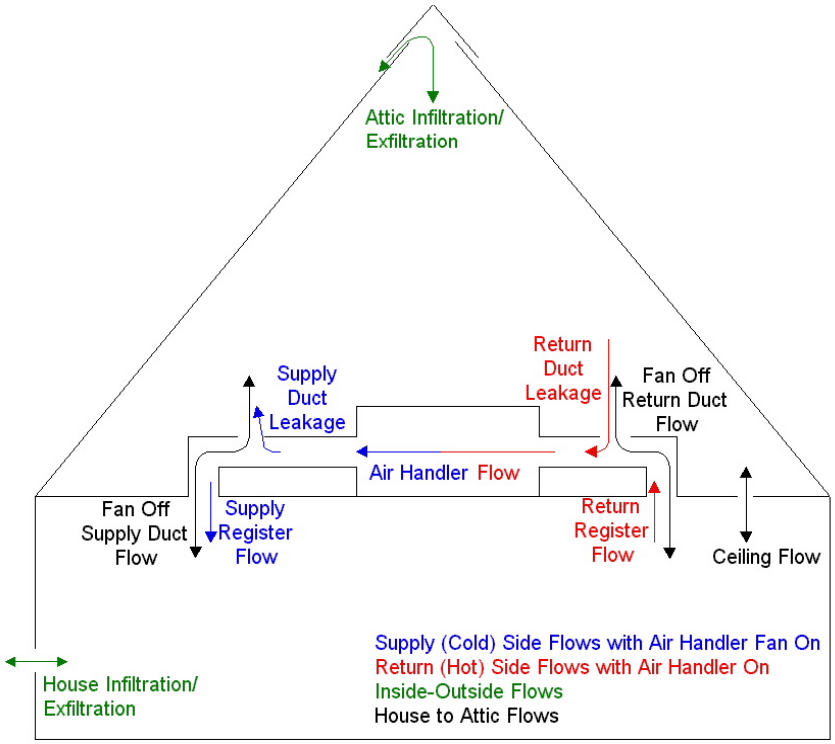

Figure 1: Schematic of duct related air flows (Arrows indicate direction). House and attic air infiltration/exfiltration is the sum of local and distributed leakage.

6.1.1. Ventilation model

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

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

- There is assumed to be no valving action in the building and attic leakage so that flow coefficients are independent of flow direction.

- The building has a rectangular planform. The planform must not have the longest side greater than about three times the shorter side because the wind pressure coefficients used in the model will be incorrect.

- The attic has two pitched roof surfaces and gable ends. This assumption affects the leakage distribution and the pressure coefficients applied to the attic leakage sites.

- The interior of both the house and the attic are well-mixed zones.

- There are no indoor or outdoor vertical temperature gradients, so that the indoor and outdoor air densities are independent of location.

- Air behaves as an incompressible ideal gas. This allows density and viscosity to be functions of temperature only.

- Wall and pitched roof leakage is evenly distributed so as to allow simple integration of height dependent mass flow equations.

- All wind pressure coefficients are averaged over a surface. This means that extremes of wind pressure occurring at corner flow separations are not included.

General flow equation

The general flow equation for each leak is given by:

$$M = \rho C \Delta P^{n} \qquad (1)$$

where M = Mass flow rate [kg/s]

ρ = Density of air flow [Kg/m^3]

C = Flow coefficient [m^3/(sPa^n)]

ΔP= Pressure difference across the leak [Pa]

n = Pressure exponent

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

Neglecting atmospheric pressure changes:

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

where T_{ref} is the absolute reference temperature (K) at which C_{ref} was measured, and T is the temperature of the airflow. For many buildings the distributed background leakage has n~2/3, which means that this correction is unity. For simplicity this temperature correction was

therefore not applied to distributed leakage. For localised leakage sites including furnace flues, passive vents and attic vents n is typically 0.5 and this correction can become significant and therefore it is included in the ventilation calculations.

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

Wind Pressures

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

$$\Delta P_U = \rho_{out} Cp \frac{(S_U U)^2}{2} \qquad (3)$$

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

$$P_U = \rho_{out} \frac{U^2}{2} \qquad (4)$$

and then Equation 3 can be written in terms of P_U:

$$\Delta P_U = Cp\, S_U^2\, P_U \qquad (5)$$

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

Indoor-Outdoor Temperature Difference Pressures

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

$$\Delta P_T(z) = -zg\rho_{out}\left(\frac{(T_{in} - T_{out})}{T_{in}}\right) \qquad (6)$$

where z is the height above a reference (grade level) [m]

g is gravitational acceleration (9.81 [m/s^2]).

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

$$P_T = g\rho_{out}\left(\frac{(T_{in} - T_{out})}{T_{in}}\right) \qquad (7)$$

P_T is the pressure gradient and is multiplied by the height of each leak above grade to find the stack effect pressure difference at that location. Substituting Equation 6 in 5 gives:

$$\Delta P_T(z) = -z\,P_T \qquad (8)$$

Total Pressure Difference

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

$$\Delta P = Cp\,S_U^2\,P_U - z\,P_T + \Delta P_I \qquad (9)$$

Equation 9 is applied to every leak for the building and the attic with the appropriate values of Cp, S_U and z.

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

$$H_{NL,i} = \left(\frac{\Delta P_I + S_{U,i}^2 Cp_i P_U}{P_T} \right) \tag{10}$$

Wind Pressure Coefficients For the house

Wind pressure coefficients are taken from wind tunnel tests and depend on the wind direction. For closely spaced houses in a row the pressure coefficients also change due to the change in flow around the building. Walker and Wilson (1994) discuss these vary in greater detail. Table 1 contains the wall averaged wind pressure coefficients used for the house by the ventilation model for wind perpendicular to the upwind wall. For the closely spaced row, the wind is blowing along the row of houses.

Table A1. Wall averaged wind pressure coefficients for a rectangular building with the wind normal to upwind wall from Akins, Peterka and Cermak (1979) and Wiren (1985).

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

When the wind is not normal to the upwind wall an harmonic trigonometric function is used to interpolate between these normal values to fit the variation shown by Akins, Peterka, and Cermak and Wiren. For each wall of the building the harmonic function for Cp from Walker and Wilson (1994) is used:

$$Cp(\theta) = \frac{1}{2}[(Cp(1) + Cp(2))(\cos^2\theta)^{\frac{1}{4}} + (Cp(1) - Cp(2))(\cos\theta)^{\frac{3}{4}}$$
$$+(Cp(3) + Cp(4))(\sin^2\theta)^2 + (Cp(3) - Cp(4))\sin\theta] \tag{11}$$

where Cp(1) is the Cp when the wind is at 0° (+0.60)

Cp(2) is the Cp when the wind is at 180° (-0.3)

Cp(3) is the Cp when the wind is at 90° (-0.65 or -0.2)

Cp(4) is the Cp when the wind is at 270° (-0.65 or -0.2)

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

This function is shown in Figure 2 together with data from Akins et. al. for a cube. The error bars on the data points in Figure 2 represent the uncertainty in reading the measured values from the figures of Akins, Peterka and Cermak.

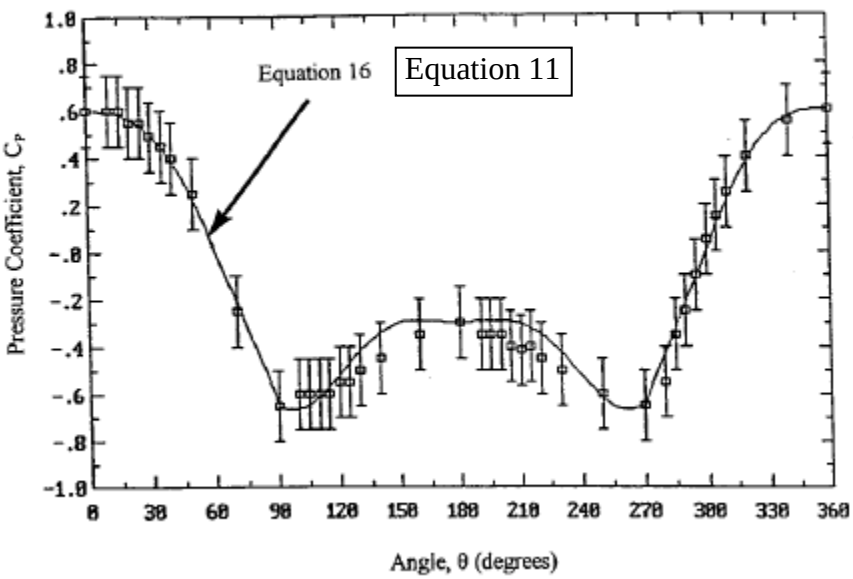

Figure 2. Angular variation in wind pressure coefficient for a rectangular building

Wind Pressure Coefficients For the Attic

The attic simulation model has been developed for a gable end attic with two pitched roof surfaces. The Cp's for gable ends or soffits are assumed to be the same as those on the walls below them and are calculated using the same procedure as for house walls. The pitched roof surfaces have Cp's that are also a function of roof slope. Table 2 gives values of Cp measured by Wiren (1985) for upwind and downwind pitched roof surfaces with wind normal to the upwind surface for different roof pitches. For wind flow parallel to the roof ridge Cp's change in the same way as for houses with Cp = -0.6 for an isolated building and Cp = -0.2 for row houses for both roof pitched surfaces. The Cp is independent of roof pitch for flow parallel to the roof ridge.

Table A2. Pitched roof wind pressure coefficients for wind normal to the upwind surface (Wiren (1985))

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

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

$$Cp(\theta) = \frac{1}{2}[(Cp(1) + Cp(2))\cos^2\theta + (Cp(1) - Cp(2))F$$
$$+(Cp(3) + Cp(4))\sin^2\theta + (Cp(3) - Cp(4))\sin\theta] \qquad (12)$$

where Cp(1) is the Cp when the wind is at 0°

Cp(2) is the Cp when the wind is at 180°

Cp(3) is the Cp when the wind is at 90°

Cp(4) is the Cp when the wind is at 270°

θ is the wind angle measured clockwise from the normal to the roof surface.

F is a switching function to account for changes in roof pitch.

$$F = \frac{1 - (|\cos\theta|)^5}{2}\left(\frac{28 - \psi}{28}\right)^{0.01} + \frac{1 + (|\cos\theta|)^5}{2} \qquad (13)$$

where ψ is the roof pitch in degrees measured from horizontal. Equation 13 acts like a switch with F ~ 1 up to ψ = 28° and F ~ cosθ when ψ > 28°. The switch point of 28° is chosen so that this relationship produces the same results as the wind tunnel data in Table 2. Equation 13 is not used to change the pressure coefficients shown in Table 2, but it changes the functional form of Equation 12 so that the interpolation fits the measured pressure coefficients.

Equation 12 is compared with pitched roof Cp's from Liddament (1986) in Figures 3 through 5 for roof pitches >30°, 10° to 30°, and <10° respectively.

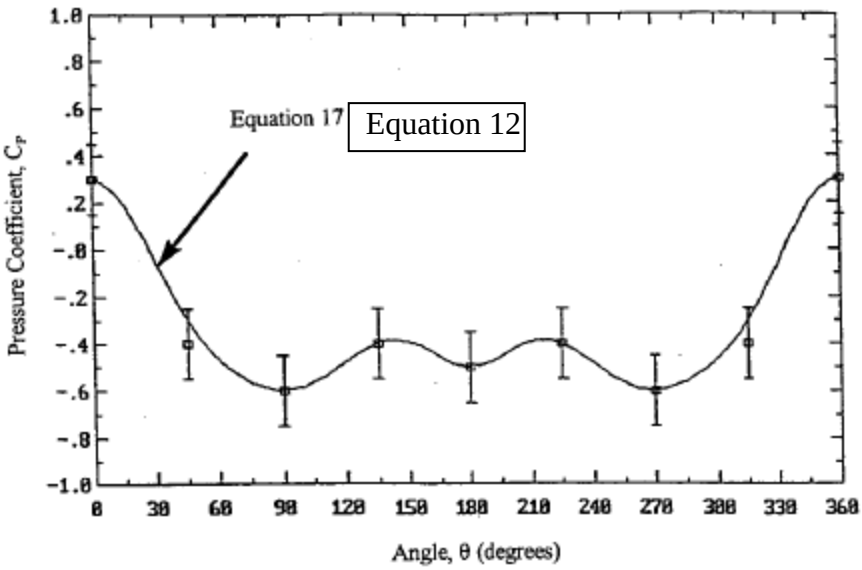

Figure 3. Roof pressure coefficients for a steep sloped roof (pitch > 30°)

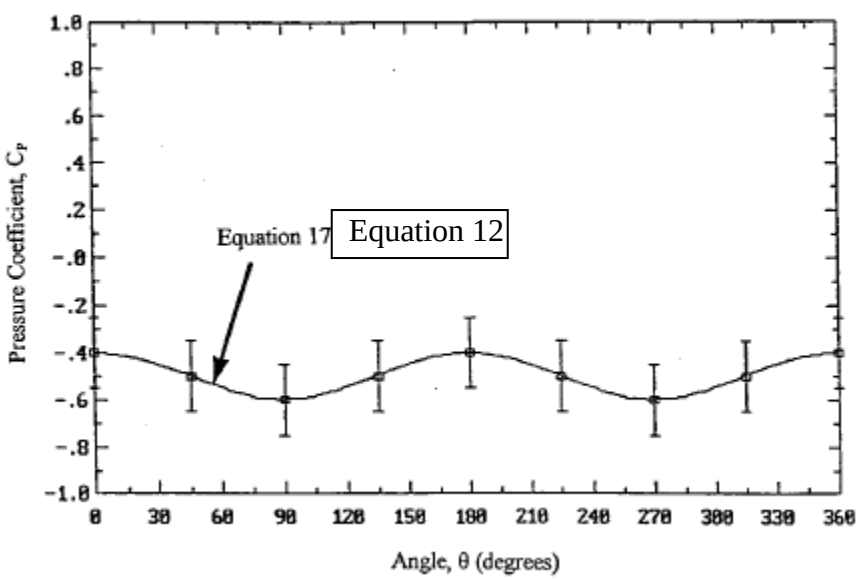

Figure 4. Roof pressure coefficients for a moderate sloped roof (10° < pitch < 30°)

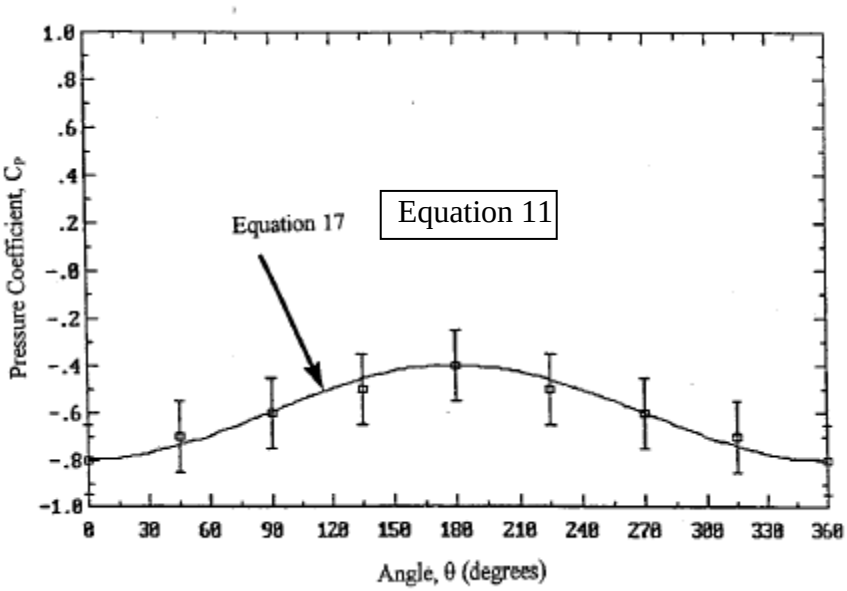

Figure 5. Roof pressure coefficients for a low sloped roof (pitch < 10°)

Wind Shelter

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

$$U_s = S_U U \qquad (14)$$

where U is the free stream wind speed with no sheltering effects.

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

U_S is the effective wind speed used for calculating surface pressures.

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

REGCAP has the following three options for wind shelter.

1. Fixed shelter for all wind directions.

2. Interpolation Function.

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

$$S_U = \frac{1}{2}[(S_U(1) + S_U(2))\cos^2\theta + (S_U(1) - S_U(2))\cos\theta$$
$$(S_U(3) + S_U(4)\sin^2\theta + (SU(3) - S_U(4))\sin\theta] \qquad (15)$$

where S_U is the wind speed multiplier

$S_U(1)$ is the S_U when the wind is at 0°

$S_U(2)$ is the S_U when the wind is at 180°

$S_U(3)$ is the S_U when the wind is at 90°

$S_U(4)$ is the S_U when the wind is at 270°

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

3. Input from data file:

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

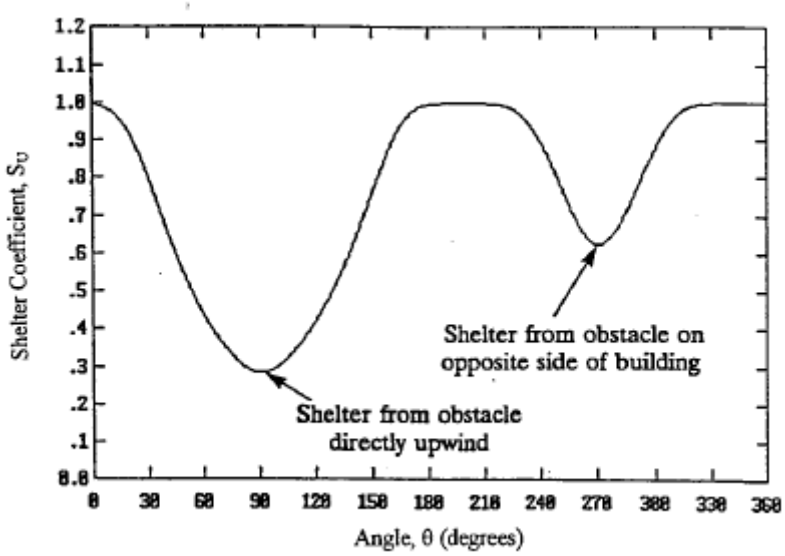

Figure 6. Wind shelter for a typical urban house

Flow Through each Leak for the Attic

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

$$C_{d,a} = \sum_{i=1}^{4} C_{s,i} + C_r \qquad (16)$$

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

Pitched roof Leakage

The two pitched roof surfaces are assumed to have equal leakage. Therefore there is $C_r/2$ leakage in each surface. Cp for the pitched roof surfaces is found using Equation 12 and Table 2. If the surrounding obstacles are taller than the building in question then S_U for the pitched roof surfaces is estimated to be the same as the wall below them, otherwise there is no shelter and S_U=1. For example, a south facing roof pitch would then have the same S_U as calculated for the south facing wall below it. For the attic roof the neutral level, $H_{NL,r}$, is calculated for the two roof pitches using the appropriate Cp and S_U values in Equation 10.

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

$$M_{r,i} = \int dM_{r,i}(z)dz \qquad (17)$$

where

$$dM_{r,i}(z) = \rho\, dC_{r,i}(\Delta P_{r,i}(z))^{n_r} \qquad (18)$$

where $\Delta P_{r,i}(z)$ is given by Equation 9. Assuming evenly distributed leakage allows easy integration over the roof because the fractional leakage $dC_{r,i}$ is given by:

$$dC_{r,i} = C_{r,i}\frac{dz}{(H_p - H_e)} \qquad (19)$$

where H_p is the roof peak height and H_e the eave height. Substituting Equations 19 and 18 in 17 gives

$$M_{r,i} = \frac{\rho C_{r,i}}{(H_p - H_e} \int \Delta P_{r,i}^{n_r} dz \qquad (20)$$

where the limits of integration depend on the neutral level height, $H_{NL,r}$, that is found for each wall using Equation 18.

When $H_{NL,r}$ is on the roof there is flow both in and out of the roof and upon integrating Equation 20 the masses flowing in and out are kept separate. This important for the total mass balance and for keeping track of all the flows through the building envelope. There are several different cases of flow through the pitched roof surfaces depending on the location of $H_{NL,r}$, T_a

and T_{out}. The pressure differences at the eave height, ΔP_e, and at the roof peak, ΔP_p, are defined as follows and are convenient to use when calculating the mass flow rates.

$$\Delta P_p = \Delta P_{I,a} + S_U^2 Cp\, P_U - H_p P_{T,a} \tag{21}$$

$$\Delta P_e = \Delta P_{I,a} + S_U^2 Cp\, P_U - H_e P_{T,a} \tag{22}$$

An example case given by Equations 23 and 24 is for $T_a > T_{out}$ with $H_{NL,r}$ somewhere on the pitched roof surface between the eave height, H_e, and the peak height H_p. There is two way flow through the roof surface in this case with flow in below $H_{NL,r}$ and flow out above $H_{NL,r}$:

$$M_{r,out} = \frac{\rho_a \frac{C_r}{2} \Delta P_p^{(n_r+1)}}{(H_p - H_e) P_{T,a} (n_r + 1)} \tag{23}$$

$$M_{r,in} = \frac{\rho_{out} \frac{C_r}{2} \Delta P_e^{(n_r+1)}}{(H_p - H_e) P_{T,a} (n_r + 1)} \tag{24}$$

Soffit and Gable Leakage

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

$$\Delta P_{s,i} = \Delta P_{I,a} + Cp_i \;\; S_{U,i}^2 \;\; P_U - H_s P_{T,a} \tag{25}$$

Attic Vent Leakage

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

$$\Delta P_{V,a} = \Delta P_{I,a} + S_{U,V}^2 \; Cp_V \; P_U - H_V \; P_{T,a} \qquad (26)$$

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

Attic Floor Leakage

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

Ventilation Fans in Attics

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

$$\Delta P_{fan,a} = \Delta P_{I,a} + S_{U,fan}^2 \, Cp_{fan} \, P_U - H_{fan} \, P_{T,a} \qquad (27)$$

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

$$M_{fan,a} = \rho Q_{rated} \left(\frac{\Delta P_{rated} + \Delta P_{fan,a}}{\Delta P_{rated}} \right)^{p_{fan}} \qquad (28)$$

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

Duct Leaks with air handler off (M_{soff} and M_{roff})

Both the supply and return leaks have the same pressure difference as the attic floor/house ceiling. The supply leakage pressure exponent is a required input, but typically a value of 0.6 is used. The flow coefficient is calculated from the leakage air flow rate, assuming a reference pressure of 25 Pa and using the pressure exponent:

$$C_{soff} = \frac{Q_{ah} \alpha_s}{25^{n_s}} \qquad (29)$$

$$C_{roff} = \frac{Q_{ah} \alpha_r}{25^{n_r}} \qquad (30)$$

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

is the return leak flow coefficient, n_r is the return leak pressure exponent and α_r is the return leakage expressed as a fraction of air handler flow.

Duct leaks with air handler on (M_{son} and M_{ron})

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

6.1.2. Flow through Each Leak for the House

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

$$C_d = \sum_{i=1}^{4} C_{f,i} + \sum_{i=1}^{4} C_{w,i} + C_c \qquad (31)$$

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

Furnace Flues and Fireplaces

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

$$Cp_F = (-0.5)\left(\frac{H_F}{H_e}\right)^{2p} \qquad (32)$$

where H_f is the flue top height and p is the exponent used in the atmospheric boundary layer wind profile (typically p=0.3 for urban surroundings and p=0.17 for rural sites). Shelter for the flue, $S_{U,F}$, is the shelter factor at the top of the flue. If the surrounding buildings and other obstacles are below the flue height then it is assumed that $S_{U,F} = 1$. If the surrounding obstacles are higher than the flue then the flue is sheltered and $S_{U,F}$ is calculated using Equation 15. The general pressure difference Equation 8 can be written specifically for the furnace flue as:

$$\Delta P_F = \Delta P_I - P_T \;\; H_F + P_U \;\; S_{U,F}^2 \;\; Cp_F \qquad (33)$$

and the mass flow rate, M_F, for the flue is given by Equation 1. Note that this is for an unheated flue or a natural draft furnace (flue without a draft inducing fan) well connected to the conditioned space. In new construction most furnace flues will be outside conditioned space in a well vented closet, garage or attic (or will be direct vented), in which case the flue leak is set to zero and only open fireplaces need to be considered, or we need to know the flow rate through the forced combustion fan for furnaces.

Floor Level Leakage

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

a. Basements and Slab on Grade

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

$$\Delta P_{f,i} = \Delta P_I + Cp_i \; S_{U,i}^2 \; P_U - H_f \; P_T \qquad (34)$$

where floor height, H_f, is measured from grade level. For a house with a basement this is the height of the main level floor above grade and the leakage coefficient, $C_{f,i}$ includes the leakage around basement windows, dryer vents etc. The mass flow rate for these floor level leaks is given by Equation 1.

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

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

$$Cp_f = \sum_{i=1}^{4} S_{U,i}^2 Cp_i \left(\frac{L_i}{L_\pi} \right) \qquad (35)$$

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

$$\Delta P_f = \Delta P_I + Cp_f P_U - H_f P_T \qquad (36)$$

and the mass flow rate through the crawlspace leakage is given by Equation 1.

Ceiling Leakage

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

$$\Delta P_c = \Delta P_I - \Delta P_{I,a} - \rho_{out} g H_e \left(\frac{T_{in} - T_{out}}{T_{in}} - \frac{T_a - T_{out}}{T_a} \right) \qquad (37)$$

The mass flow rate through the ceiling is given by Equation 1.

Wall Leakage

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

Fan Flow

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

Vent Leakage

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

Flow through open Doors and Windows

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

$$P_b = Cp\, S_U^2 U^2 - 2g\, H_b \left(\frac{T_{in} - T_{out}}{T_{in}} \right) + \frac{2\Delta P_I}{\rho_{out}} \qquad (38)$$

$$P_t = Cp\, S_U^2 U^2 - 2g\, H_t \left(\frac{T_{in} - T_{out}}{T_{in}} \right) + \frac{2\Delta P_I}{\rho_{out}} \qquad (39)$$

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

H_b = Height above grade of the bottom of the opening

H_t = Height above grade of the top of the opening

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

$$M_{out} = (\rho_{out}\rho_{in})^{\frac{1}{2}} \frac{KWT_{in}}{3g(T_{in} - T_{out})} P_t^{\frac{3}{2}} \qquad (40)$$

$$M_{in} = \rho_{out} \frac{KWT_{in}}{3g(T_{in} - T_{out})} P_b^{\frac{3}{2}} \qquad (41)$$

Window and Door Flow Coefficient, K

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

$$K = 0.400 + 0.0045\,|T_{in} - T_{out}| \qquad (42)$$

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

Grille Air Flows

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

6.1.3. Air Flow Solution Method

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

$$\sum M = M_F + M_f + M_c + M_{\sup} + M_{ret} + M_{son} + M_{soff} + \sum_{i=1}^{4} M_{ws,i} + M_v + M_{fan} = 0 \quad (43)$$

where the various mass flows are:

M_F : Flue

M_f : floor level leaks

M_c : Ceiling

M_{sup} : supply register air handler on

M_{ret} : return register air handler on. For CFI systems the CFI supply flow is subtracted from M_{ret}.

M_{soff} : supply register air handler off

M_{roff}: return register air handler off

M_V : sum of all passive vent flows

M_{fan} : is the sum of all the ventilation fans.

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

For the attic the mass balance equation is given by

$$\sum M = M_r + M_c + M_{soff} + M_{roff} + M_{son} + M_{soff} + \sum_{i=1}^{4} M_{s,i} + M_{v,a} + M_{fan,a} = 0 \qquad \textbf{(44)}$$

where

M_r : sum of the in and the out flows through the pitched roof surfaces,

Mc : ceiling

M_{soff} : supply leak air handler off

M_{roff}: return leak air handler off

M_{son} : supply leak air handler on

M_{ron} : return leak air handler on

$M_{s,i}$ flow through soffit (or gable) component i.

$M_{fan,a}$: sum of the mass flows through all the attic fans

$M_{V,a}$: sum of the flows through all the attic vents.

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

Zone Coupling

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

6.1.4. Heat Transfer Model

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

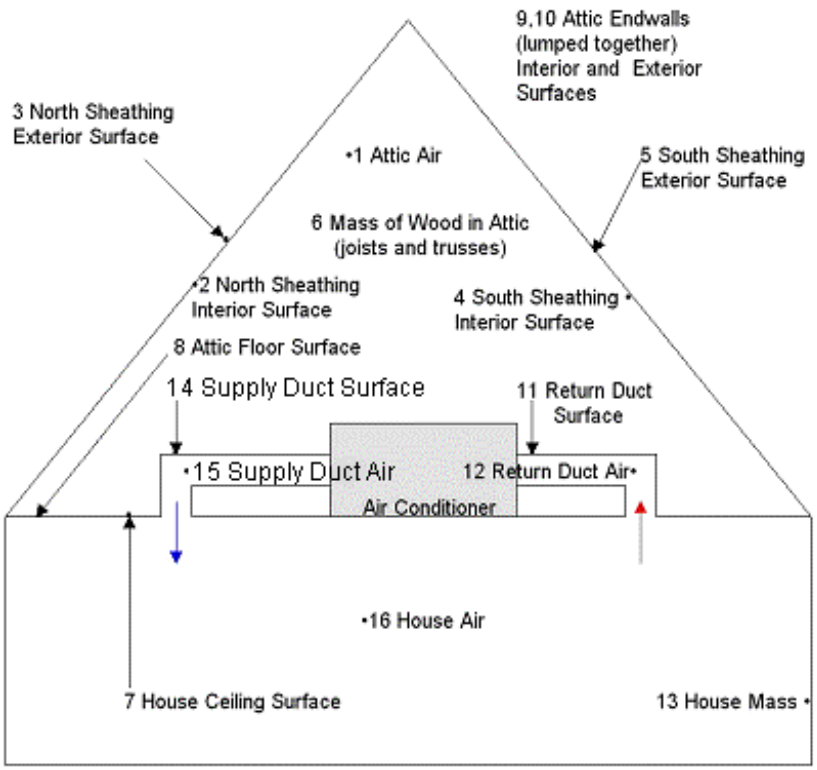

Figure 7: Nodes For Heat Transfer Model

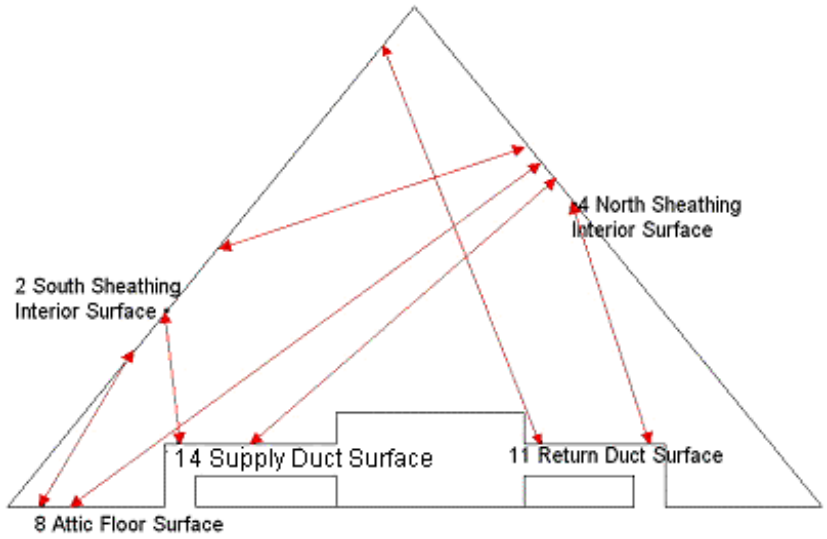

Figure 8: Radiation Transfer for Ducts in Attic

Attic and duct system heat transfer nodes:

1 = Attic Air

2 = Inner Surface1 Sheathing

3 = Outer Surface1 Sheathing

4 = Inner Surface2 Sheathing

5 = Outer Surface2 Sheathing

6 = All of the wood (joists, trusses, etc) lumped together

7 = Ceiling of the house

8 = Floor of the attic

9 = Inner End Wall

10 = Outer End Wall

11 = Return Duct Outer Surface

12 = Return Duct Air

13 = Mass of the house

14 = Supply Duct Outer Surface

15 = Supply Duct Air

16 = House Air

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

$$\rho_i V_i C_{sh,i} \frac{dT_i}{dt} = \sum q \qquad (45)$$

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

$$\rho_i V_i C_{sh,i} \frac{T_i^{\,j} - T_i^{\,j-1}}{\tau} = \sum q \qquad (46)$$

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

Radiation Heat Transfer

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

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

$$q_i = A_i h_{R,i-j}\left(T_i - T_j\right) + A_i h_{R,i-k}\left(T_i - T_k\right) \qquad (47)$$

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

$$h_{R,i-j} = \frac{\sigma\left(T_i + T_j\right)\left(T_i^2 + T_j^2\right)}{\dfrac{1-\varepsilon_i}{\varepsilon_i} + \dfrac{1}{F_{i-j}} + \dfrac{\left(1-\varepsilon_j\right)A_i}{\varepsilon_j A_j}} \qquad (48)$$

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

The emissivity of surfaces found in building construction is given by ASHRAE (1989)(Chapter 37). For the inside sheathing surfaces a typical value for wood is $\varepsilon = 0.90$ and for

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

Solar Radiation (Nodes 3 and 5)

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

$$q_R = A\alpha G \qquad (49)$$

where q_R is radiation heat transfer rate [W]

A = Surface area [m^2]

α = Surface absorbtivity

G = Total Solar Radiation [W/m^2], both direct and diffuse.

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

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

In addition to the daytime solar gain the outside of the pitched roof sheathing has low temperature long wave radiant exchange with the sky and the ground. This exchange is responsible for cooling of the sheathing at night as it radiates energy to the cooler sky. On a cloudy night the cooling of the sheathing is reduced because the radiation exchange is with clouds that are warmer than the sky temperature. Both the clouds and the ground are assumed to be at the outside air temperature. The view factors that account for the proportion of sky, cloud or ground seen by the pitched roof surface are from Ford (1982). Cloud cover is taken from the WYEC2 CEC ACM weather data files (Total Sky Cover).

The net radiation exchange for exterior pitched roof sheathing surfaces has the same form as Equations 41 and 42 for the internal radiation because this is a three body problem involving the roof surface, the sky and the ground and the clouds (which are assumed to be at the same temperature). The sky temperature T_{sky} depends on the water vapour pressure in the air. The view factors give the fraction of exposure to the ground (and clouds) and the sky for the

pitched roof surfaces. Using the same view factors for both pitched roof surfaces assumes that the cloud cover is uniformly distributed over the sky.

Effective Sky Temperature for Radiation

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

$$T_{sky} = T_{out}\left(0.55 + 5.68\times10^{-3}\sqrt{P_v}\right)^{0.25} \qquad (50)$$

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

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

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

Convection Heat Transfer

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

$$q_T = h_T A \Delta T \qquad (51)$$

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

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

A is the surface area

ΔT is the temperature difference

$$h_T = 3.2\left(\Delta T\right)^{\frac{1}{3}} \qquad (52)$$

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

The house ceiling uses a convection heat transfer coefficient of 6 W/m²Kwith the air handler off (based on values in ASHRAE Fundamentals 200, Chapter 3) and 9 W/m²K with the air handler on (based on typical indoor air velocities). These same heat transfer coefficients are used for the exterior surfaces of ducts when they are inside the conditioned space.

Forced Convection

Forced convection heat transfer is calculated using:

$$h_{forced} = (18.192 - 0.037T_{film})U^{0.8} \qquad (53)$$

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

$$U_{atticconvection} = \frac{(Matticenvin - Matticenvout)}{\rho_{attic} 4Al_4} \qquad (54)$$

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

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

6.1.5. Equipment Capacity

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

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

Refrigerant charge effects

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

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

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

Figures 9 and 10 represent the measured data from Rodriguez and the model in the above tables. The “old” model was based on a previous analysis of laboratory data and is not currently used in REGCAP.

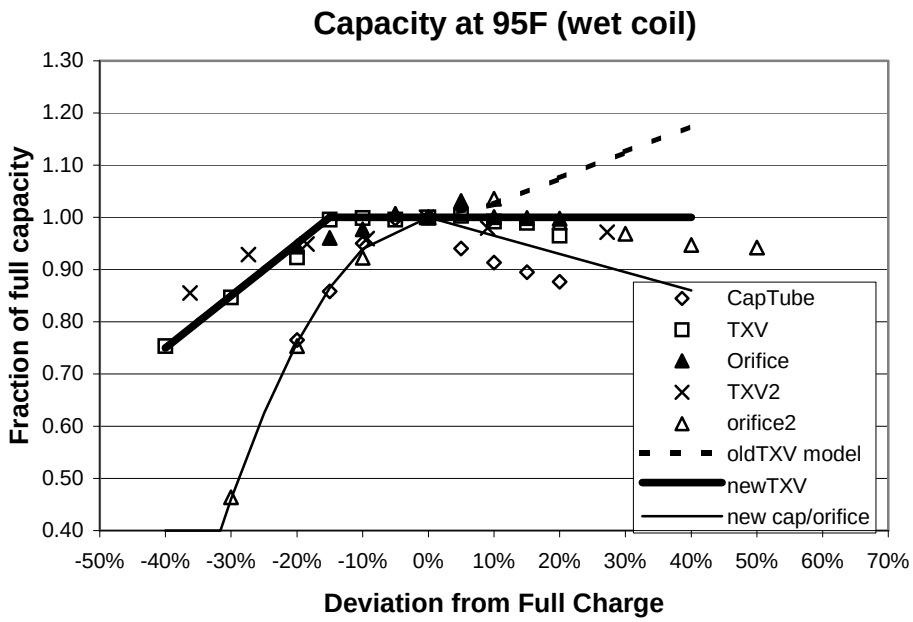

Figure 9. Wet Coil capacity variation with charge

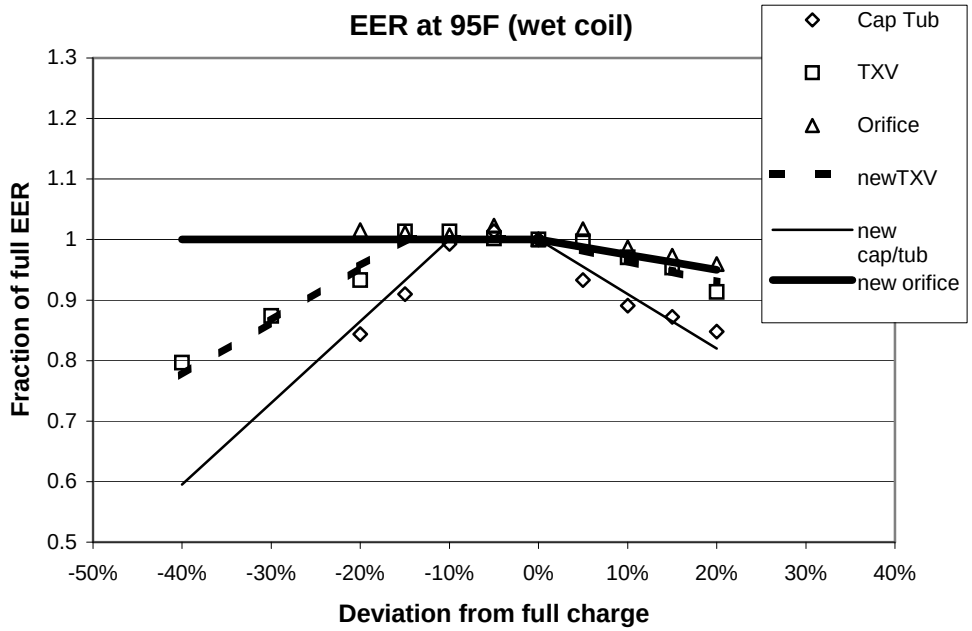

Figure 10. Wet Coil EER variation with charge

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

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

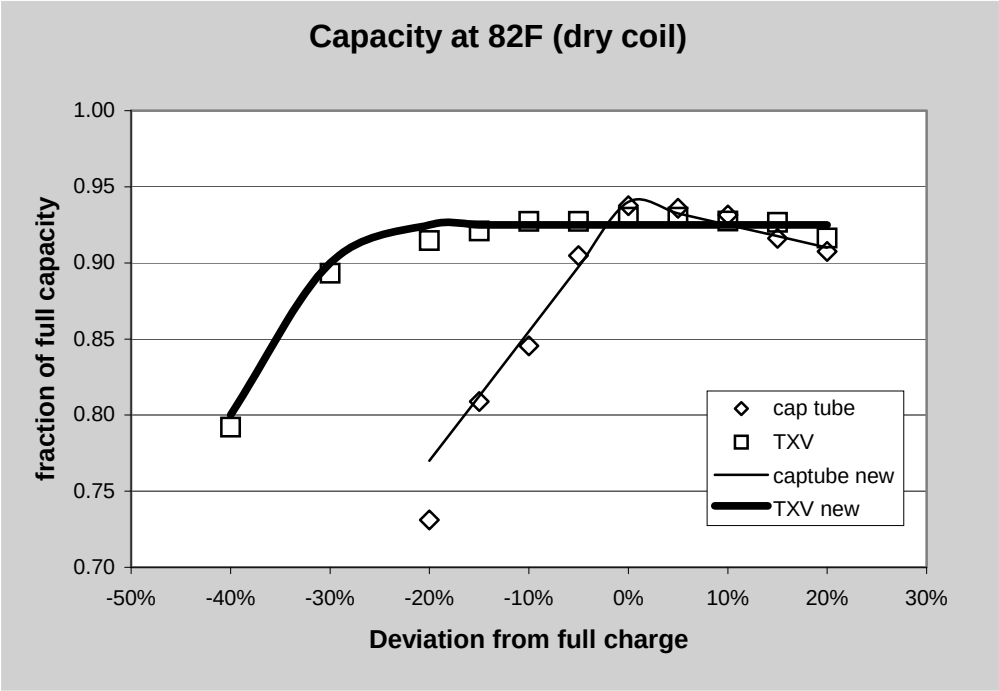

Figure 11. Dry Coil capacity variation with charge

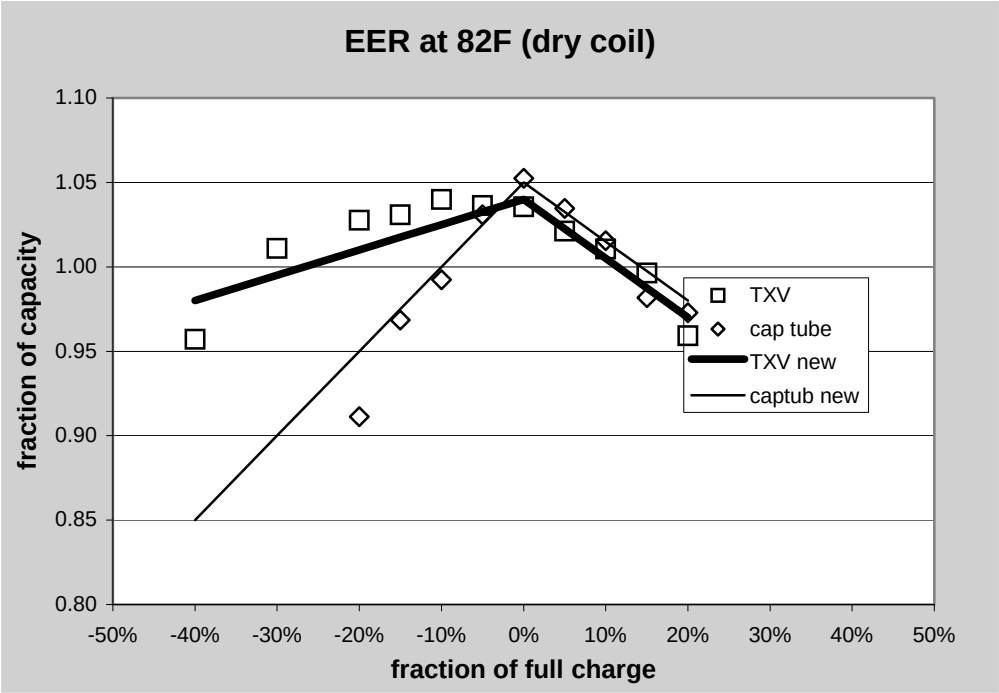

Figure 12. Dry Coil EER variation with charge

Outdoor air temperature

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

- Capacity:

  Correction = (-0.00007)*(T-Tref)^2-0.0067*(T-Tref)+1

- EER:

  Correction = (-0.00007)*(T-Tref)^2-0.0085*(T-Tref)+1

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

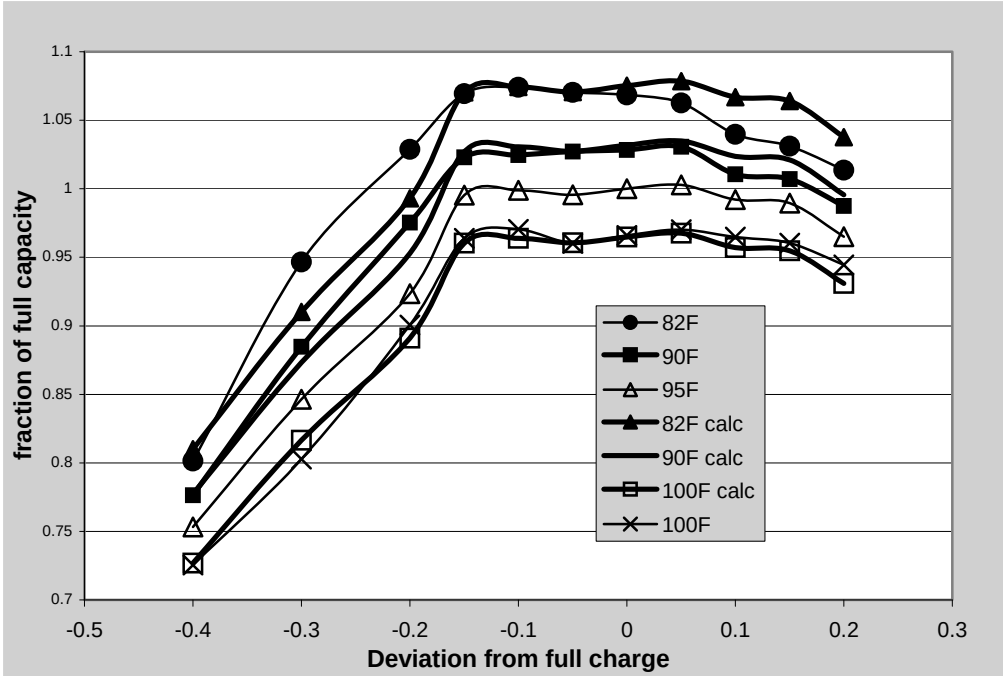

Figure 13. Dry Coil capacity variation with outdoor temperature

Air handler flow

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

- for TXV:

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

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

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

6.1.6. Node Heat Transfer Equations

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

Node 1. Attic Air

The attic air has convective (the h_U terms) heat transfer from all the interior attic surfaces - nodes 2, 4, 8, 6 and 9 as shown Figure 7. Although each convection term uses the same velocity, U_U, the different temperatures will change the film temperature, T_ε, and thus the heat transfer coefficient. In addition the convective flows in and out of the attic, M_a, and the flow through the ceiling, M_c, duct leakage and duct leak air handler off flows transport heat in and out of the attic air.

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

These nodes all experience internal conduction with surface convection and radiation. The differences are that the exterior sheathing surfaces have daytime solar gains and nightime radiation cooling.

The areas of nodes 3 and 5 (exterior surfaces) are increased by 50% for tile roofs.

Node 6. Attic Joists and Trusses

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

Nodes 7 and 8. House Ceiling/Attic Floor

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

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

Node 11. Return duct external duct surface

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

Node 12. Return Duct Air

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

Node 13. House mass

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

Node 14. Supply duct external duct surface

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

Node 15. Supply Duct Air

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

Node 16. House Air

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

coefficient and the solar gain through the windows in each of the four cardinal directions. Any internal loads go directly to the house air.

Envelope load

$$Load = UA\left(t_{solair} - t_{in}\right) + 0.05 q_{solgain} \qquad (55)$$

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

$$t_{solair} = t_{out} + 0.03 q_{incidentsolar} \qquad (56)$$

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

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

6.1.7. Solution of the Attic Heat Transfer Equations

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

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Appendix B: Latitude and Altitude Taken from ACM Joint Appendix

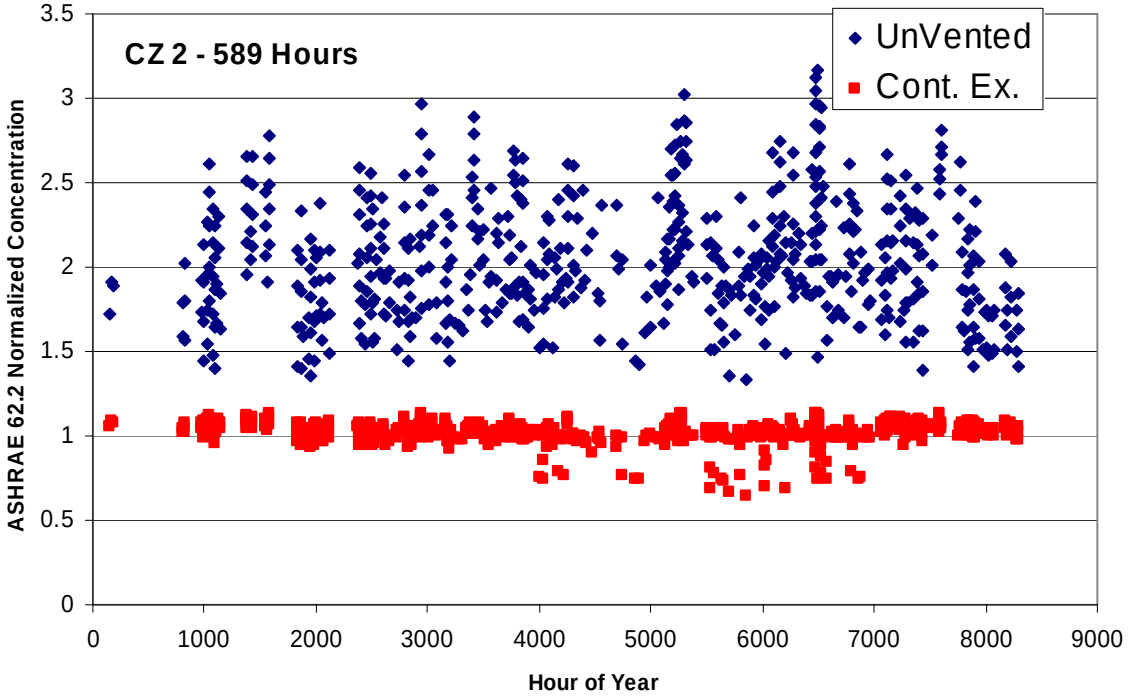
Table B1–California Climate Zone Summary

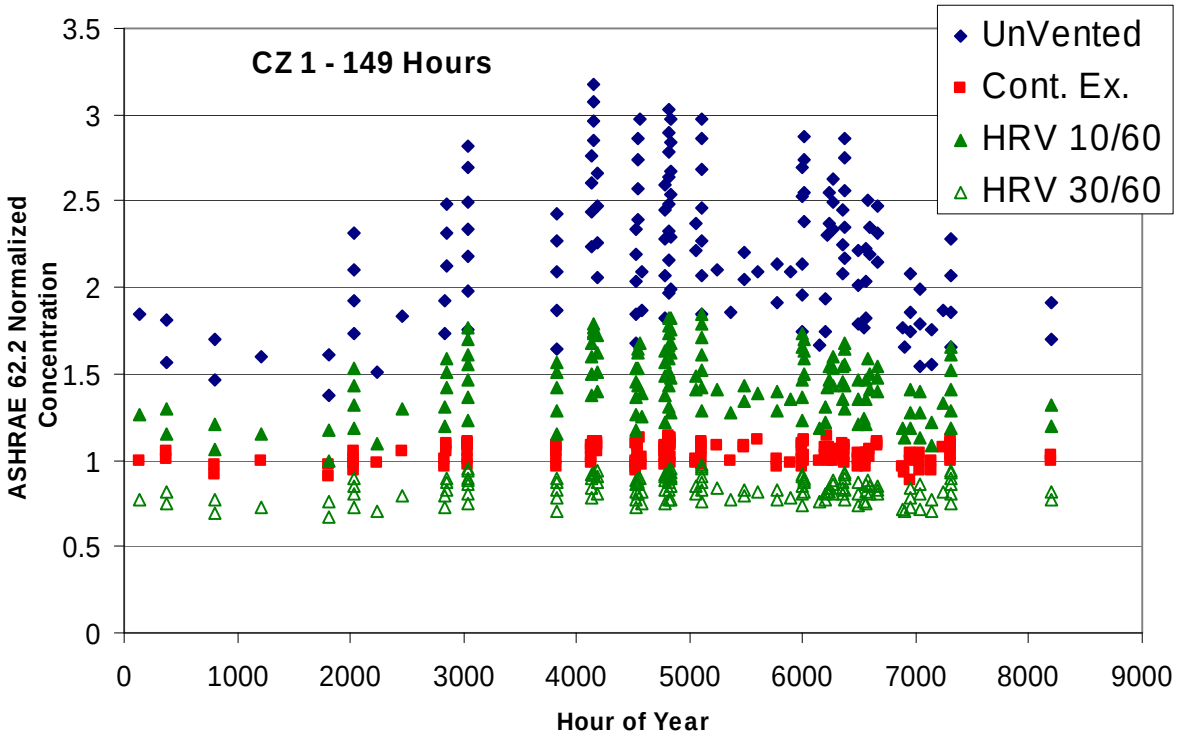
Climate Zone	City	Latitude	Longitude	Elevation
1	Arcata	40.8	124.2	43
2	Santa Rosa	38.4	122.7	164
3	Oakland	37.7	122.2	6
4	Sunnyvale	37.4	122.4	97
5	Santa Maria	34.9	120.4	236
6	Los Angeles AP	33.9	118.5	97
7	San Diego	32.7	117.2	13
8	El Toro	33.6	117.7	383
9	Burbank	34.2	118.4	655
10	Riverside	33.9	117.2	1543
11	Red Bluff	40.2	122.2	342
12	Sacramento	38.5	121.5	17
13	Fresno	36.8	119.7	328
14	China Lake	35.7	117.7	2293
15	El Centro	32.8	115.6	-30
16	Mt. Shasta	41.3	122.3	3544

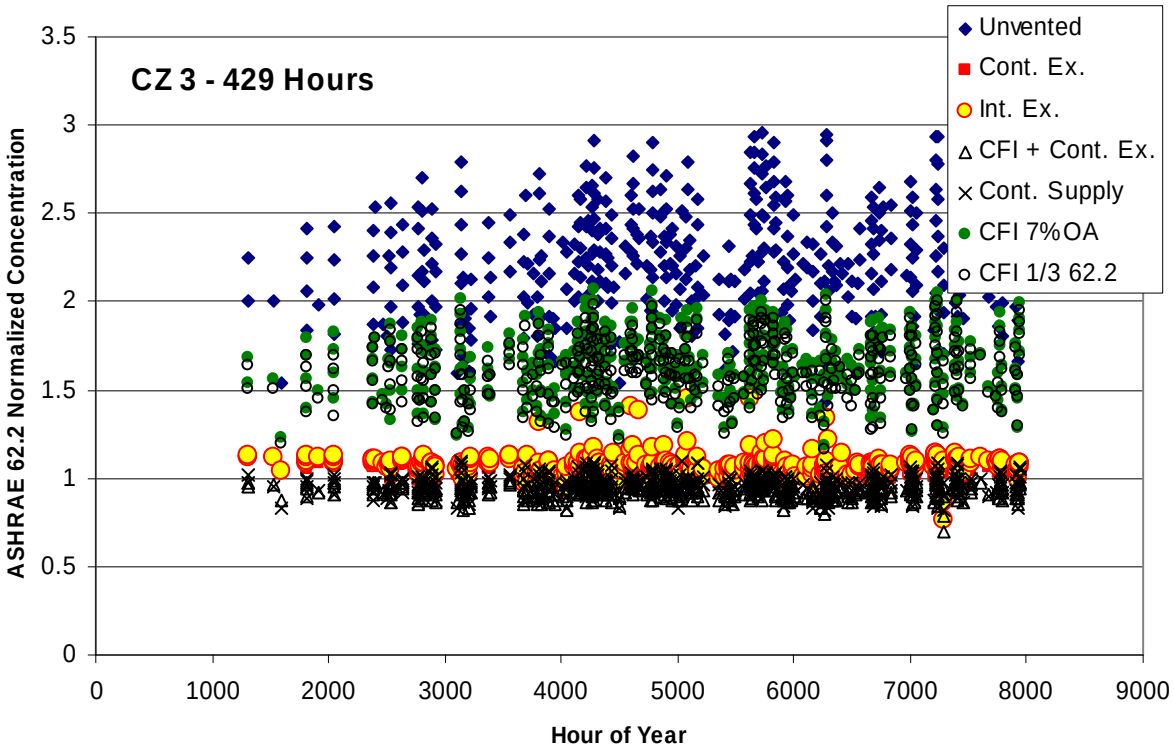
Appendix C: Summary of Sizing Based on Chitwood Field Data

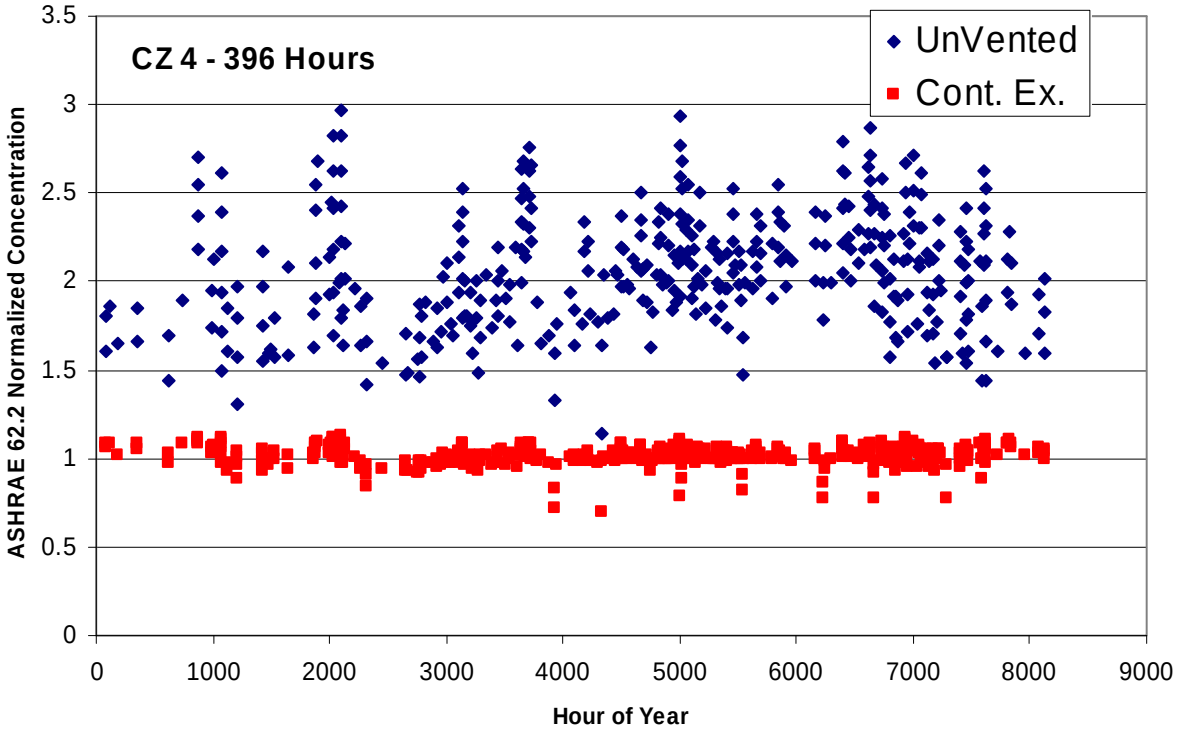
Location		Cooling Sizing	Heating Sizing	Cooling/Heating Ratio
		tons/1000ft^2	kBtu/1000ft^2	tons/100kBtu
ALL CZ	average	2.2	45.8	4.9
	max	5.0	113.2	7.5
	min	0.8	20.1	2.8
	sdev	0.7	16.9	1.0
	sdev%	31.9	37.0	19.8
CZ11	average	1.8	39.3	4.6
	max	2.6	56.3	6.0
	min	0.8	20.1	3.2
	Sdev%	33.6	31.1	13.7
CZ12	average	1.6	40.5	3.9
	Max	2.0	55.5	5.0
	Min	1.1	30.9	2.8
	Sdev%	22.0	21.2	20.9
CZ8	average	2.0	32.6	6.5
	Max	2.1	45.7	7.5
	Min	1.8	24.2	4.4
	Sdev%	7.2	35.1	27.9
CZ15	average	2.9	61.7	4.9
	Max	5.0	113.2	5.8
	Min	2.2	38.0	3.0
	Sdev%	25.7	37.4	15.9
CZ10	average	2.1	33.0	6.3
	Max	2.1	35.1	6.7
	Min	1.9	31.9	5.6
	Sdev%	5.2	5.4	10.2
CZ13	average	2.3	46.6	5.0
	Max	2.6	58.7	6.0
	Min	1.5	26.2	4.0
	Sdev%	13.2	19.2	15.4

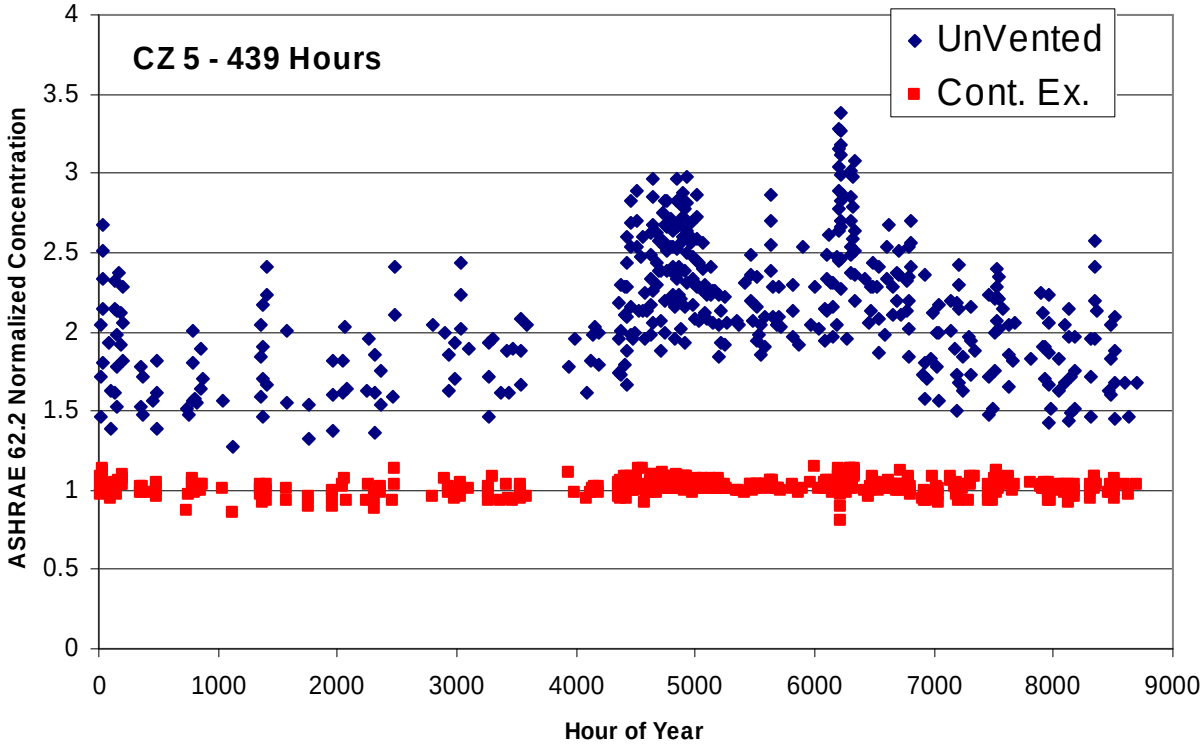
Appendix D. Results of Low Ventilation Rate Indoor Concentration Calculations

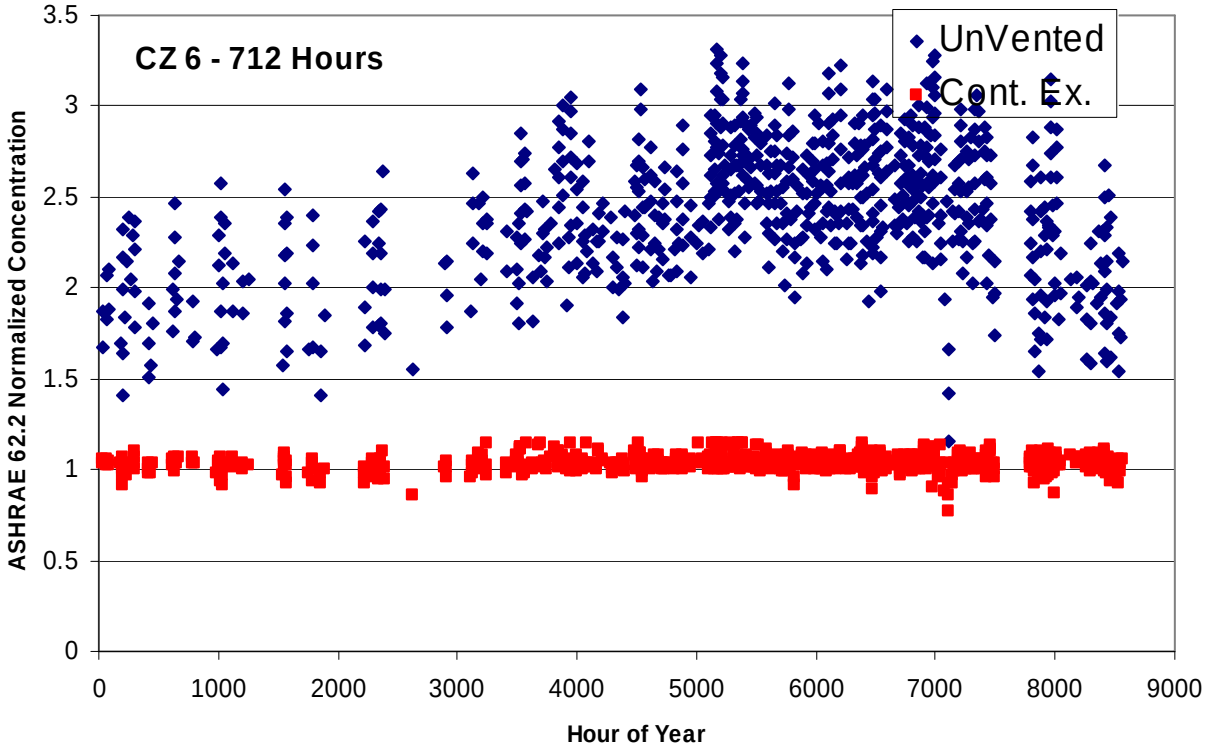


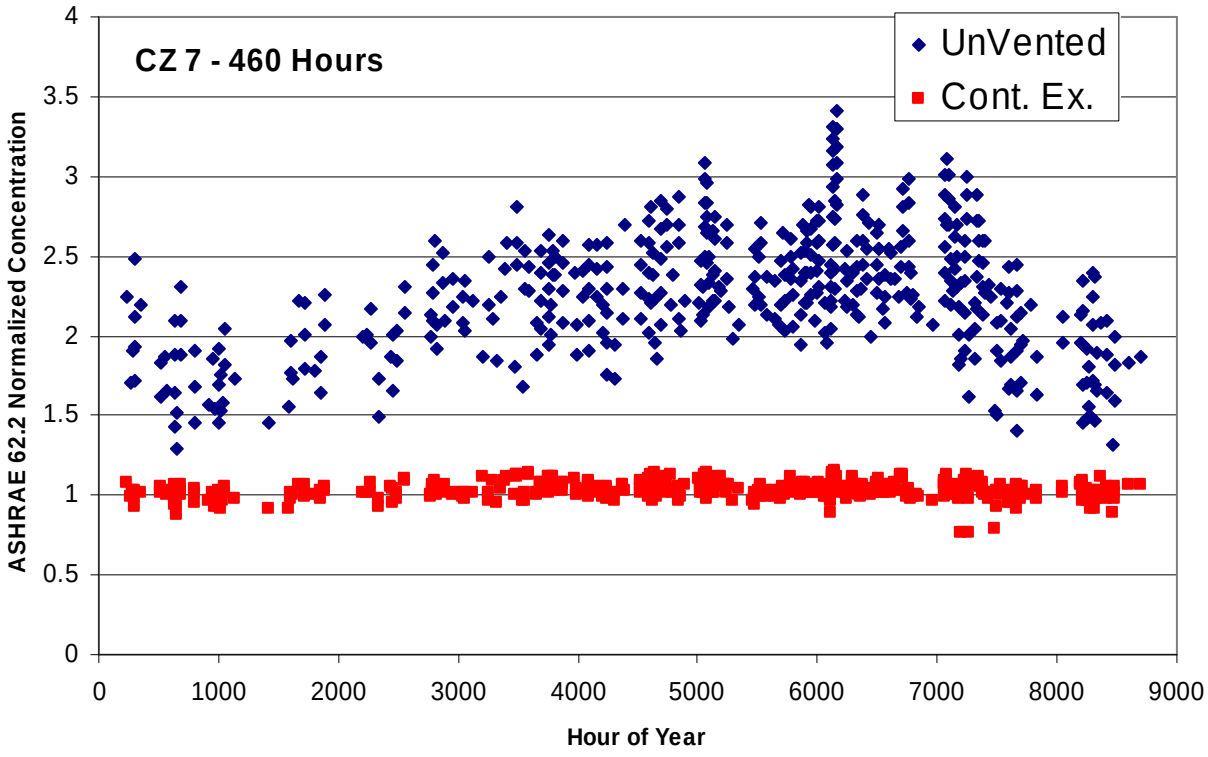


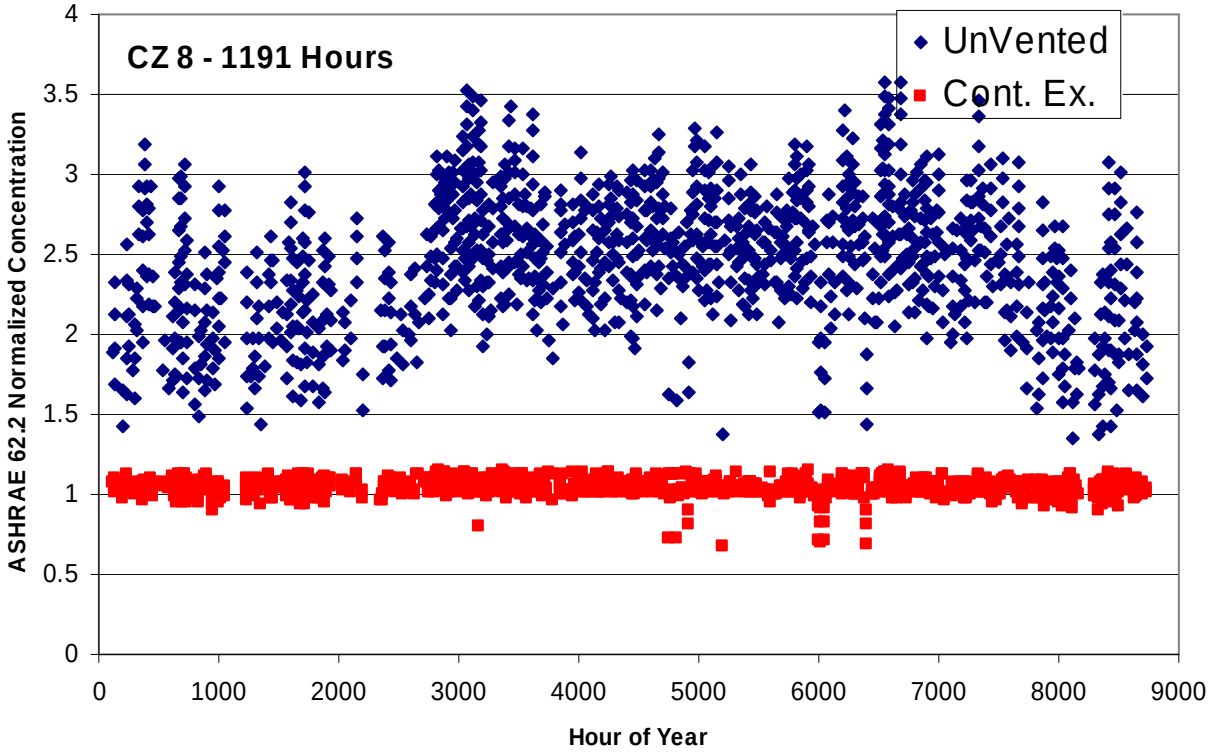

CZ 3 - 429 Hours
Unvented
Cont. Ex.
Int. Ex.
CFI + Cont. Ex.
Cont. Supply
CFI 7%OA
CFI 1/3 62.2
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

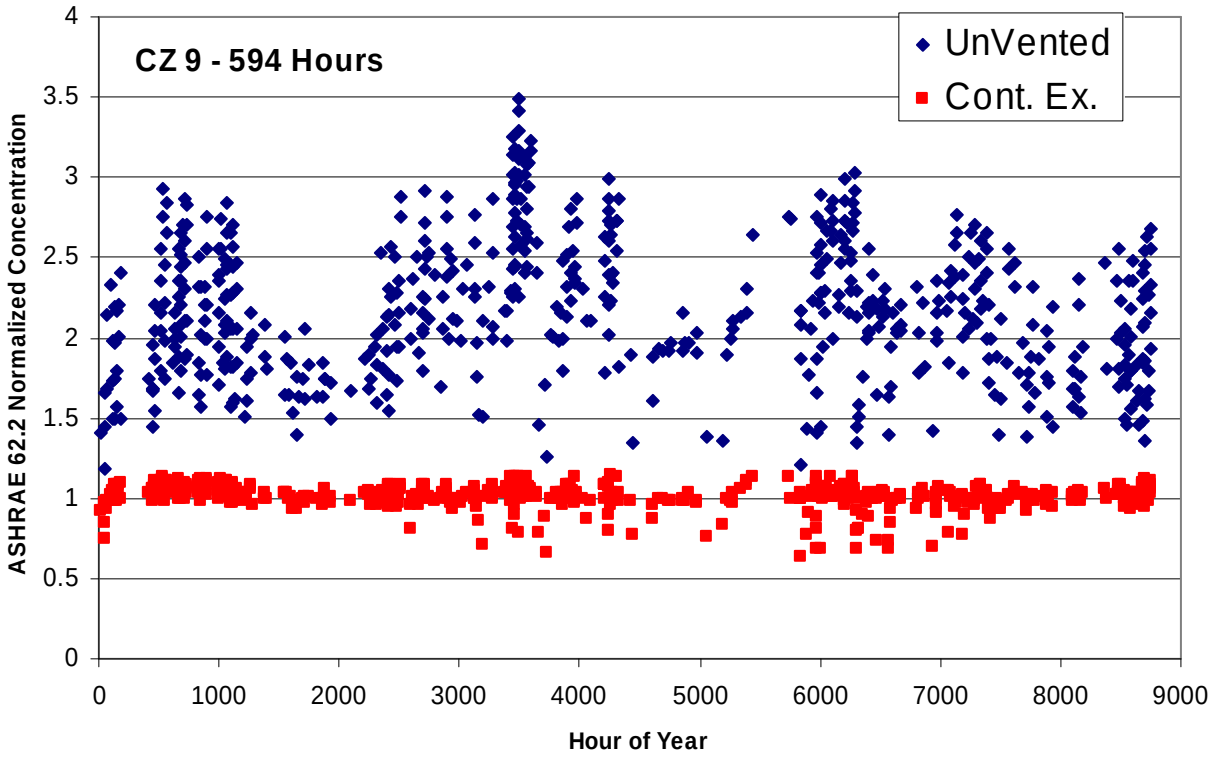

CZ 4 - 396 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

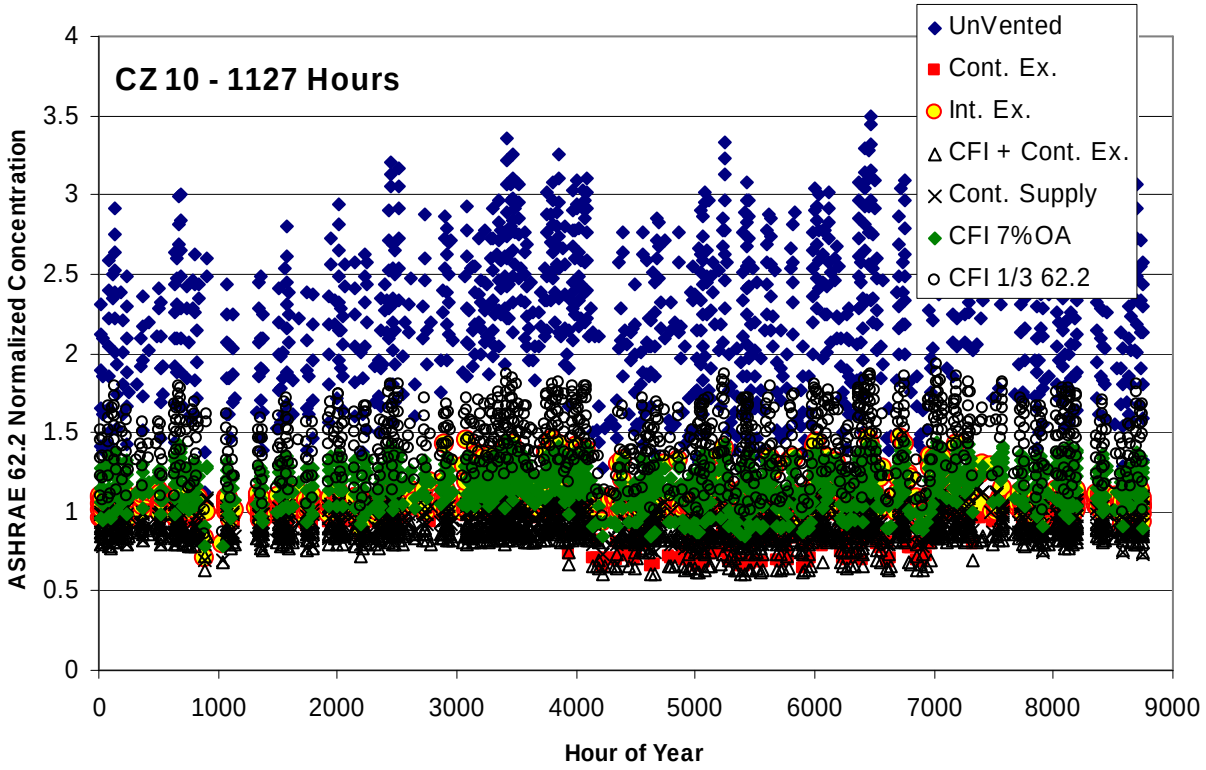

CZ 5 - 439 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
4
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

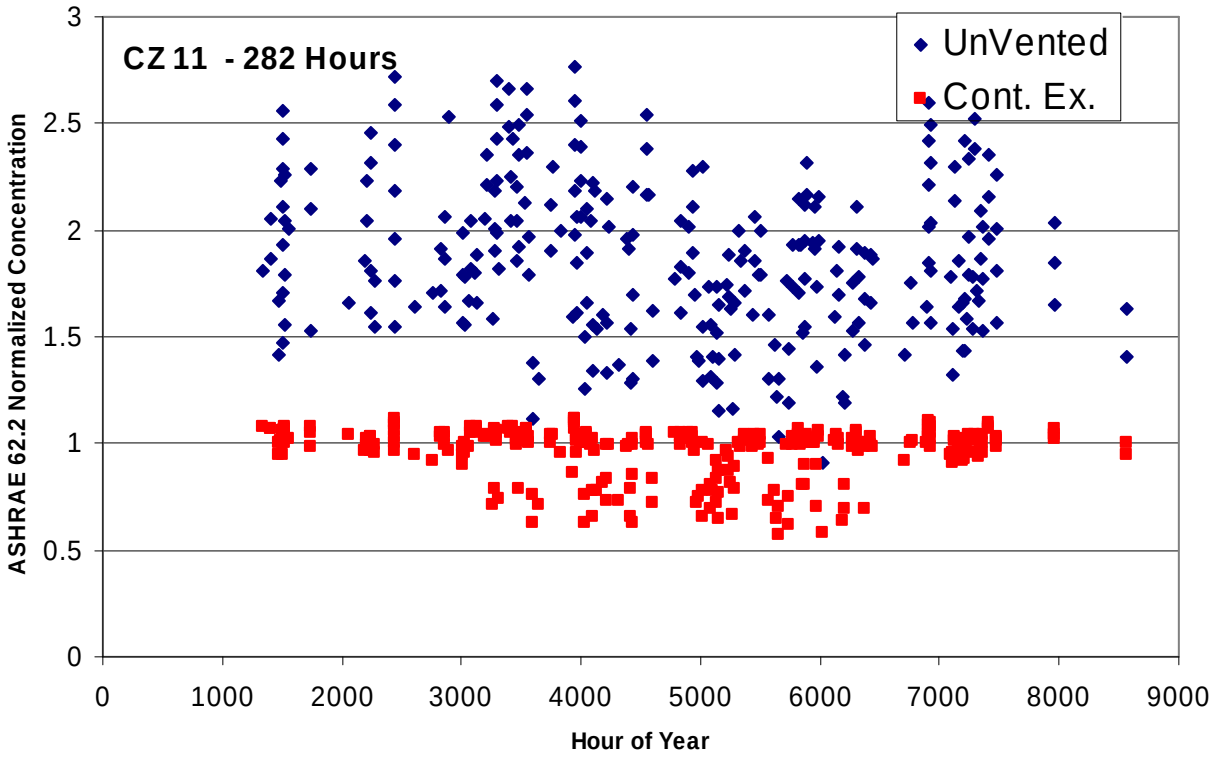

CZ 6 - 712 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

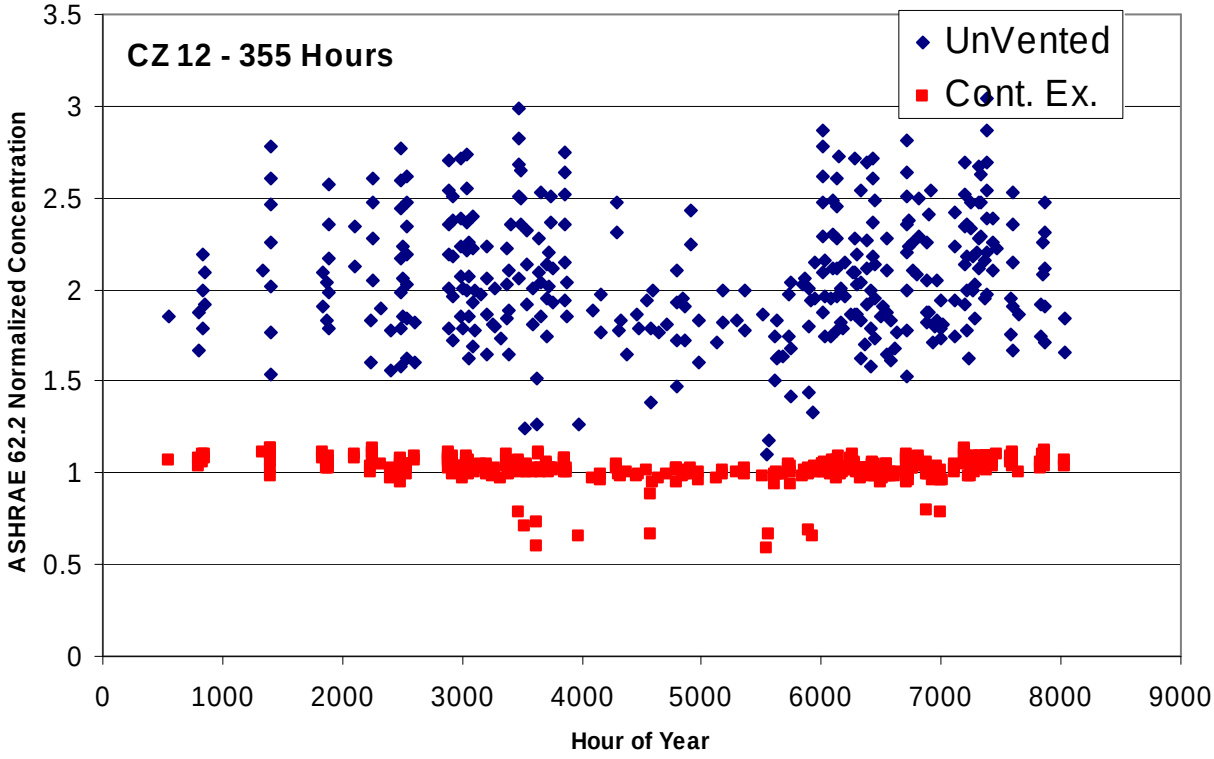

CZ 7 - 460 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
4
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

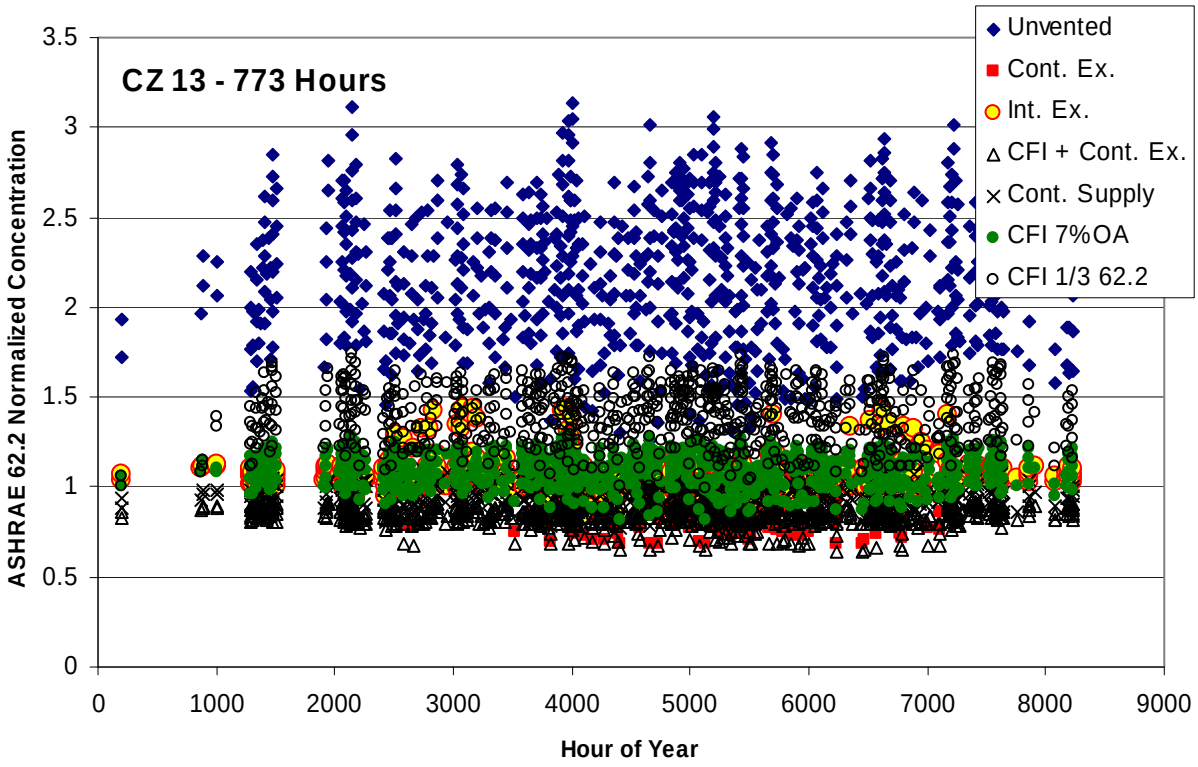

CZ 8 - 1191 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
4
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

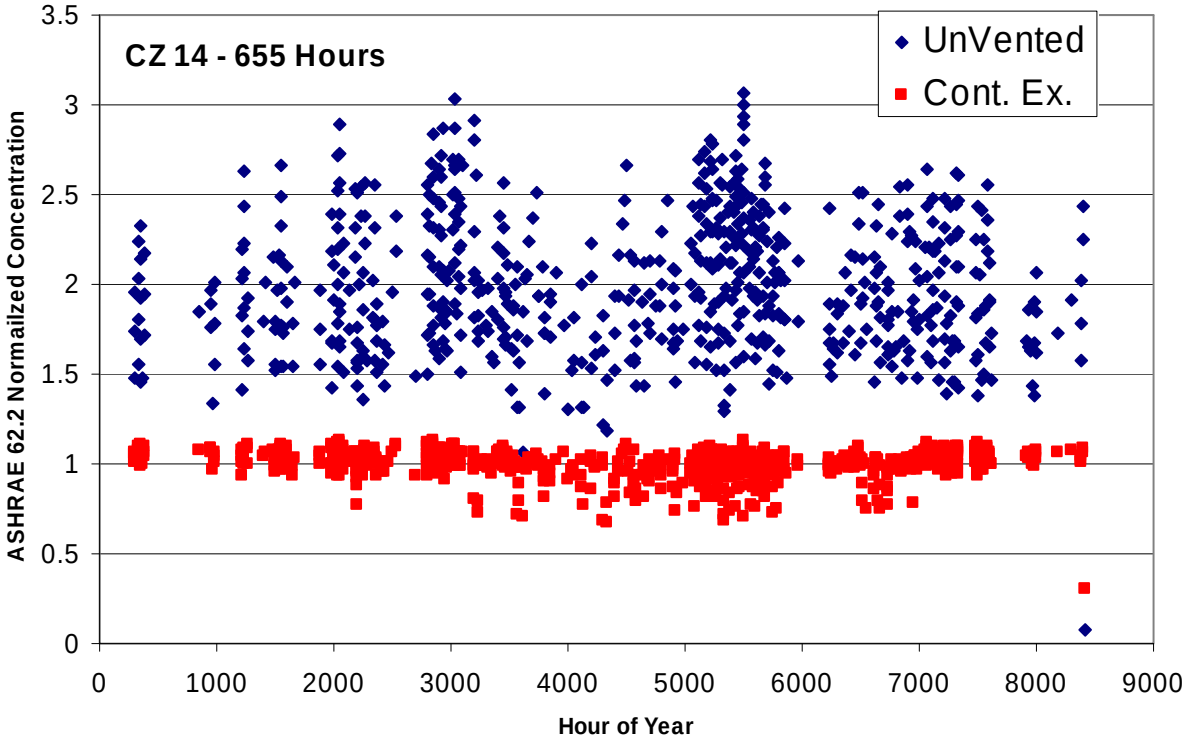

CZ 9 - 594 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
4
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

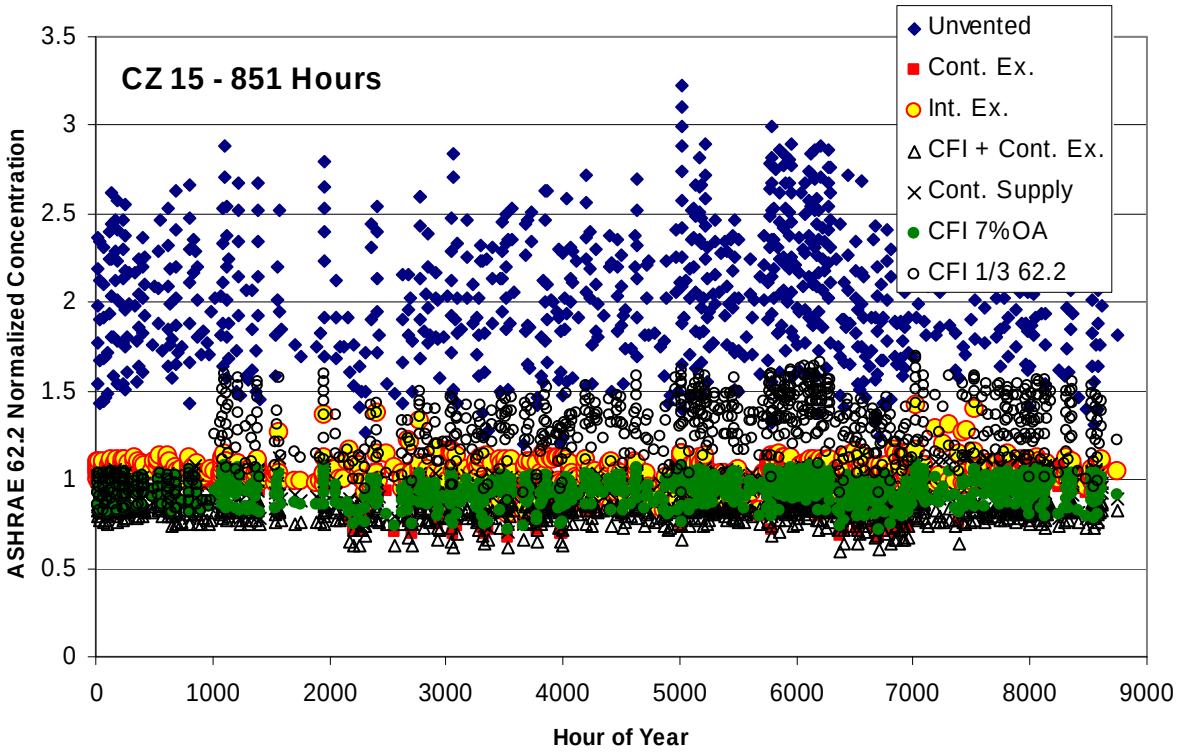

CZ 10 - 1127 Hours
UnVented
Cont. Ex.
Int. Ex.
CFI + Cont. Ex.
Cont. Supply
CFI 7%OA
CFI 1/3 62.2
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
4
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

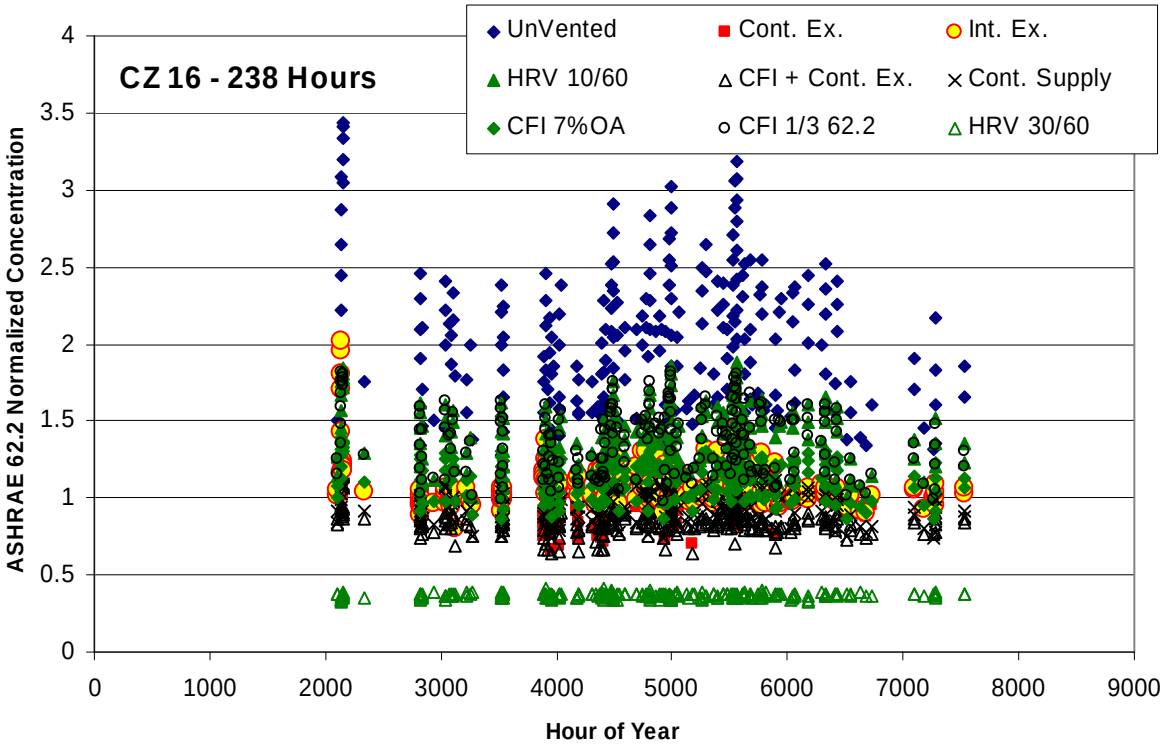

CZ 11 - 282 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
0
1000
2000
3000
4000
5000
6000
7000
8000
9000


CZ 12 - 355 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
0
1000
2000
3000
4000
5000
6000
7000
8000
9000


CZ 13 - 773 Hours
Unvented
Cont. Ex.
Int. Ex.
CFI + Cont. Ex.
Cont. Supply
CFI 7%OA
CFI 1/3 62.2
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
0
1000
2000
3000
4000
5000
6000
7000
8000
9000


CZ 14 - 655 Hours
UnVented
Cont. Ex.
ASHRAE 62.2 Normailzed Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
0
1000
2000
3000
4000
5000
6000
7000
8000
9000


CZ 15 - 851 Hours
Unvented
Cont. Ex.
Int. Ex.
CFI + Cont. Ex.
Cont. Supply
CFI 7%OA
CFI 1/3 62.2
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
0
1000
2000
3000
4000
5000
6000
7000
8000
9000


CZ 16 - 238 Hours
UnVented
Cont. Ex.
Int. Ex.
HRV 10/60
CFI + Cont. Ex.
Cont. Supply
CFI 7%OA
CFI 1/3 62.2
HRV 30/60
ASHRAE 62.2 Normalized Concentration
Hour of Year
0
0.5
1
1.5
2
2.5
3
3.5
4
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

Appendix E. Equipment Capacity and Blower Power Consumption

The equipment capacity was based on the results of a field survey of 60 new California houses performed as part of another PIER study (Rick Chitwood). The resulting heating and cooling capacities are generally greater than those estimated using sizing calculations such as ACCA Manual J/S procedures. In some cases the cooling capacity determines the heating capacity due to the limited packaging alternatives that are commercially available. Primarily this is an issue of furnace blower motor operating ranges that restrict the differences in heating and cooling capacities that can be serviced by an individual blower.

Climate Zone	Heating Capacity (KBtu/h)	Cooling Capacity (Tons)	Heating Blower Power (W)	Cooling (and Ventilating) Blower Power (W)
1	94	1.5	630	300
2	97	4	655	800
3	84	1.5	563	300
4	84	2	563	400
5	61	3.5	412	700
6	61	3.5	412	700
7	57	4	386	800
8	72	3.5	487	700
9	87	4	588	800
10	73	3.5	487	700
11	87	3	580	600
12	73	3	596	600
13	103	4	689	800
14	107	5	722	1000
15	136	5	916	1000
16	147	3.5	983	700