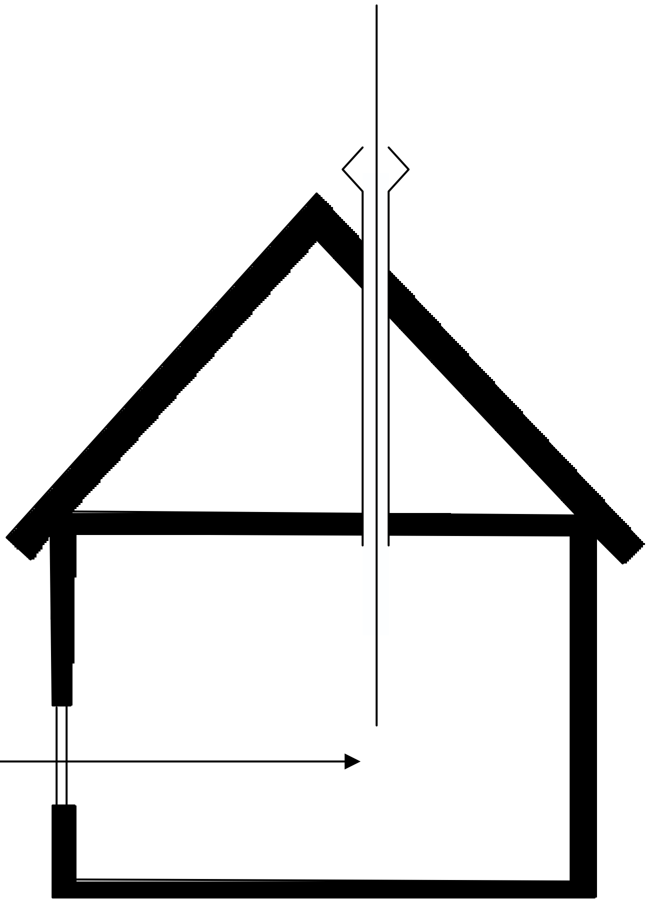
INTERNATIONAL ENERGY AGENCY
energy conservation in buildings and
community systems programme

Residential Passive Ventilation Systems: Evaluation and Design

James W. Axley



AIVC

Residential Passive Ventilation Systems: Evaluation and Design

A Critical Evaluation of the Potential for Adapting European Systems for use in North America and Development of a General Design Method

James W. Axley

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James Axley teaches structural and environmental technology courses and related seminars at the graduate level and an introductory structures course at the undergraduate level. Prior to joining the Yale faculty, he taught at U.C. Berkeley, Cornell University, and MIT over a 15 year period. In the professional realm, he has served as technical consultant to a number of influential architectural firms including those of Christopher Alexander, Fernau & Hartman, Lyndon & Buchanan, and Koetter & Kim and worked as a research engineer at the U.S. National Institute of Science and Technology. He has published and presented a series of papers relating to the development of computational and experimental techniques for building thermal, airflow, and air quality analysis; serves as consultant and adviser to the U.S. Environmental Protection Agency, U.S. Department of Energy, and National Institute of Science and Technology; and is an associate editor of the journal Indoor+Built Environment. He received his B.S. from the University of Wisconsin, Madison, and his M.Arch., M.S., and Ph.D. from the University of California at Berkeley.

As a visiting researcher at the AIVC in 1998, James Axley undertook the background research for this report.

This report is part of the work of the IEA Energy Conservation in Buildings and Community Systems Programme (ECBCS)

Publication prepared by
Annex V Air Infiltration and Ventilation Centre

Document AIC-54-2001

Participating countries of the AIVC:
Belgium, Denmark, Finland, France, Germany, Greece, the Netherlands, New Zealand, Norway, Sweden, United Kingdom and the United States of America.

Additional copies of this report may be obtained from:

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Preface

International Energy Agency

The International Energy Agency (IEA) was established in 1974 within the framework of the Organisation for Economic Co-operation and Development (OECD) to implement an International Energy Programme. A basic aim of the IEA is to foster co-operation among the twenty four IEA Participating Countries to increase energy security through energy conservation, development of alternative energy sources and energy research development and demonstration (RD&D).

Energy Conservation in Buildings and Community Systems (ECBCS)

The IEA sponsors research and development in a number of areas related to energy. In one of these areas, energy conservation in buildings, the IEA is sponsoring various exercises to predict more accurately the energy use of buildings, including comparison of existing computer programs, building monitoring, comparison of calculation methods, as well as air quality and studies of occupancy.

The Executive Committee

Overall control of the programme is maintained by an Executive Committee, which not only monitors existing projects but also identifies new areas where collaborative effort may be beneficial. To date, the following projects have been initiated by the Executive Committee (completed projects are identified by *):

1. Load Energy Determination of Buildings *
2. Ekistics and Advanced Community Energy Systems *
3. Energy Conservation in Residential Buildings *
4. Glasgow Commercial Building Monitoring *
5. Air Infiltration and Ventilation Centre
6. Energy Systems and Design of Communities *
7. Local Government Energy Planning *
8. Inhabitant Behaviour with Regard to Ventilation *
9. Minimum Ventilation Rates *
10. Building HVAC Systems Simulation *
11. Energy Auditing *
12. Windows and Fenestration *
13. Energy Management in Hospitals *
14. Condensation *
15. Energy Efficiency in Schools *
16. BEMS - 1: Energy Management Procedures
17. BEMS - 2: Evaluation and Emulation Techniques *
18. Demand Controlled Ventilating Systems *
19. Low Slope Roof Systems *
20. Air Flow Patterns within Buildings *
21. Calculation of Energy and Environmental Performance of Buildings*
22. Energy Efficient Communities *
23. Multizone Air Flow Modelling (COMIS) *
24. Heat Air and Moisture Transfer in Envelopes *
25. Real Time HEVAC Simulation *
26. Energy Efficient Ventilation of Large Enclosures *
27. Evaluation and Demonstration of Domestic Ventilation Systems
28. Low Energy Cooling Systems*

29. Daylight in Buildings
30. Bringing Simulation to Application
31. Energy Related Environmental Impact of Buildings
32. Integral Building Envelope Performance Assessment
33. Advanced Local Energy Planning
34. Computer-aided Evaluation of HVAC System Performance
35. Design of Energy Efficient Hybrid Ventilation (HYBVENT)
36. Retrofitting in Educational Buildings - Energy Concept Adviser for Technical Retrofit Measures
37. Low Exergy Systems for Heating and Cooling of Buildings
38. Solar Sustainable Housing.
39. High Performance Thermal Insulation (HiPTI)

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Synopsis

Infiltration has long served the residential ventilation needs in North America. In Northern Europe it has been augmented by purpose-provided natural ventilation systems – so-called *passive ventilation systems* – to better control moisture problems in dwellings smaller than their North American counterparts and in a generally wetter climate. The growing concern for energy use, and the environmental impacts associated with it, has however led to tighter residential construction standards on both continents and as a result problems associated with insufficient background ventilation have surfaced.

Recognizing the energy penalty of uncontrolled natural ventilation, building researchers and practitioners in North America are turning to mechanical systems to provide the necessary ventilation for air quality control. Northern Europeans are following suit but have not completely abandoned the passive ventilation methods that have served them for the past century. Research programs have been initiated in Britain, The Netherlands and France, in particular, to improve the understanding and performance of these traditional and largely empirically-based ventilation methods in the hope that they can more reliably provide basic background ventilation while avoiding the energy penalty associated with uncontrolled over-ventilation.

This state of affairs begs, then, a simple question:

Can European passive ventilation systems be adapted for use in North American dwellings to provide ventilation in an energy conservative manner?

This Technical Note attempts to answer this question. The configuration, specifications and performance of the preferred European passive ventilation system – the *passive stack ventilation (PSV)* system – will be reviewed; innovative components and system design strategies recently developed to improve the traditional *PSV* system performance will be outlined; and alternative system configurations will be presented that may better serve the climatic extremes and more urban contexts of North America. While these innovative and alternative passive ventilation systems hold great promise for the future, a rational method to size the components of these and other systems to achieve the control and precision needed to meet the conflicting demands of new ventilation and airtightness standards has not been forthcoming. Such a method will be introduced in this Technical Note, based on a review of existing simulation and design methods, and a series of applications of this method will be presented. Finally, provisions of the new *International One- and Two-Family Dwelling Code* that are likely to relate to the installation of passive ventilation systems will be reviewed and proposals for changes to this code will be put forward.

Acknowledgment

This work was supported by the Assistant Secretary for Energy Efficiency and Renewable Energy, Office of Building Technology of the U. S. Department of Energy under contract no. DE-AC03-76SF00098 through Lawrence Berkeley National Laboratory (LBNL) and the Air Infiltration Centre (AIVC) of the International Energy Agency.

The critical input and assistance of Mark Limb and Malcolm Orme of the Air Infiltration and Ventilation Centre during the summer of 1998 proved to be most important to the development of the ideas presented in this Technical Note. In addition, a number of able researchers in the field including David Etheridge of the University of Nottingham, Great Britain, Stina Homberg and Johnny Kronvall of AB Jacobson & Widmark, Sweden, Craig Wray of LBNL, and, again, Malcom Orme of the AIVC generously reviewed the first draft of this document and offered many suggestions for change that were all enthusiastically included.

Nomenclature

Recent developments in building systems simulation have integrated areas of analysis that were largely independent in the past. Building thermal analysis, airflow analysis and air quality analysis are now more commonly approached as coupled phenomena simply because the computational power is now available to do so. As a result the assignment of specific symbols to physical quantities has become problematic and, therefore, demands reconsideration. For example, in the past the symbol q was commonly used for heat transfer rates in thermal analysis, volumetric airflow rates in (isothermal) airflow analysis, and even contaminant mass transfer rates in air quality analysis. When formulating theory for coupled thermal, airflow and air quality analysis these distinct physical quantities must necessarily have different and unique symbols.

To the extent possible, the nomenclature and units used in this Technical Note follows the international conventions established by the American Society of Heating Refrigeration and Air Conditioning Engineers (ASHRAE), the American Society of Test Methods (ASTM), and the International Union of Pure and Applied Chemistry (IUPAC) (ASTM ; Mills, Cvitas et al. 1988; ASHRAE 1997).

Rate quantities are signified by the use of a dot over the variable name and time averaged values by the use of a bar over the variable name. Thus, for example, volumetric flow rate is represented by the symbol $\dot{V}$ and the time-averaged value of a given temperature time history is represented by the symbol $\overline{T}$.

Subscripts are used to associate a variable with a specific location within a building idealization (e.g., a *node* or a discrete flow path of the building idealization), or to more specifically identify the variable. Thus the temperature at a building node i is represented by the symbol T_i, the volumetric flow rate through flow path l as $\dot{V}_l$, and the effective area associated with this flow path opening as A_{eff-l} .

Variables

A	cross-sectional area (m^2)
A_{eff}	effective leakage area (m^2)
ACH	air exchange rate (1/h)
a	empirical exponent for ASHRAE terrain adjustment relation
a_0, a_1, a_2,…	empirical fan performance curve coefficients
C	constant coefficient (various dimensions)
C_d	discharge coefficient (dimensionless)
C_f	duct fitting loss coefficient (dimensionless)
C_i	indoor air pollutant concentration (kg/m^3)
C_P	wind pressure coefficient (dimensionless)
D	diameter; fan diameter (m)
D_h	hydraulic diameter (m)
D_o	base fan diameter (m)
f	duct friction factor (dimensionless)
$f_l(\Delta P_l)$	a mathematical model for the volumetric airflow rate (m^3/s) through a flow path l expressed in terms of the pressure difference ΔP_l across the flow path
G_i	air pollutant generation rate in zone i (kg/s)
g	the acceleration of gravity (9.8 m/s^2)

$g(\dot{V}_l, \phi_l)$	a mathematical model for pressure difference ΔP_l (Pa) across the flow path component *l* expressed in terms of the airflow rate $\dot{V}_l$ and the component design variable ϕ_l .
K	empirical coefficient for ASHRAE terrain adjustment relation
L	duct length (m)
$\dot{m}_l$	mass airflow rate through flow path *l* (kg/s)
N	fan rotational speed (rpm)
N_o	base fan rotational speed (rpm)
n	empirically determined exponent (dimensionless)
P	pressure (Pa)
Re	Reynolds number (dimensionless)
T	temperature (°C)
t	time (s)
S_U	wind shelter factor (dimensionless)
U	wind velocity corrected for sheltering effects (m/s)
U_H	estimated wind speed at eve height (m/s)
U_{ref}	reference wind velocity (m/s)
V	zone volume (m^3)
$\dot{V}_l$	volumetric airflow rate through flow path *l* (m^3/s)
$\dot{V}_o$	nominal controlled flow rate of a self-regulating vent (m^3/s)
z	elevation (m)
ΔP	pressure difference (Pa)
ΔP_o	control threshold pressure difference of a self-regulating vent (Pa)
Δz	change of elevation (m)
$\Delta z_{i,j}$	change of elevation from node *i* to node *j* (m)
ϕ	wind direction (rd); component design variable (various dimensions)
μ	viscosity of air (Pa·s)
ρ	air density (kg/m^3)

Subscripts

1,2,3,…	node identifiers
N, E, S, W	compass directions
a,b,c,…	flow element or component identifiers
c	ceiling
bldg	building
eff	effective
f	floor; forward flow
i	indoor
o	nominal; outdoor
r	reverse flow
s	stack
w	wall; wind

1 Introduction

Now, mild may be thy life!

For a more blustrous birth had never babe:

Pericles, on the birth of his daughter (Shakespeare 1608).

The first breath of a babe and the cooling sensation of air motion it simultaneously experiences gives birth to a lifelong obsession for fresh air and a preoccupation with air motion. The obsession, driven by survival instincts at one level and by simple pleasure at another, has shaped individuals' and thus civilizations' quest to replace stale air with fresh – i.e., to *ventilate for air quality control*. The preoccupation with air motion and the comfort or discomfort that it may create has, likewise, shaped the form and detail of human clothing, shelter and even settlement patterns. Given the primeval nature of this obsession and preoccupation, the development of ventilation systems for human habitation most certainly precedes any written history of it. Furthermore, the universality of the human need for fresh and sometimes cooling airflow suggests aspects of this development was most likely ubiquitous and not tied to any specific civilization, place or time. One can easily imagine the design of ventilation systems for air quality control began as individuals in all civilizations attempted to redirect the smoke from a cooking fire to avoid its personal impact and, for cooling, when others moved to shaded hill top locations to gain the benefits of cooling breezes.

From such modest beginnings we know, even from a casual reading of the history of building technologies, that the desire to control smoke from heating and cooking fires led to the development of *natural draft* methods. Thus the European open hearth fire of medieval halls and the simple roof opening that served it, was transformed into the *system* of the contained fireplace and later the fire box; the chimney, details of its configuration, and smoke damper devices placed within it; and the chimney terminal device or ''chimney pot'' fabricated for durability and formed to inhibit back-drafting and in some instances, to prevent rain water entry. While the search for comfort in hot climates lead to the development of *natural ventilation* methods – the site planning and dwelling configuration and detail strategies that fostered wind-driven and/or buoyancy-driven airflow through dwellings to cool the occupants in hotter climates.

It was inevitable that *natural draft* and *natural ventilation* technologies would be combined to advantage. Thus, notwithstanding other innumerable examples, by the beginning of the 14^{th} century in northern Wales chimney-like *stacks* were used in the construction of the Beaumaris Castle on some sixteen paired latrines to induce airflow to continuously displace the foul air of the latrines with fresh air without any need for intervention or control – that is to say to *passively ventilate* the latrines (Taylor 1988).

By the end of the 19^{th} century Europeans and North Americans, in particular, became particularly concerned about the quality of air within dwellings even, ironically, as outdoor air was increasingly being fouled by the progress of the industrial revolution. Epidemiological studies of the eighteen-sixties revealed health impacts of poor indoor air quality could be fatal and, importantly, proper ventilation could mitigate their risks (Banham 1969). Perhaps for this reason it was medical doctors who first took action rather than architects. Dr. Drysdale and Dr. Hayward of Liverpool each designed and built personal homes in the 1860's with innovative and complex passive ventilation systems and later in 1872 jointly published a text *Health and Comfort in House Building*. Reyner Banham writes of the design of Hayward's Octagon house (Banham 1969):

> *''…the whole plan, section, and construction of the house, has been affected by his determination to control the ventilation, and the matching manner in which practically everything within the house, including the gas-lighting, is consciously*

> *set to work to assist the structure in realizing that aim." (page 35 of (Banham 1969))*

Later in 1894, a British pathologist, Professor Jacobs of Yorkshire College, Leeds published *Notes on the Ventilation and Warming* that stressed the need to take a ''whole-building'' approach to the design of heating and ventilation systems and complained:

> *"Real ventilation is so uncommon that … the architect usually thinks this object has been attained if some of the windows can be opened. Some think that the presence of 'ventilators', especially if they have long names and are secured by 'Her Majesty's letters patent', ensures the required end. We may as well supply a house with water by a trap-door in the roof to admit rain."* (*from page 32 of (Banham 1969)*)

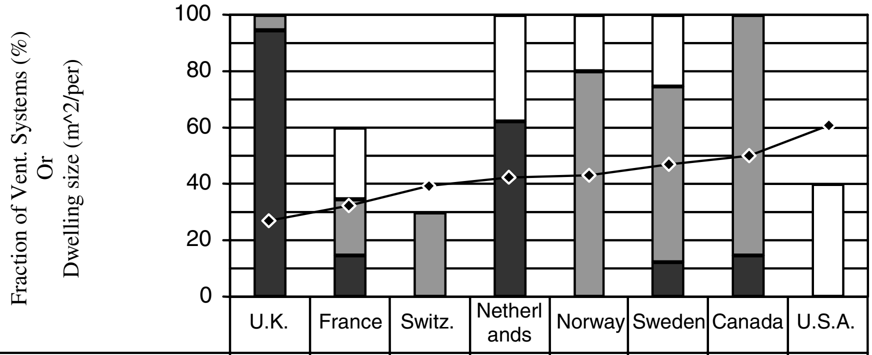
By 1899 construction details for the passive ventilation system that was to become the preferred European system – the *passive stack ventilation* (PSV) system – had been published in Sutcliffe's *Principles a & Practice of Modern Construction* (Sutcliffe 1899).

As interesting as these early passive ventilation systems may be, one feature of their operation has become unacceptable by today's standards. They relied heavily on the use of combustion heating devices directly and gas lighting devices indirectly to heat air to provide the buoyancy forces needed to drive the ventilation. Indeed, Professor Jacobs lamented the loss of gas lamps to electric lighting fixtures in concert halls as the former provided heat to drive ventilation (Banham 1969). Thus in these, perhaps exceptional, examples of the 19th century fossil fuels were not only consumed to heat ventilation air for thermal comfort reasons but also to increase buoyancy forces to drive ventilation airflows.

Concurrent with these innovative developments in passive ventilation, fan-forced ventilation began to have a significant impact on larger building design. The limited efficiency of these fans and the use of steam engines to drive them, however, allowed passive ventilation systems to remain competitive in larger buildings for some time (Banham 1969). Gradually, however, passive ventilation systems were left behind in larger buildings as mechanical systems took over and, as we now know, transformed, along with the elevator, electric lighting, and steel construction, the very nature of large building design. In Northern Europe, however, passive ventilation systems (of far greater simplicity than that of Drs. Drysdale and Hayward) continued to serve the residential sector. These systems do not now, however, nor did they in the past provide all the ventilation air needed to control indoor air quality.

Presently, ventilation for air quality control in residences is provided by a) unintended natural ventilation – *infiltration*, b) purpose-provided natural ventilation – *passive ventilation*, c) mechanically induced ventilation, or d) combinations of these three means. While detailed data of the distribution of residential ventilation systems is scarce, Table 1.1 assembled from data found in (Månsson 1995) provides some insight into this distribution in selected European countries, Canada and the U.S. as of the early 1990's. This table and the associated bar chart shows the percentage of single-family houses with a) passive ventilation stacks alone, b) both passive ventilation stacks and mechanical kitchen exhaust hoods, and c) other mechanical ventilation systems. The sum of the first two values – i.e., the height of the stacked gray bars in the accompanying chart – indicates the fraction of single-family houses with passive stacks and thus the fraction of homes relying in part on passive ventilation in these countries.

Table 1.1 Fraction of single-family ventilating systems with passive stacks or mechanical ventilation systems and average dwelling sizes of selected European and North American countries – early 1990's. Assembled from data reported in (Månsson 1995).

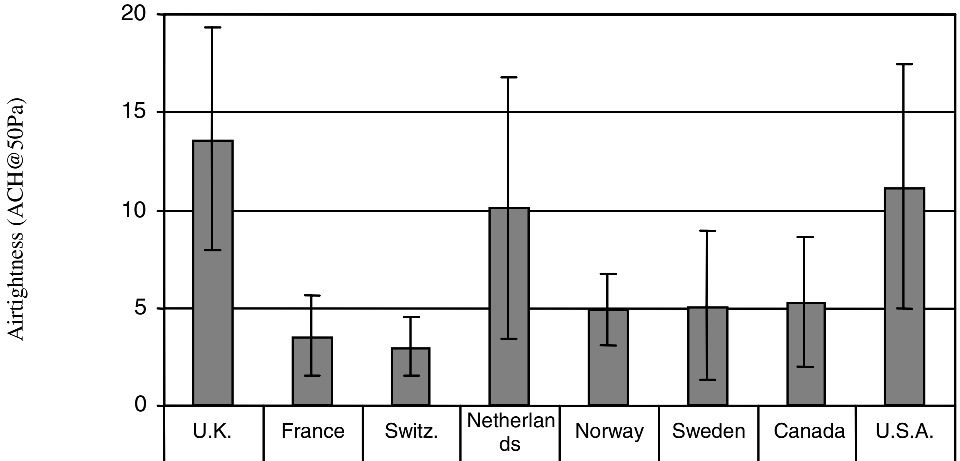


	U.K.	France	Switz.	Netherlands	Norway	Sweden	Canada	U.S.A.
Other Mechanical (%)	0	25	0	38	20	25	0	40
Stack + Kitch. Hood (%)	5	20	30	0	80	63	85	0
Stack (all N-systems: UK) (%)	95	15	0	62	0	12	15	0
Dwelling Size m^2/per)	27	32	39	42	43	47	50	61

* Data reported in row 3 for the U.K. includes all natural ventilation - stack, window airings, and infiltration. For the other countries listed, only stack ventilation is included.

This data clearly indicates that natural ventilation systems of one sort or another, including stack ventilation systems, are now practically ubiquitous in the United Kingdom and passive stack ventilation is quite common in other Northern European countries and Canada.

Table 1.2 Average airtightness levels, and their variation, of single-family homes in selected Euopean and North American countries. Assembled from data reported in (Orme, Liddament et al. 1998).



	U.K.	France	Switz.	Netherlands	Norway	Sweden	Canada	U.S.A.
ACH @ 50Pa	13.62	3.59	3	10.14	4.95	5.1	5.31	11.18

Yet passive stack ventilation is practically nonexistent in U.S. single-family homes. In contrast to Europe, U.S. dwellings have relied on infiltration to meet their ‘‘background’’ ventilation needs. Thus it should come as no surprise that U.S. homes are generally built to lower airtightness levels than many of their European counterparts. Table 1.2 presents average airtightness levels assembled from data reported in (Månsson 1995) – measured in terms of the number of air changes per hour (ACH) resulting from pressurizing homes to 50 Pa in standard building pressurization tests. The associated graph also indicates the variation of airtightness levels measured in terms of the standard deviation of the measured data about the mean value.

Somewhat surprisingly, U.S. homes while leakier than homes in the colder countries shown are just about as airtight as homes in the United Kingdom and The Netherlands. In fact, infiltration provides some part of the ventilation needed in homes in all countries as it is simply too difficult to make a house more airtight than the lowest values indicted in the chart associated with Table 1.2 (i.e., the lower extent of the bars showing the variation of airtightness levels in France, Switzerland, and Sweden).

What then does this data indicate about the selection of ventilation systems for dwellings? The specific mix of infiltration, passive ventilation and mechanical ventilation systems selected – as implicitly indicated by this data – has apparently served the ventilation needs of each individual country. These ventilation needs are related to a number of factors including dwelling size, climatic conditions, and cultural predisposition. Thus, in the relatively wet climate of the United Kingdom where dwelling sizes are, on average, relatively small (i.e., as indicated by the fourth row of data in Table 1.1) and cultural traditions favor the maintenance of lower temperatures in bedrooms, infiltration alone has proven to be insufficient and thus has been augmented by passive stack ventilation and window airings. Conversely, infiltration alone has sometimes provided sufficient background ventilation in the U.S. where homes are relatively larger, the climate is relatively dryer and cultural traditions favor higher and more uniform temperature conditions within dwellings.

To confound matters even more, ventilation systems are invariably selected to provide either continuous general *background* ventilation – i.e., to replace air contaminated by relatively continuously emitting sources – or to provide intermittent ventilation for intermittent but typically stronger sources – e.g., due to cooking or bathing. In the U.S. infiltration has traditionally been relied upon to provide background ventilation, while mechanical exhaust fans for kitchens and bathrooms in newer construction and operable openings in older construction have been installed to provide the needed intermittent ventilation. In Northern Europe, on the other hand, passive ventilation systems have traditionally assumed both roles – indirectly inducing low level background ventilation in living and bedrooms while directly venting moisture and odors generated intermittently in bathrooms and kitchens.

The salient point to be made here is that the actual combination of ventilation systems used is in itself relatively unimportant when one is concerned with air quality control. A recurrent theme of this Technical Note is that ventilation system design is not well determined – many different ventilation systems can serve the ventilation needs of any given dwelling and thus the ventilation needs of a country. Consequently, we should not try to read too much into the actual distribution of ventilation systems used in any given country.

On the other hand, changes to existing ventilation systems have the potential to upset the balance that has developed empirically over the years. Thus, for example, if we seek to increase building airtightness then the resulting reduction in infiltration must be offset by an increase in passive or mechanical ventilation if health or moisture-related problems of under-ventilation are to be avoided. That is to say, when considering air quality control we must not be preoccupied with finding the ‘‘best’’ ventilation system but should seek instead to devise methods that will enable us to design ventilation systems that achieve the air quality desired.

1.1 Energy Conservation

Ventilation system choice can have a significant impact on energy use however (e.g., see (Sherman and Matson 1993; Orme 1998)). Building energy uses account for approximately 40% of total primary energy use in developed countries. Of this, the residential sector consumes some 60% – roughly 70% of which is used for space conditioning. As building insulation levels have increased in recent years so too has the fraction of the energy consumed for heating or cooling ventilation air – now between 30 to 50% of the energy used for space conditioning. Thus, there is the potential to conserve as much as 5% to 9% of total primary energy use via reduced or more efficient ventilation strategies.

Infiltration, by its nature, and traditional passive ventilation systems, by their design, provide inadequate control of ventilation rates and air distribution. They tend to suffer over-ventilation yet poor air distribution during windy or cold climatic conditions and under-ventilation during milder conditions or when sheltered by adjacent buildings (Wilson and Walker 1991; Wilson and Walker 1991; Wilson 1992). The energy consequences of the former and air quality impact of the latter have fostered the promotion of airtight construction and the use of mechanical ventilation systems in residences in North America and Japan (Lubliner, Stevens et al. 1997; Reardon and Shaw 1997; Roberson, Matson et al. 1997; Murakami, Kobayashi et al. 1998). Indeed, the limiting constraints of new residential air-tight building regulations (e.g., (ASHRAE 1988; ASHRAE 1994)) and more demanding ventilation standards (e.g., (ASHRAE 1989; Murakami, Kobayashi et al. 1998)) seem to some to leave no other alternative.

In spite of these shortcomings there has been a renewed interest in passive ventilation systems in Northern Europe that has lead to new standards for their installation in dwellings (Department_of_the_Environment 1995; Bossaer, Demeester et al. 1998). Recent developments in system components and design strategies, which have emerged from national research programs in, especially, Britain, The Netherlands and France, may now provide the control needed to maintain acceptable ventilation rates while still avoiding the tendency of these systems to over-ventilate during more extreme climatic conditions. On the basis of these developments alone, passive ventilation systems should be considered for the North American context as well.

Clearly, providing residential ventilation needs *passively* – that is, using natural forces provided by wind and buoyancy – has the potential to avoid the energy that would be consumed using mechanical means. With clever design and some luck, it also has the potential to save first and operating costs, to provide ventilation without occupant intervention, and to provide truly fresh, odor-free outdoor air. All this while avoiding, or at least minimizing, the significant energy penalty, and consequent environmental impact associated with over-ventilation (Sherman and Matson 1997; Orme 1998) and, of course, the comfort, moisture-related, and health problems that may be caused by under-ventilation. It must be emphasized, however, that the level of control needed to compete with the mechanical alternatives is great and a variety of other pitfalls relating to such matters as rain and dust entrainment, acoustical isolation, improper air distribution, disruptive air velocities and cold drafts, fire and smoke control, occupant sabotage, and, most important of all, heat recovery must be addressed to fully realize the potential passive ventilation offers.

It is natural to ask:

> *Can European passive ventilation systems be adapted for use in North American dwellings to provide ventilation in an energy conservative manner?*

This Technical Note attempts to answer this question.

1.2 Air Quality Control

That ventilation is needed for air quality control is not controversial. The meaning and means of *air quality control*, however, are. ASHRAE's position here is clear – air quality control is to provide *"air in which there are no known contaminants at harmful concentrations … and with which a substantial majority … of the people exposed do not express dissatisfaction"* (ASHRAE 1989). To this end ASHRAE Standard 62-89 offers both performance (i.e., the Indoor Air Quality Procedure) and prescriptive (i.e., the Ventilation Rate Procedure) standards for ventilation.

Importantly, ASHRAE Standard 62-89 acknowledges that the prescriptive standard *"provides only an indirect solution to the control of contaminants"* while the performance standard *"provides a direct solution by restricting the concentration of all known contaminants of concern to some specified acceptable level."* Nevertheless, ASHRAE's prescriptive standards stipulating outdoor air requirements for ventilation (e.g., *"0.35 air changes per hour but not less than 15 cfm (7.5 l/s) per person"* for living areas) often shape the discourse of air quality control.

The most relevant elements of the ASHRAE Standard 62-89 performance standard for residential buildings are that a) *"humidity in habitable spaces preferably should be maintained between 30% and 60% relative humidity to minimize growth of allergenic or pathogenic organisms"* and b) *"comfort (odor) criteria are likely to be satisfied if the ventilation rate is set so that 1000 ppm CO_2 is not exceeded."* Satisfaction of these often conflicting criteria – the control of air quality fundamentally, especially odors and humidity in the residential context, or the control of ventilation rates practically – will define the meaning of *air quality control* in this report and be the basis of evaluation of ventilation system performance.

1.3 Organization

Chapter 2 of this Technical Note will review the design and performance of the preferred European passive ventilation system – the passive stack ventilation (PSV) system – giving special consideration to those components and system design strategies that appear at this time central to improved airflow control. Emerging new developments to complement PSV systems with energy conserving heat recovery devices will also be considered. Finally, complicating related design issues will be outlined.

As the PSV system may not be well suited to all North American climates, alternative passive ventilation options will next be discussed in Chapter 3.

Simulation and design methods that may be applied to the analysis of passive ventilation systems will be reviewed in Chapter 4 in an effort to set the stage for the development of a new design method.

Chapter 5 will present this new method for the sizing of components of passive ventilation systems in general and consider the application of this method to eight design problems of increasing complexity.

Chapter 6 will review the provisions of the new North American *International One- and Two-Family Dwelling Code* that are likely to relate to the installation of passive ventilation systems. Proposals for changes to these provisions will then be put forward that would be required to support the use of passive ventilation systems in North America.

Finally, Chapter 7 will summarize and draw conclusions from the preceding chapters.

2 Review of European Passive Ventilation Methods

The number and variety of devices installed in British, Scandinavian, and Northern European dwellings intended to provide passive ventilation is bewildering. Terracotta ‘‘air-bricks’’, cast iron, sheet metal, wood, and plastic inlet vents of an endless variety of forms placed within wall constructions, fabricated as part of window or door assemblies, or placed directly into the glazing of windows and door lights are the most obvious. Operable outlets placed above windows, within skylight and clerestory assemblies, and in other higher locations are less obvious but nevertheless evident to the careful observer. Ventilation stacks are the most difficult to detect, of course, but they too can be found in many houses if one is careful to look for them.

It is clear, however, that many of these devices are not serving their intended purpose. All too often inlet devices are not installed as part of a total passive ventilation system – that is to say they are not associated with outlet devices nor devices to provide air flow from inlet to outlet. Operable inlet and outlet devices are often not maintained or are left in their closed position throughout the year. Inlet and outlet vents are often blocked by debris or new construction, plastered or wallpapered over, or intentionally filled-in during renovations. Finally, some popular passive ventilation devices have wind-driven moving components intended to reveal their operation that may actually introduce unnecessary resistance that reduces air flow and thus the efficacy of the devices.

2.1 Passive Stack Ventilation (PSV) Systems

Given this state of affairs, little may be gained from a comprehensive review of existing passive ventilation systems in Northern Europe. Instead it is most useful to consider *best-practice* passive ventilation system in Britain – the *Passive Stack Ventilation* (PSV) system. The Building Research Establishment has assumed a lead role in developing and promoting PSV systems (Stephen and Uglow 1989; Uglow and Stephen 1989; Parkins 1991; Palmer, Parkins et al. 1994; Parkins 1994; Parkins 1994; Shepherd, Parkins et al. 1994; Shepherd, Parkins et al. 1994; Stephen, Parkins et al. 1994; Welsh 1994; Welsh 1995; Welsh 1995; Welsh 1995). Consider, then, the representative PSV system based on their recommendations illustrated in Figure 2.1.

Given the relatively wet climatic conditions and typical North European dwelling sizes – small by North American standards – passive ventilation systems in the North European context are often directed primarily toward moisture and odor control. Hence, the BRE *best-practice* systems are configured to directly ventilate bathrooms and kitchens (i.e., to supply outdoor and exhaust stale air directly to and from these rooms) and only indirectly ventilate other occupied rooms in a dwelling (i.e., to supply outdoor air and/or exhaust stale air via adjacent rooms). Thus in the *best-practice* example shown the bathroom and kitchen are served by independent ventilation subsystems consisting of fresh air inlets and an exhaust stack while stale air in other habitable rooms is exhausted indirectly via openings in doorways or interior walls to the exhaust stacks in the bathroom or kitchen. By directly exhausting the bath and kitchen and directing other ventilation airflows to these service rooms moisture and odors generated in the bathroom and kitchen will tend not to migrate to other rooms of the house.

It is interesting to note that the new Belgium ventilation standard NBN D50-001 takes a different approach – basic ventilation is to be supplied through *dry* rooms, transferred through corridors, and exhausted through *wet* rooms (Bossaer, Demeester et al. 1998). The BRE approach offers local and independent control of moisture and odors sources in kitchens and bathrooms while providing whole house general ventilation – the Belgium approach offers an integrated approach to these two distinct ventilation objectives.

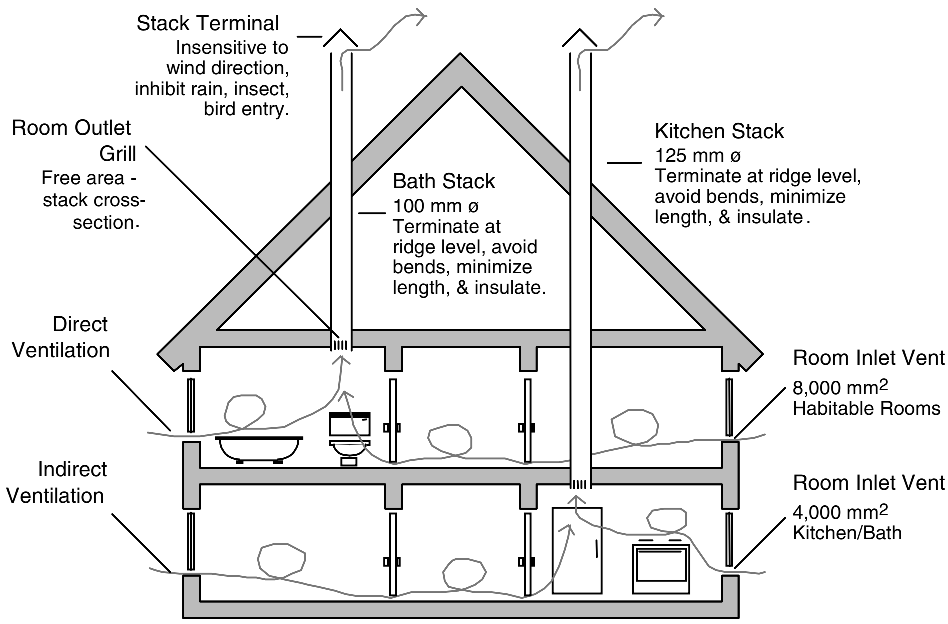


Figure 2.1 A representative Passive Stack Ventilation (PSV) system for a typical two-storey Northern European dwelling – British specifications (Stephen, Parkins et al. 1994; Department of the Environment 1995).

The direct ventilation components of the BRE *best-practice* PSV systems include (Stephen, Parkins et al. 1994):

- a service room *inlet vent,* with an open free area of at least 4,000 mm^2 (6.2 in^2), intended to maintain a continuous background and relatively small ventilation rate and, as such, often called a ‘‘trickle vent’’,
- a room *outlet grill,* having a free area equal to or greater than the stack cross-sectional area, ideally placed in the ceiling plane or high on a wall above the primary sources of moisture and odors that provides a suitably finished transition from the room to the ventilation stack,
- a ventilation *stack duct*, 125 mm (5 in) in diameter for kitchens and 100 mm (4 in) for bathrooms, that directs ventilation airflows through the dwelling to a high point ideally above the roof ridge, and
- a *stack terminal* device through which the ventilation air flows exhaust outdoors that is intended to maintain a positive (suction) pressure due to wind flow and, possibly, inhibit rain water, insect, and animal entry into the stack.

The indirect ventilation of the habitable rooms of the dwelling is to be provided with:

- habitable room *inlet vents* with an open free area of at least 8,000 mm^2 (12.4 in^2) , and
- *transfer grills* and/or undercut doors that provide air flow paths from habitable rooms to the service rooms to complete the circuit from inlet to exhaust.

Fine tuning and occupant control of both the direct and the indirect PSV systems have traditionally been provided by manually adjustable inlet vents and/or outlet grills. The manual control of passive ventilation systems is, however, a subtle and time-demanding process. Consequently, much of recent research has been directed toward the development of automatic control devices – most commonly for inlet vent components – that respond automatically but ‘‘passively’’ (i.e., without the need for additional power or servo-mechanisms) to changes in temperature, relative humidity, air contaminant levels, and/or pressure differences (Knoll and Kornaat 1991; Knoll 1992; Knoll and Phaff 1998). We shall consider these *self-regulating* components in greater detail below.

2.1.1 Operating Principles

Air flows through PSV systems due to pressure differences derived from both wind-driven airflow passing over the dwelling and buoyancy differences between indoor and outdoor air.

For the typical Northern European dwelling without air conditioning, indoor air temperatures are consistently higher than outdoor temperatures throughout the year, thus the warmer and less dense indoor air tends naturally to rise up through the ventilating stack and as it does it draws in cooler outdoor air through the inlet vents provided. From another but equivalent point of view, these air flows are driven by the combination of positive (inward) pressures acting on the PSV inlets and negative (outward or suction) pressures acting on the stack terminals – the *buoyancy* or *stack* pressure differences that result from the density differences between indoor and outdoor air, Figure 2.2.

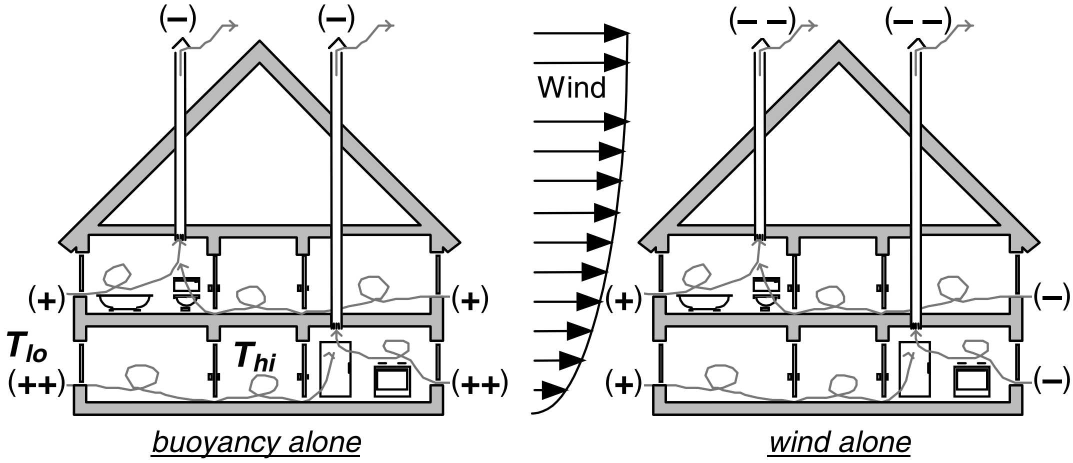


Figure 2.2 Positive (+) and negative (–) envelope pressures resulting from buoyancy alone and wind alone. Relative magnitude indicated by number of symbols.

Wind flows, on the other hand, tend to create positive pressures on the windward side and negative pressures on the leeward side of dwellings that, alone, tend to cause cross-ventilating airflows. In PSV systems, however, the position of the stack terminal and its detailed configuration assure that the greatest negative pressure occurs at the stack terminal itself. Thus air will flow in through all inlets and out through the ventilation stack as desired, Figure 2.2.

Under less desirable and, often, extreme wind conditions this objective may not actually be achieved. However, if the pressure differences due to buoyancy effects are large enough they will overcome the reverse-flow pressure difference induced by the wind and the ventilation air will flow as intended. Due however to differences in the magnitudes of these pressures, as indicated in Figure 2.3, ventilation flow rates will tend to vary significantly from room-to-room unless measures are taken to control the airflow rates in relation to the pressure differences experienced. Upper, leeward rooms, in particular, may be expected to be under-ventilated relative to other rooms in these cases and may be subjected to unintended reverse flows at times. Consequently, traditional PSV systems tend to suffer from poor air distribution in these circumstances.

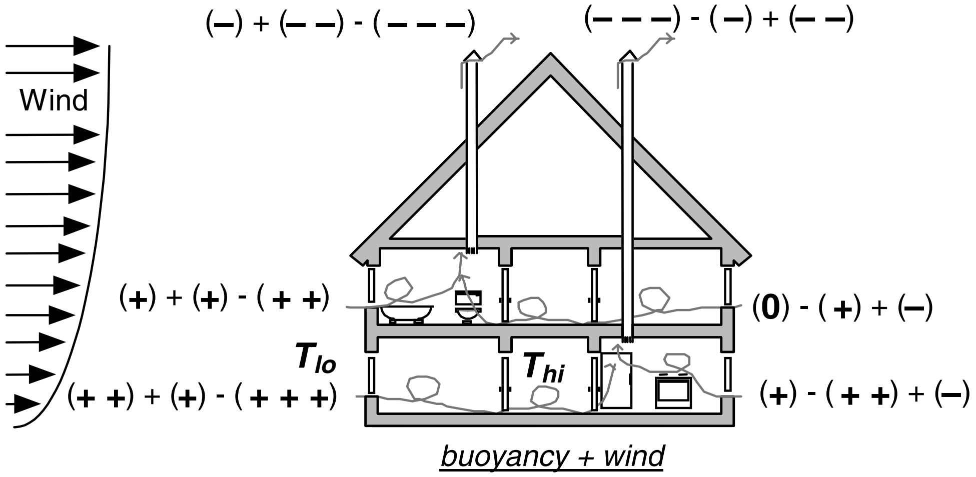


Figure 2.3 Positive (+) and negative (–) envelope pressures resulting from the combined effects of buoyancy and wind. Relative magnitude indicated by number of symbols.

For dwellings with indoor temperatures consistently greater than outdoor temperatures, the ventilation stack serves the important role to control the direction of airflow under the variety of climatic conditions that may prevail. Furthermore, for these dwellings a taller stack acts to create greater buoyancy pressure differences thereby providing greater control over the flow direction. Terminating a stack at or above the ridge level has the added advantage of rendering the stack less sensitive to variations of wind direction gaining additional control over ventilation flow magnitudes in time. The ventilation stack also acts to limit momentary reverse flow events driven by wind turbulence through the inertia and viscous damping offered by the air contained within the stack. While this phenomena has yet to be completely elucidated (Etheridge 1998), it is clear that a taller stack can be expected to more effectively inhibit dynamic flow reversals than a shorter one.

The rate at which air flows through the PSV system is dependent on the actual magnitude of the wind and buoyancy-driven pressures acting at any moment in time and the resistances to air flow encountered along each of the direct and indirect ventilation air flow paths. A common strategy used to gain some control over these airflow rates involves adjusting flow resistances along each path so that a single component's resistance limits flow (i.e., offers the primary resistance to airflow along the given path). Typically, the inlet vent components are designed to be the flow-limiting component and are fitted with operable shutters so that their resistance may be manually adjusted to achieve the desired flow rate. To complete the system design using this strategy, one then need only size and configure all other components to have negligible (non-limiting) resistances. While the principle here is clear, analytical methods to achieve this objective have been wanting and thus reliance has been placed on more empirical approaches to design based on experience, laboratory and field investigations. Such an analytical method will be discussed subsequently.

Unsteady meteorological conditions lead inevitably to unsteadiness in the buoyancy and wind-induced pressures that drive airflow in PSV systems. Thus to truly control the magnitude of air flow rates it is not enough to govern flows via limiting inlet vent resistances – these resistances must also be adjustable and, to avoid the complications of occupant intervention, self-regulating. Furthermore, the need for ventilation will, in general, vary depending on occupancy, moisture levels, odor, or other air contaminant levels once again demanding regulation of airflow.

Consequently, much of recent research has been directed toward the development of automatic, self-regulating, flow components that respond to changes in temperature, pressure, relative humidity, or air contaminant concentrations (Eriksson, Masimov et al. 1986; Wouters and Vandaele 1990; Knoll and Kornaat 1991; Knoll 1992; Heikkinen and Pallari 1994; Palmer, Parkins et al. 1994; Martin 1995; Palin, McIntyre et al. 1996; de Gids 1997; de Gids 1998).

2.1.2 Performance

The performance of complete, well-designed PSV systems has been investigated by both field studies of actual installations and computational simulation – most commonly annual simulations using multi-zone models. Overall, these systems have been relatively successful. The desired background ventilation has been achieved and moisture and humidity control have been provided for modest initial investments (projected or actual) without added operational expense or the need for occupant control or intervention. Nevertheless, PSV systems have not always lived up to expectations and in one key respect – that of heat recovery – have fallen well short of the advantage balanced mechanical system alternatives offer.

It is instructive to examine the successes and failures of the PSV systems investigated in detail. Here we will make a temporal distinction between long-term, seasonal, and short-term response and consider spatial differences between global or whole-building and local ventilation characteristics. The results of some thirteen different research groups were reviewed. Of these, eleven groups considered *traditional* PSV systems utilizing constant area inlets (Johnson and Pitts 1982; Shaw and Kim 1984; BRECSU 1985; Johnson, Gaze et al. 1985; Edwards and Irwin 1986; Parkins 1991; Wilson and Walker 1991; Wilson 1992; Bassett 1994; Bassett 1994; Heikkinen and Pallari 1994; Parkins 1994; Parkins 1994; Shepherd, Parkins et al. 1994; Shepherd, Parkins et al. 1994; Woolliscroft 1994; Enai, Aratani et al. 1996; Hayashi and Yamada 1996; Palin, McIntyre et al. 1996; Månsson 1998) while one group, at TNO in the Netherlands, developed and investigated an *innovative* PSV system using automatic, self-regulating inlet vents (Knoll and Kornaat 1991; Knoll 1992; de Gids 1997; de Gids 1998; de Gids 1998; Knoll and Phaff 1998).

In considering these and other studies a simple complication invariably arises – some investigators set nominal ventilation rates as their criteria of success while others focus more directly on measures of air quality including, typically, moisture or CO_2 levels. Regrettably, in some instances the criteria used is not entirely clear. Central to these evaluations are questions of ‘‘under’’ or ‘‘over-ventilation’’. A given system may provide ventilation rates that fall below a nominal design ventilation rate and still provide acceptable control of air quality over the long term depending, of course, on the nature of moisture or contaminant sources within the dwelling. Conversely, during short term episodes (e.g., showering or certain kinds of cooking activities) no reasonable amount of ventilation can limit peak contaminant levels to desirable levels. Consequently, the terms ‘‘under’’ or ‘‘over-ventilation’’ will have to remain largely in the qualitative domain below.

2.1.2.1 Long-Term Performance – Traditional PSV Systems

Over the long term, properly designed traditional PSV systems effectively controlled average indoor humidity and odor (Johnson, Gaze et al. 1985; Parkins 1994; Palin, McIntyre et al. 1996) comparing well with mechanical extract ventilation systems (Shepherd, Parkins et al. 1994). Conversely, improperly designed PSV systems (i.e., with smaller inlet vents than currently recommended in the U.K.) did not (Palin, McIntyre et al. 1996).

PSV systems can not mitigate peak excursions of odors and moisture levels in kitchens and baths as effectively as simple extract fans placed in these rooms (Palin, McIntyre et al. 1996). The absolute energy involved in limiting these peaks with fans is small, however, in comparison to that needed for ventilation in general (Woolliscroft 1994) suggesting the integrated use of mechanical extract fans with passive ventilation systems should be considered. However, containment of local sources may be as important as choice of the ventilation method. Shepherd notes: *"Closing the door while cooking is as important as providing extract ventilation because it prevents a large proportion of the moisture from migrating upstairs"* (Shepherd, Parkins et al. 1994).

Field tested PSV systems provided either somewhat less (Johnson and Pitts 1982; Palin, McIntyre et al. 1996) or somewhat more (Wilson 1992) than the desired ventilation rate in the long-term. Tightly constructed houses and/or row houses with relatively small leakage areas due to common side walls are, however, more likely to be under-ventilated (Wilson 1992).

> *Passive Ventilation Strategy*
>
> *These results clearly indicate that sizing of the components of PSV systems is critical and, importantly, this sizing must account for the impact of infiltration.*

2.1.2.2 Seasonal Performance– Traditional PSV Systems

During the winter season, temperature differences between indoors and outdoors and, hence, buoyancy forces become relatively large. Winter storms with increased wind velocities act to exacerbate these conditions. Consequently, ventilation airflow rates in traditionally designed PSV systems may be expected to reach annual highs during the winter season. PSV systems designed to provide a reasonable ventilation flow rate on average will then tend to over-ventilate during the winter season if the PSV system is not regulated accordingly (Wilson 1992; Heikkinen and Pallari 1994; Månsson 1998). Conversely, for relatively calm and mild conditions during the shoulder seasons traditional PSV systems will tend to under-ventilate (Johnson, Gaze et al. 1985; Wilson 1992; Heikkinen and Pallari 1994; Månsson 1998).

If truly dead calm conditions prevail and indoor and outdoor temperatures are nearly equal, then no amount of regulation can significantly increase ventilation rates in PSV systems. This problem is often cited as an important fundamental shortcoming of PSV systems. Yet, dead calm conditions may be expected to occur very infrequently in most locations (Alexander, Jenkins et al. 1996) and during these conditions occupants are likely to open windows and doors thus significantly reducing the resistance of their residences to wind or buoyancy driven airflow. Furthermore, given typical interior volumes of residences and the often short duration of calm conditions, i.e., when they occur, indoor contaminant level increases will be attenuated in actuality (Yuill 1991). Consequently, under-ventilation during mild seasons may not prove to be a significant problem in PSV or other passive ventilation systems although additional study is certainly warranted to better evaluate this problem.

Annual simulations indicate that self-regulating temperature-controlled inlet vents and outlet grills can mitigate wintertime over-ventilation. Results also indicate that these devices save some of the energy (5%) needed to condition ventilation air when compared to a system providing a constant ventilation rate throughout the year. This saving proved to be marginally significant, however, when compared to the ventilation energy savings offered by a balanced mechanical system which saved 50% (Heikkinen and Pallari 1994).

2.1.2.3 Short-Term Performance – Traditional PSV Systems

Over short periods of time, PSV systems can tend to over- and under-ventilate due to variations in wind speed that induce variations in the pressures acting on both the PSV system and the building envelope itself. As might be expected, sensitivity to wind speed variations is greatest with systems not employing a stack at all (Bassett 1994; Bassett 1994). The stack of the PSV system acts to reduce sensitivity to wind speed variations (Wilson 1992; Bassett 1994; Bassett 1994) as does tighter dwelling construction (Bassett 1994; Bassett 1994; Enai, Aratani et al. 1996).

Similarly, PSV systems tend to be sensitive to variations of wind direction, which again induce variations in wind pressures acting on the building and PSV system. Row houses are especially sensitive to these wind direction due to the sheltering offered by adjacent units in the row (Wilson 1992). Sensitivity to wind direction may be mitigated using properly designed and terminated stacks and tight construction (Wilson 1992; Parkins 1994).

A fundamental strategy emerges from these observations:

> *Passive Ventilation Strategy*
>
> *Successful PSV systems must be able to provide consistent ventilation airflow rates under conditions of constantly changing wind speeds and wind directions. The use of stacks alone provides control authority over ventilation flow direction for variations in wind speed with taller stacks providing greater authority over shorter ones. The proper termination of stacks – i.e., termination above the highest point on the roof and using a wind direction insensitive terminal device – provides control authority over ventilation flow magnitudes. Finally, as infiltration can not be controlled, building tightness improves PSV system control over ventilation flow magnitudes (Johnson and Pitts 1982; Bassett 1994; Bassett 1994; Heikkinen and Pallari 1994; Enai, Aratani et al. 1996; Hayashi and Yamada 1996).*

Conventional PSV systems solve the problem of air distribution by providing airflow inlets (e.g., trickle vents) for each habitable room of a dwelling that directly admit outdoor air to the rooms. In colder climates, this direct ventilation of rooms may result in unpleasant cold drafts – a problem that is exacerbated by the tendency of PSV systems to over-ventilate during colder weather.

> *Passive Ventilation Strategy*
>
> *To avoid cold drafts use inlet vents that inject outdoor air high in the space at low velocities (e.g., using trickle vents placed above windows rather than below them in wall construction). Alternatively, outdoor air may be introduced into a dwelling through floor ("beamspace") air distribution ducts or plenums that act to preheat the incoming air and injected into rooms through floor registers (Enai, Aratani et al. 1996; Hayashi and Yamada 1996). The success of such floor distribution systems depends, again, on tight construction.*

Subfloor distribution systems offer another potential advantage. If they are configured with supply inlets placed around the perimeter of a dwelling, they may be expected to improve the wind direction insensitivity of passive ventilation systems (Etheridge 1998). The subfloor air distribution ducts or plenum serve to average wind pressures that act over the perimeter of the building.

> *Passive Ventilation Strategy*
>
> *Subfloor distribution systems can, in principle, improve wind direction insensitivity, albeit at the expense of reducing the total wind pressure difference driving natural airflow in the dwelling.*

Passive ventilation stacks, like smoke stacks, can suffer momentary flow reversals or back drafting. At the component level, these flow reversals are simply due to reverse-flow pressure differences along the stack. At the system level potential causes become more complex. Changes in wind pressures acting on inlets, stack terminals, and leakage points can alter pressure distributions in the stack. A given system may be sensitive to passing turbulence and/or particular wind directions due to both its design and/or the sheltering effects of adjacent structures, vegetation, or topography. Mechanical systems or competing natural ventilation systems may also act to depressurize rooms served by a given stack and thereby induce the reverse flow. In field and laboratory tests, however, the tendency to back-draft is clearly related to three key factors (Parkins 1994; Welsh 1994; Welsh 1995; Welsh 1995; Welsh 1995) that add to the list o passive ventilation strategies:

> *Passive Ventilation Strategies*
>
> *To minimize reverse flow in passive stacks four factors are key:*
>
> *i) the height of the stack –taller stacks suffer fewer reverse-flow events,*
>
> *ii) the location of the stack terminal – stacks terminated above roof ridges perform better than those terminated below,*
>
> *iii) the detailed configuration of the stack terminal device, and*
>
> *iv) the thermal characteristics of the stack – stacks that can maintain elevated temperatures (e.g., insulated, massive and/or internal stacks) perform better.*

Welsh defines a wind performance indicator for stack terminals to be used in passive ventilation systems and a test methodology that provides the means to evaluate the back-draft potential of a given stack terminal device (Welsh 1995). Of the nine terminal devices Welsh studied, the so-called ‘‘H-pot’’ and ‘‘balloon’’ configurations were least likely to back draft. However, it may be unwise to jump to conclusions from these limited studies since the dynamic flow behavior of stack terminal devices is sensitive to subtle differences of geometry. In particular, the measured performance of two geometric similar H-pot terminals reported by Welsh were markedly different.

While reverse-flow in exhaust stacks may not be intended and, in some instances, may result in uncomfortable cold drafts and undesirable dispersion of moisture or odors, reverse-flow may still provides ventilation that can serve to improve air quality. Consequently, back drafting of PSV systems may amount to little more than a momentary loss of control but certainly not a failure of the system as it would be in a smoke stack.

2.1.2.4 Performance of Innovative PSV Systems

Initial studies of *innovative* PSV systems employing pressure-controlled self-regulating inlet devices have been very encouraging (Knoll and Kornaat 1991; Knoll 1992; de Gids 1997; de Gids 1998; de Gids 1998). They may well be able to eliminate the problems of under- and over-ventilation and the cold drafts associated with the latter under all climatic conditions. Traditional inlet vents with constant cross-sectional opening areas have pressure-flow characteristics similar to that of a simple orifice – the airflow rate through these devices increases proportionately with the square root of the pressure difference (approximately) across the device. Innovative inlet vents developed in The Netherlands and France, on the other hand, effectively maintain constant airflow rates over the range of pressure differences likely to be encountered in practice (i.e., 1 to 20 Pa). They have even been designed to reduce airflow rates at the higher pressures associated with the windiest conditions (i.e., above 10 to 20 Pa) to *compensate* for the increased infiltration that may be expected to result during these conditions, shown in Figure 2.4.

For the pressure differences likely to be encountered in North American construction (i.e., the 1 to 10 Pa range), the Dutch self-regulating inlet vent would seem ideal. At this point in time it seems to have only one small flaw – its internal mechanism tends to foul over time necessitating periodic cleaning to maintain performance. From these encouraging results a passive ventilation strategy may be formulated:

> *Passive Ventilation Strategy*
>
> *Pressure-controlled self-regulating inlet devices may well be able to eliminate the problems of over-ventilation and the cold drafts associated with it and minimize problems of under-ventilation if they effect their self-regulation at pressure differences likely to be experienced in the field (e.g. from 0.5 to 20 Pa).*

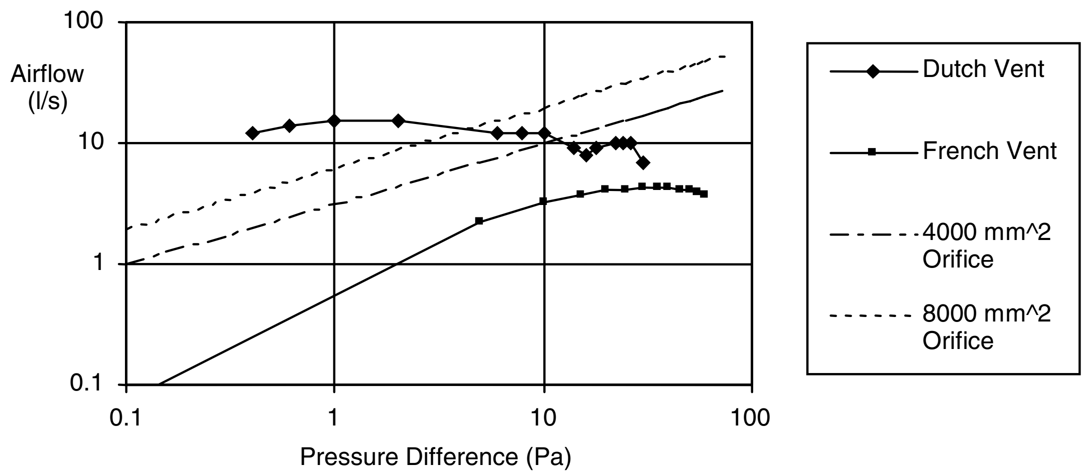


Figure 2.4 Comparison between the pressure-flow characteristics of self-regulating inlet vents developed in The Netherlands and France (de Gids 1997) with 4,000 mm^2 and 8,000 mm^2 orifice openings.

Other self-regulating inlet and outlet control devices have been considered. These include outlet devices that respond to internal humidity levels (Wouters and Vandaele 1990; Palin, McIntyre et al.

1996) and other air quality measures, inlet devices that respond to outdoor air temperatures, and occupancy sensors. Although these devices can serve to control airflow when natural driving forces are large enough to be throttled, they can not act to increase these forces under milder climatic conditions. Consequently, their actual annual efficacy remains uncertain and few design recommendations have come forward short of the BRE recommendation that relative humidity controlled outlets be set to start opening at 40% RH (Stephen, Parkins et al. 1994).

2.2 The PSVH Research Initiative

Energy use associated with ventilation systems falls into one of two categories – energy consumed to transport and energy consumed to condition the ventilation air. Energy consumed to condition ventilation air accounts for 30% or more of the total energy consumed for space conditioning in North American or European dwellings. The energy consumed to transport air in mechanical ventilation systems is typically a small fraction of this value although it may well still prove to be significant – e.g., in proposals to operate fans in conventional force-air systems fitted with outdoor air supply ducts continuously to induce minimal fresh air exchange rates.

PSV systems have one clear advantage over mechanical alternatives – by relying on natural forces they consume no energy to transport air. Mechanical ventilation systems, on the other hand, can be controlled to avoid over-ventilation, and consequently the energy use required to condition the unneeded ventilation air. They also can be fitted with heat recovery devices to conserve as much as 70% of the energy consumed to condition needed ventilation air.

Self-regulation of PSV systems may well provide the control to eliminate over-ventilation, and the energy penalty associated with it, but heat recovery has been practically ignored in the development of residential PSV systems (a PSV heat recovery system for office buildings has been investigated, however, (Skaaret, Blom et al. 1997)). Recognizing this shortcoming an international research program was initiated in 1996 – the SAVEHEAT research project funded in part by the Commission of the European Union – to develop innovative PSV systems with heat recovery devices identified as PSVH systems (Riffat and Gan 1997; Riffat, Gan et al. 1997; Sirén, Riffat et al. 1997; Etheridge and Zhang 1998).

Air-to-air heat recovery is not easily realized in PSV systems for three reasons. First, fresh air inlets are normally distributed and not placed physically close to the exhaust stacks, thus conventional heat recovery systems (e.g., side-by-side air-to-air or run-around-loop systems) can not be fitted to PSV systems. Second, heat recovery will lower stack temperatures and thus tend to reduce ventilation airflow rates. Finally, the heat exchange surfaces needed in air-to-air recovery devices will introduce an additional resistance to airflow in passive stacks that will tend to further reduce ventilation airflow rates.

The SAVEHEAT research team is investigating a number of truly innovative alternatives to solve these difficult inter-related problems. Current investigations include:

- *Heat-Pipe Heat Recovery Systems* Studies of a number of different configurations of air-to-air heat-pipe heat recovery devices have led to the development of devices providing relatively high heat recovery rates yet low resistance to airflow that appear to be well-suited to passive ventilation systems. These devices are intended to be installed in a PSV system having a central ducted fresh air intake stack that is placed adjacent to the exhaust stack to provide the adjacency needed. A two-story test cell has been constructed to investigate the efficacy of this strategy and the performance of the system has been measured. From these results the research team has concluded that acceptable heat recovery (i.e., from 20 to 55%) can be realized if stack velocities are kept below 1 m/s. Pressure losses due to both lowered temperatures in the stack and the flow resistance of the heat-pipes may, however, demand flow assistance. It is suggested that this could be provided by wind-driven turbines or solar heating of the upper part of the stack. (Riffat and Gan 1997; Riffat, Gan et al. 1997)

- *Solar-Assisted Natural Ventilation* Experimental and theoretical studies of a *solar chimney* – a passive exhaust stack design to collect solar radiation in a 2 m extension above the roof of a two zone test cell in Portugal – have demonstrated a 47% increase in passive stack airflows when compared to an identical two zone test cell without the solar collector. It is hoped that a properly design solar chimney will provide the necessary flow assistance to overcome the loss in ventilation flow due to the heat recovery device (Sirén, Riffat et al. 1997).
- *Wind-Turbine Assisted Natural Ventilation* A vertical axis wind turbine designed to drive a fan within the passive stack is also being investigated to provide flow assistance especially when buoyancy effects reach their minimum due to overcast condition for the solar chimney or mild outdoor temperatures. So far these studies have shown that a radial fan within the passive stack is far more effective than axial fan due to the dynamic matching of the ventilation fan to the driving wind turbine (Sirén, Riffat et al. 1997).
- *Dynamic Insulation as an Alternative Heat Recovery Strategy* The simultaneous conduction of heat and infiltration of air through building wall construction results in heat transfer between the airflow and the construction that serves to preheat incoming air during the heating season and to precool incoming air during the cooling seasons. Heat transfer between exfiltrating air and the wall construction creates the opposite effect. This phenomenon, which is now commonly called *dynamic insulation*, provides a form of heat recovery. The feasibility of employing this heat recovery strategy in passive ventilation system is currently being studied at Nottingham University (Etheridge and Zhang 1998).
- *Simulation Tools & Pilot Study* In addition to these individual studies computational simulation tools and a pilot plant are being developed to investigate the integrated application of these individual strategies.

Purpose-provided dynamic insulation may offer an attractive passive ventilation inlet strategy not only because of the heat recovery it can realize by, in effect, reducing conductive heat losses but because it may, conceivably, be integrated with transpired solar collectors (Kutscher, Christensen et al. 1993; Kutscher 1994; Kutscher and Christensen 1994; Kutscher 1996) to further enhance energy savings. The nature of the dynamic insulation phenomenon is not yet well understood however. Consequently the heat recovery that may be realized via this phenomenon may not be sufficient to warrant its use (Buchanan and Sherman 1998; Buchanan and Sherman 1998). Whether installed in a passive or mechanical ventilation system, the efficiency of air-to-air heat recovery depends critically on controlling (unintended) infiltration. Thus tight building construction is not only essential to effective control of passive ventilation it also results in maximizing the benefit heat recovery devices might offer.

The PSVH research initiative is directed to the single greatest shortcoming of passive ventilation – the challenge to passively ventilate with heat recovery. The initial results of this initiative are certainly encouraging, thus we may tentatively drawn a passive ventilation strategy from them.

> *Passive Ventilation Strategy*
>
> *While additional research and development is needed, heat recovery using innovative air-to-air heat recovery devices or purpose-provide dynamic insulation may be possible with passive ventilation system. Airtight building construction is required, however, for both mechanical and passive ventilation systems if the full benefit of heat recovery is to be realized.*

2.3 Related Design Issues

Passive ventilation systems are designed to provide ventilation for air quality control. Unfortunately, however, they may be expected to have additional impacts on the use and operation of residences related indirectly to their purpose. The confounding problem of cold drafts has been noted as one such related design issue, others will be enumerated in this section.

The recent publications of the International Energy Agency (IEA) Annex 27 *Evaluation and Demonstration of Domestic Ventilation Systems,* summarized in their final report (Månsson 1998), review a number of these related design issues including a) user needs and interactions, b) thermal comfort, c) acoustical isolation, d) architectural and construction complications and constraints, e) energy use, and f) life cycle costs. These publications outline the scope and complexity of these related design issues and attempt to provide the designer with a semi-quantitative method to rank their importance. As such, the design *Hand Book* published by Annex 27 offers a useful complement to this Technical Note (Månsson 1998).

The problem of overcoming technical and practical *barriers* to the use of natural ventilation in larger non-domestic buildings has been a central focus of the recent European NatVent™ project and the ALTERNER Programme of the European Commission (Allard 1998; Kukadia 1998; Liddament 1998; Wouters 1998). While the context of these studies pertains to ventilation for cooling of larger, non-domestic buildings, some of the related design issues are similar thus the interested reader may well benefit from these earlier publications.

2.4 Airflow Systems Interaction

To the naïve, a building is a closed system that protects the occupants from the extremes of the outdoor environment. In reality a building is a porous yet cellular open system subjected to airflows induced by purpose-provided passive ventilation, infiltration/exfiltration, heating ventilating and air conditioning, local exhaust, combustion device, plumbing, and fireplace subsystems. As each of these systems can act to pressurize or depressurize rooms in the house they necessarily interact with each other.

In the moderate and colder climates of North America, passive ventilation systems may be expected to depressurize the dwellings they serve. Consequently, they may act in competition with naturally vented combustion devices by decreasing smoke stack pressures and thereby increasing the chances of smoke stack back-drafting and spillage of combustion gases. Spillage of combustion gases may be mitigated through the careful design and selection of combustion stack components (de Gids and den Ouden 1974; Welsh 1995) or, more reliably, through the use of closed-system combustion devices that will not interact with the passive ventilation system. Passive ventilation systems may also induce airflows in nonoperating exhaust fans, duct networks of heating, ventilating and air conditioning subsystems, or chimneys that results in unintended infiltration of outside air (Shaw and Kim 1984). On the other hand, the operation of the dwelling heating, ventilating, and/or air conditioning system may induce unintended airflows in inlets or outlets of the passive ventilation systems (Persily 1998).

In warmer climates or during windy conditions, some passive ventilation systems may be expected to pressurize a house (e.g., if indoor air temperatures are maintained lower than outdoor temperatures). Again, as a result, unintended airflows may be induced in nonoperating exhaust fans, ducts, and chimneys. Furthermore, the migration of indoor moisture outward through wall construction may be enhanced and moisture damage may result.

The degree of pressurization or depressurization of a house by a passive ventilation system can, however, be controlled to some extent through the sizing of the components of passive ventilation systems and tight envelope construction. Furthermore, computational tools for macroscopic analysis can be used to investigate the interaction of the several airflow subsystems contained in representative dwellings (Persily 1998) to identify and rectify problems associated with airflow system interaction. To put this particular problem in perspective, however, it must be emphasized that a properly designed passive ventilation system installed in an appropriately tight building can not be expected to exceed the range of indoor pressures that may be found in traditionally constructed North American houses.

2.5 Comfort, Health & Safety

Passive ventilation systems may, as noted above, result in poor air distribution and cold drafts if not properly designed. Disruptive air velocities can also compromise comfort. A number of strategies have been put forward to mitigate these problems. If the designer chooses to supply air through the envelope of the building the use of low-velocity, low-induction vents is likely to solve problems of cold drafts locally. High-velocity, high-induction inlet vents, on the other hand, if positioned so that the supply direction is not likely to cause drafts, can provide better air distribution within a room (Månsson 1998). Automatic regulation of either type of vent can maintain whole-building air distribution objectives and practically eliminate over-ventilation on windy and/or cold days that is invariably associated with cold drafts.

Alternatively, the designer may elect to supply air using an internal air distribution system employing plenum (e.g., subfloor or access floor plenum connected directly to outdoor air inlets) or ducted systems. Both approaches will tend to temper incoming air via conduction transport from surrounding spaces. Furthermore, internal distribution systems may more readily be equipped with heat exchangers (e.g., possibly part of a heat recovery system) to more directly temper incoming air. The more elaborate internal distribution systems, while technically attractive, would normally be considered appropriate for larger buildings due to their complexity and higher first costs (Skaaret, Blom et al. 1997).

Rain entrainment and acoustical isolation must be considered in the design of any air inlet component whether for a passive or mechanical ventilation system. Security of inlets, or at least protection against animal and insect entry, is also an issue. These problems are not new nor particularly challenging – they may be solved using well-established component details developed by ventilation system component manufacturers.

Outdoor air pollution presents a greater challenge. Dust filtration could conceivably be provided at all passive ventilation inlets but the number and size of these inlets may prohibit developing a practically acceptable filter design solution. Gas phase filtration would be even more problematic. Small, distributed mechanical systems such as through-the-wall air conditioning units face the same problems that a distributed passive ventilation system would. Apart from filtration at the inlet devices the designer may position fresh air inlets to avoid potential pollutant sources (e.g., away from roads and parking areas, at elevated positions, and remote from trash or other storage areas). Again, internal distribution systems can more readily be designed to provide air filtration and even gas-phase cleaning but may not be appropriate for the residential scale.

Passive ventilation systems necessarily require fresh air inlets and free movement of this fresh air to exhaust vents or stacks. The free flow of air provided by passive ventilation systems can regrettably allow the free movement of smoke and enhance the spread of fires in dwellings. Maldonado notes building fire codes establish two generic class of regulations – envelope requirements intended to inhibit the spread of fires from adjacent structures and zoneing requirements intended to limit both fire and smoke within the building (Chapter 5 of (Allard 1998)).

In larger buildings, envelope requirements would demand that fresh air inlets be operable so that they could be closed in the event of a fire to inhibit smoke movement or to limit the supply of oxygen to a fire within. Furthermore, these inlets would be required to provide the same fire resistance, in the closed position, as the adjacent exterior wall. Beyond this, fire codes limit the location of inlets to areas remote from property lines to inhibit spread of fires from adjacent buildings. These fire safety measures can be readily implemented in the residential context for passive ventilation systems although relative to fire code provisions for other larger operable openings such as doors and windows these measures may seem unreasonably conservative. In any event, fire-safe envelope requirements for passive ventilation systems warrant careful consideration and further study. (This particular issue will be considered again in Chapter 6.)

Passive ventilation systems may be configured to divide a dwelling into separate fire zones. The BRE *best-practice* PSV system discussed above separates a dwelling into independent kitchen and bathroom zones that may act to limit the spread of smoke and fire in a house, even though that may not have been the original intention of the PSV system design. Again, the zoning potential of passive ventilation systems for fire safety also warrants careful consideration and further study.

3 Alternative Passive Ventilation Options

The PSV system has slowly evolved over the past century, largely via a trial and error process, to serve the relatively wet, temperate climatic conditions and dwelling sizes of Northern Europe – i.e., ranging on average from 65 m^2 (699 ft^2) to 130 m^2 (1399 ft^2) in the Northern European countries (Woolliscroft 1997). The North American context has some important differences in that it includes more extreme climatic conditions and larger dwelling sizes on average larger – i.e., ranging on average from 134 m^2 (1442 ft^2) in Canada to 152 m^2 (1636 ft^2) in the U.S. (Woolliscroft 1997). Consequently, it is useful to consider, briefly and rather speculatively, alternative system configurations and design strategies that may better serve these North American contexts.

3.1 Centralized Passive Stack Ventilation System

In Canada and the United States it has become common practice, generally enforced by local building regulations, to use local mechanical ventilation to control moisture and odors in kitchens and bathrooms. So the traditional European use of PSV system to directly ventilate these rooms is not required. There is, however, a need to more generally provide *"minimum ventilation rates and indoor air quality that will be acceptable to human occupants … to avoid adverse health effects"* (ASHRAE 1989) to all occupied rooms of dwellings. This could be provided by a centralized PSV system as shown in Figure 3.1 – here, utilizing an "H-Pot" stack terminal device that has demonstrated wind direction insensitivity and good resistance to momentary back drafting (Welsh 1994; Welsh 1995; Welsh 1995; Welsh 1995)).

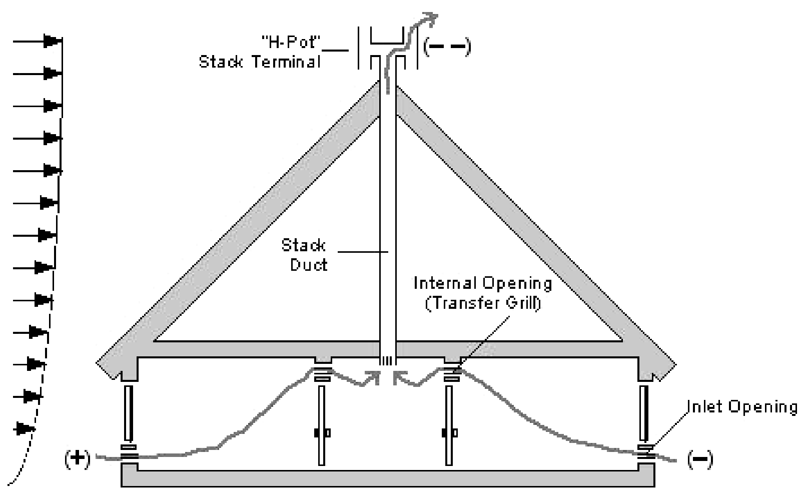


Figure 3.1 A centralized PSV system for general indoor air quality control utilizing an "H-Pot" stack terminal device to provide insensitivity to wind direction and resistance to back drafting.

3.2 Subfloor Inlet Plenum

Even greater insensitivity to wind direction can be achieved by linking inlet vents to a common plenum or duct network located below or between floors, Figure 3.2. This particular strategy also offers the possibility of tempering incoming cold air to avoid cold drafts and therefore is well suited to colder climatic conditions (Hayashi and Yamada 1996).

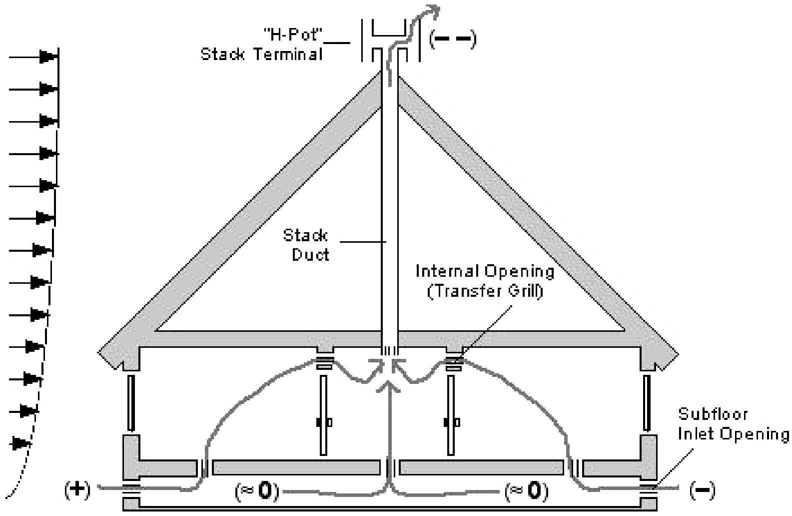


Figure 3.2 A centralized PSV system for general indoor air quality control utilizing a subfloor inlet plenum to provide insensitivity to wind direction and partial tempering of incoming air.

3.3 Balanced Wind Stack (BWS) System

During extreme summer conditions in the hot-humid regions of the U.S. indoor air will inevitably be mechanically cooled below outdoor air temperatures – nevertheless, ventilation for air quality control will still be needed. PSV systems have been developed for conditions where indoor air temperatures are consistently higher than outdoor air temperatures. Consequently if applied in these circumstances the buoyancy-driven airflows will act in opposition to the wind-driven airflows and the performance of the systems will suffer. Furthermore, as conventionally configured, PSV systems provide no means to dehumidify incoming air.

An alternative passive ventilation system that may be more suitable is a *balanced wind stack (BWS) system* that provides both inlet and exhaust air flow via stacks that terminate at or above the roof ridge level, Figure 3.3. This form of wind-driven system is currently being promoted by European manufacturers for larger building applications. BWS systems with concentric or adjacent supply and exhaust stacks offer the added possibility of installing sensible and latent heat recovery devices that should prove central to their use in hot, humid climates. The proximity of supply inlets and exhaust outlets of some commercially available systems may, however, make it difficult to develop the pressure differences needed to drive airflows to overcome the internal resistances of the systems.

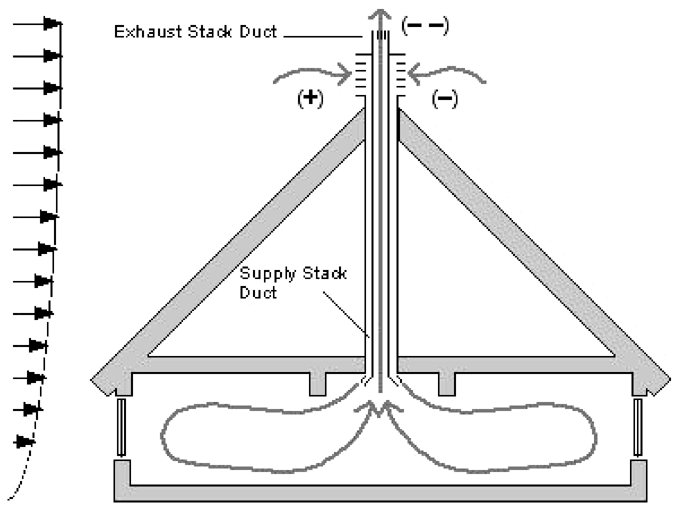


Figure 3.3 A centralized balanced wind stack (BWS) system for general indoor air quality control that offers the possibility of sensible and latent heat recovery.

Recent research on a related passive ventilation system has demonstrated that displacement airflow driven by thermal plumes of local heat sources (e.g., a stove in the residential context) can be sufficient to drive ventilation in a similar balanced stack configuration (Hunt and Holford 1998). These *top-down chimney* systems utilize a ‘‘top-down’’ supply stack and a vertical atrium space for exhausting air (that could also, presumably, be configured as a stack). With the results of this research in mind, it may prove advantageous to position room outlets to BWS exhaust stacks above residential heat sources in kitchens and bathrooms to create a PSV/BWS hybrid. Such a hybrid may better serve dwellings in hot humid climates.

BWS, top-down chimney, and related hybrid systems may usefully provide ventilation for air quality control in more urban or dense suburban situations where buildings effectively shelter each other from the wind. This would be similar to the traditional wind tower schemes of the Mid Eastern countries that have been used for ventilation cooling (Harris and Webb 1996).

3.4 Cross-Ventilation with Self-Regulating Vents

Finally for sites not sheltered by adjacent buildings, vegetation, or topography, traditional cross-ventilation cooling strategies may prove effective for air quality control if now combined with self-regulating inlet and outlet vent devices, Figure 3.4, based, presumably, on those considered above. Again, however, air distribution problems associated with wind direction sensitivity would have to be addressed to avoid the dual problems of over- and under- ventilation.

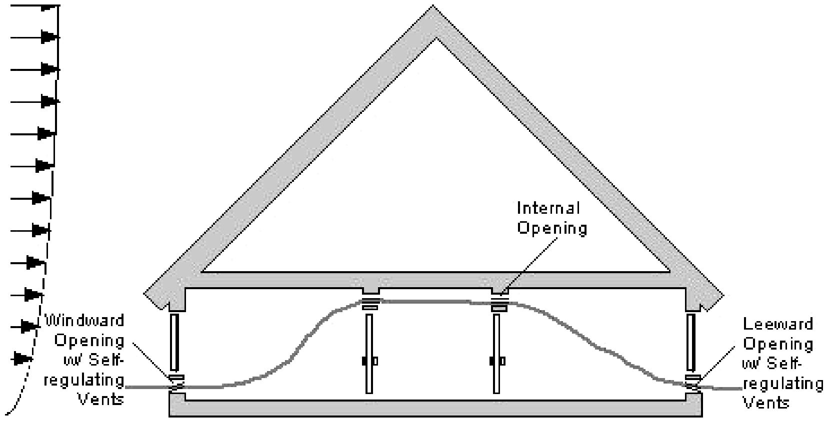


Figure 3.4 Wind-driven cross-ventilation with self-regulating inlet and outlet vents may transform the traditional cooling strategy for hot climates to one for air quality control with or without central air conditioning.

The design and annual performance of a representative wind-driven cross-ventilation system with self-regulating vents will be considered in Chapter 5 and compared to the design and performance of a conventional cross-ventilation system.

3.5 Conclusion

The alternative passive ventilation systems presented above offer representative examples of the much larger variety of systems that could be devised using the small number of generic passive components introduced so far. If mechanical components are added – i.e., to create either hybrid or mechanically-assisted passive ventilation systems – then the number of possible systems that could be considered would easily overwhelm any experimental program to evaluate their behavior. Clearly, the development of new passive ventilation systems can not proceed along the empirical path used to develop the PSV system – analytical tools are needed. Indeed, these tools are even needed for PSV systems – i.e., to predict the details of their behavior over long time periods for realistic environmental conditions and to size system components to achieve air quality design objectives. The next section will address this need.

4 Analytical Methods – Simulation & Design

Analytical methods or, equivalently, *mathematical models* serve one of two purposes – *simulation* or *design*. *Simulation methods* provide the means to predict the response of a building system to environmental excitation and occupant interaction. In the present context, simulation methods are used to predict indoor air quality, most fundamentally, or ventilation airflow rates over some chosen period of time for the building/passive ventilation system being considered. The environmental excitation would normally be specified in terms of time histories of wind speed and direction, thermal parameters such as outdoor air temperatures and solar radiation, and contaminant parameters such as contaminant emission rates, removal rates, sorption characteristics, and the like. In general, building occupants interact with the building system by, for example, altering ventilation openings, introducing internal heat gains, and introducing contaminant sources.

Design methods, on the other hand, are used to size system components to achieve reasonable design objectives – the *design criteria* – for given *design conditions*. In the present context, design criteria would be defined most fundamentally in terms of appropriate measures of air quality (e.g., relative humidity or CO_2 concentration) that should not to be exceeded. Alternatively, the design criteria could be defined, less fundamentally, in terms of code-specified minimum ventilation rates. Design conditions would be defined in terms of an appropriate environmental excitations and occupant actions (e.g., annual or critical shorter-time period records of weather and source emission rates). The tripartite character of design:

1. *sizing* system components
2. to satisfy design criteria
3. for reasonable *design conditions*

often proves to be so complex that no simple, closed-form mathematical procedure can be formulated for it. Consequently, design methods are invariably defined in terms of multi-step algorithms. Most often, individual steps within these algorithms are based on mathematical models used for simulation. Thus, there is a close relationship between simulation and design methods. Indeed, when all else fails simulation methods may be used directly in a trial and error manner (algorithm) to size system components – a common yet generally inefficient approach to design.

4.1 Simulation Methods

The number and variety of simulation methods that could conceivably be applied to passive ventilation analysis is so great as to be overwhelming. The scope, complexity, and utility of these possibilities, however, may be understood by classifying these methods by a) system idealization used and b) analytical category.

4.1.1 System Idealization

Simulation demands the formulation of an idealized representation of a building system – a *system idealization* – intended to capture the essential features of the system and the aspects of its behavior thought to be relevant. Commonly building systems are idealized using either *microscopic* or *macroscopic* idealizations.

For *microscopic idealizations,* a region of the building system is spatially subdivided into a 1D, 2D or 3D mesh. Schematic 2D microscopic idealizations are shown in Figure 4.1. The mesh defines discrete points or *nodes* (e.g., the mesh intersections) and small but finite control volumes (e.g., the mesh rectangles) that become the focus of microscopic methods. The region selected may be external to the building for *external flow analysis,* internal to the building structure for *internal flow analysis*, or one or more portions of the solid or porous solid regions of the structure for mass or heat transfer analysis.

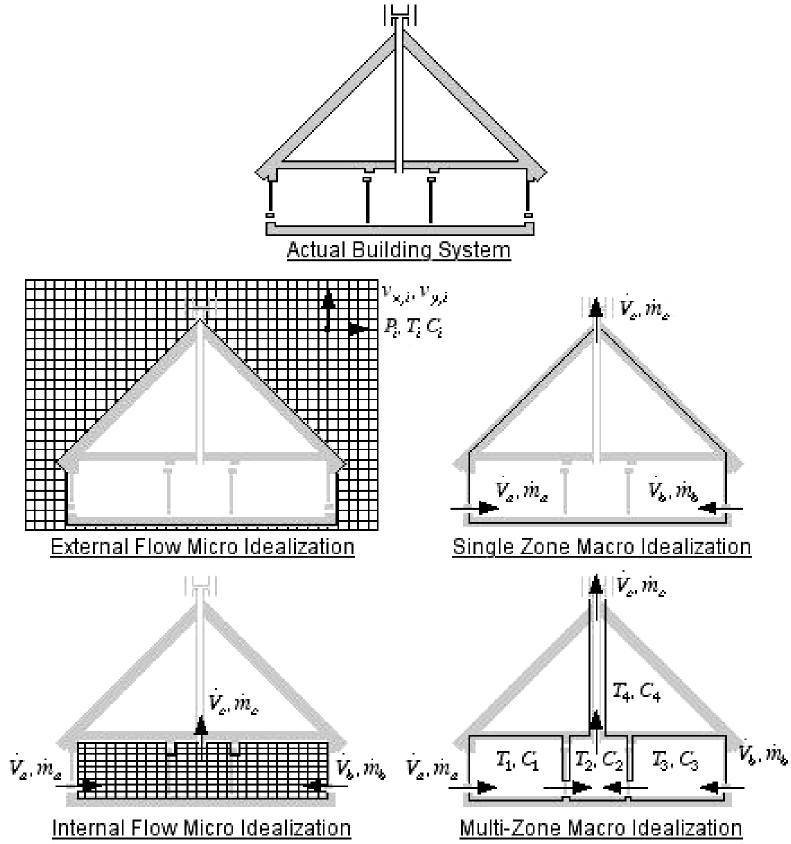


Figure 4.1 Possible microscopic and macroscopic idealizations of a centralized passive stack system .

For *macroscopic idealizations,* the building system is represented by a collection of *control volumes* (e.g., well-mixed *zones*) linked by discrete mass transport (e.g., ventilation airflows) and/or energy transport (e.g., heat conduction) paths. For example, the centralized passive stack system considered above might be idealized as a single control volume or a number of linked control volumes as shown diagrammatically in Figure 4.1.

Importantly, whether a microscopic and macroscopic approach is used a number of different idealizations is possible – each serving different analytical objectives, providing greater or lesser detail, and requiring more or less computational, personnel, and time resources. Although many idealizations may be inappropriate, no one idealization can be considered best or correct in an absolute sense.

4.1.2 Microscopic Modeling

For microscopic air quality and ventilation modeling, system behavior is described in terms of the fundamental variables of:

- *flow velocity* – air velocity in each of three orthogonal directions v_x, v_y and v_z,

- *temperature and concentration* – air temperature T and concentration C of air pollutants or water vapor at the *nodes*, and
- *pressure* – air pressure P at the *nodes* of the mesh.

With these fundamental microscopic variables defined, systems of algebraic and/or ordinary differential equations may be formulated using *discretized approximations* of general partial differential equations that describe mass, momentum or energy conservation relations at the microscopic level. The most familiar microscopic approach used for ventilation analysis is based on the Navier-Stokes equations for fluids. In the building field microscopic approximations of the Navier-Stokes equations are most often identified simply *as computational fluid dynamics* or *CFD* although strictly speaking the CFD field embraces other approaches as well.

Other less conspicuous uses of microscopic analysis include those based on the *diffusion* equations for solids and the *convection-diffusion* equations for porous solids that may be used to model heat and mass transfer in solids and porous solids respectively.

The details of the *discretized approximations* – most commonly *finite difference, finite volume,* or *finite element* approximations – are well beyond the scope of this Technical Note and need not be considered here. It is worth noting, however, that they can sometimes be formulated directly by applying the macroscopic conservation principles to the microscopic control volumes defined by the mesh. Thus, as microscopic and macroscopic models share the same fundamental basis – the conservation principles – they may be considered to differ only in the resolution of detail they provide.

The greater detail provided by microscopic analysis comes, unfortunately, at a cost. It is computationally demanding, as might be expected, and requires special expertise and considerable time to complete even the simplest analysis. Due to the former problem, internal flow microscopic analysis is presently limited to the investigation of a small number of inter-connected volumes of relatively simple geometry (e.g., two rooms, large or small, with limited or no furnishings) for simple stationary (i.e., steady excitation) conditions. Consequently, its application to complete building ventilation system analysis has been limited to simple building configurations (e.g., see (Barozzi, Nobile et al. 1991)) for a limited number of flow boundary conditions. Annual dynamic nonstationary internal flow microscopic analysis of even the simplest building system is simply out of the question at this time.

On the other hand, external flow microscopic analysis provides a relatively economic means, when compared to wind tunnel investigations, to predict wind pressure coefficients that may then be used in macroscopic analysis, but here the accuracy of the computed results has been mixed (Holmes and McGowan 1997). While the integration of external and internal microscopic analysis would appear to be ideally suited to passive ventilation simulation, again due to computational limits it is presently not feasible.

The special expertise and time required for microscopic flow analysis – and the cost associated with both – has limited this simulation approach, more or less, to research studies. For larger buildings, however, CFD is beginning to have a significant impact on the design of natural ventilation systems for cooling (Warburton 1992; Irving and Uys 1997). One must expect that with time whole-building microscopic techniques will find practical application in dwelling design as well. In the interim, microscopic analysis has been integrated with macroscopic simulation methods to allow analysis of the details of behavior of a portion a building system (e.g., a single room) while accounting for whole-system interaction to improve the accuracy of microscopic investigations (Axley 1988; Chen 1988; Kerestecioglu 1988; Li and Holmberg 1993; Schaelin, Dorer et al. 1993; Clarke and Beausoleil-Morrison 1997). In spite of the promise it holds, integrated microscopic and macroscopic analysis has yet to have any significant impact on passive ventilation research and practice.

4.1.3 Macroscopic Modeling

In macroscopic air quality and ventilation modeling, system behavior is described in terms of the fundamental variables of:

- *airflow rate* – the volumetric airflow rate $\dot{V}$ or air mass flow rate $\dot{m}$ for each of the discrete airflow paths or the ventilation air exchange rate (i.e., the outdoor air volumetric flow rate divided by the volume of a control volume) for selected control volumes,
- *temperature and concentration* – most often, the spatial average air temperature T, and concentration C of air pollutants or water vapor within each control volume, and
- *pressure* – P, defined at a specific point or node in the control volume and, most often, assumed to vary hydrostatically with elevation.

Theoretical models may be formulated that implicitly relate these variables to environmental conditions by applying mass, momentum or energy conservation relations to each of the control volumes included in the idealization. The resulting *system equations* may be solved analytically, for simpler cases, or numerically, for more complex cases, to predict the behavior of the system being studied. These models range from models based on detailed single-control volume idealizations – *single-zone models* – to more complete multi-control volume idealizations – *multi-zone models* – where individual heat and mass transfer paths are modeled explicitly.

Some multi-zone ventilation analysis software offers *fixed-form* idealizations using a set number of control volumes intended to model a certain type of building system. For example, Villenave, Millet, and Ribéron present a specialized model for PSV systems in multi-story apartment dwellings utilizing an idealization consisting of stacked two-zone unit models (Villenave, Millet et al. 1994). Other multi-zone software, based on *element* or *component-assembly* procedures, allows the analyst to formulate system idealizations of arbitrary complexity as deemed appropriate. These *general-form* multi-zone models include the CONTAM, COMIS, BREEZE, ESP, and IDA programs among others (Feustel and Raynor-Hoosen 1990; Hensen 1990; Solomons 1990; Feustel and Smith 1992; Sahlin and Bring 1993; Pelletret and Keilholz 1997; Walton 1997).

Alternatively, semi-empirical or correlation models, invariably based on single-zone idealizations, may be formulated. Using field measured data or data generated with more complete theoretical models these semi-empirical models may then be *calibrated* to develop mathematical relations that explicitly relate the fundamental macroscopic variables to environmental conditions. Of these, the LBL infiltration model has proven, perhaps, most influential being incorporated into the ASHRAE *Handbook of Fundamentals* and now part of LBL's RESVENT program. The British model BREVENT is similar in approach if not detail (Sherman and Modera 1984; ASHRAE 1989; ASHRAE 1997; Forowicz 1997). Wilson and his colleagues in Canada have taken their closely related model and extended it to account for purpose-provided passive ventilation components to create the modeling tool LOCALLEAKS (Wilson 1992; Walker and Wilson 1994) – elements of this particular model will be discussed in more detail below.

Finally, some recent research has been directed to the problem of modeling imperfect mixing within individual rooms in buildings. A number of different models, commonly identified as *zonal models,* have been proposed to capture some of the flow details of imperfectly mixed rooms without the computational expense of detailed microscopic analysis. In zonal models, rooms are idealized by assemblages of two or more control volumes instead of one and physically consistent relations are devised to describe heat and mass transfer between the modeled control volumes (Inard and Buty 1991; Inard and Buty 1991; Allard 1998). While considerable progress has been made, at this time zonal models remain in the developmental stage. If reliable zonal models emerge, they should prove particularly useful for the analysis of passive ventilation strategies relying on airflow driven by thermal plumes within rooms such as the *top-down chimney* systems discussed above.

4.1.4 Summary – System Idealizations

The range of system idealizations currently being used may be organized as illustrated in Figure 4.2. Liddament (Liddament 1986; Liddament 1997), Awbi (Awbi 1991), Santamouris (in (Allard 1998)) and the ASHRAE *Handbook of Fundamentals* (ASHRAE 1989; ASHRAE 1997) provide useful reviews of the simulation methods developed for infiltration and ventilation analysis for each class of system idealization.

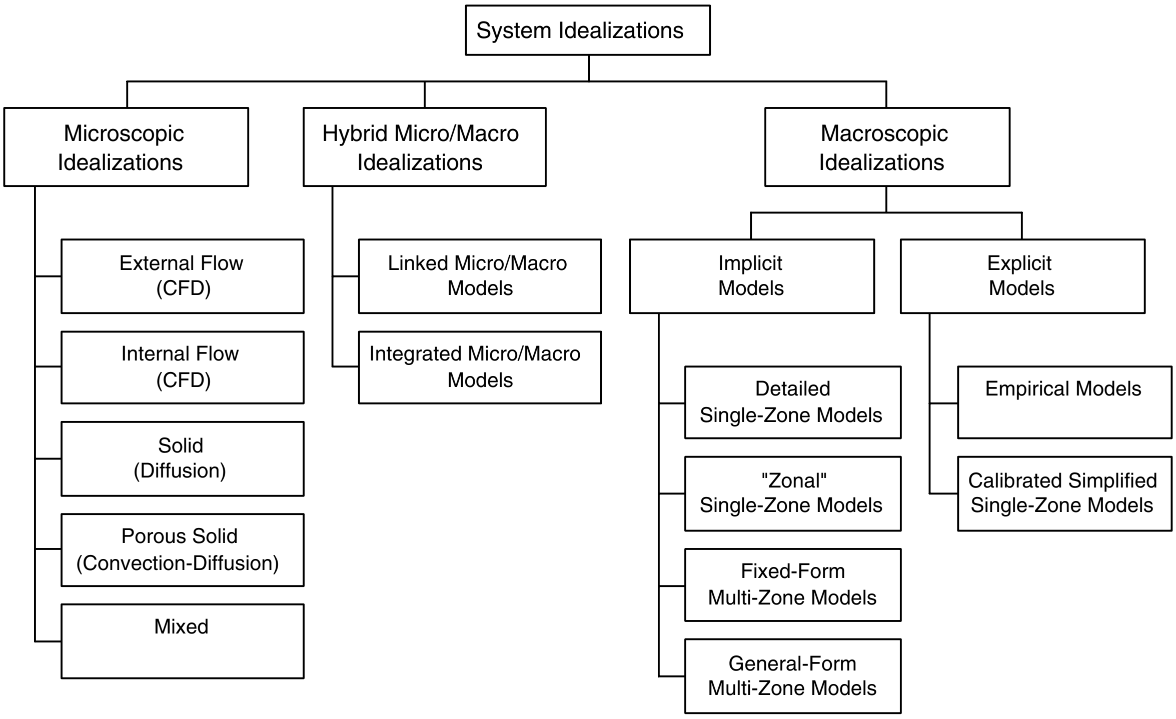


Figure 4.2 System Idealization Catgegories

4.1.5 Analytical Categories

Microscopic and macroscopic models may be formulated for each of the analytical categories enumerated in Figure 4.3.

Simulation theory has been developed in each of three broad analytical categories:

1. *contaminant dispersal analysis* – the prediction of air pollutant or moisture levels in a building given a) building characteristics, b) air quality environmental conditions, c) source generation characteristics, and d) airflow or ventilation flow time histories,
2. *airflow or ventilation analysis* – the prediction of building airflow or ventilation flow rates given a) building characteristics, b) wind and temperatures fields inside and out, and c) mechanical system operating characteristics and criteria, and
3. *thermal analysis* – the prediction of indoor air temperatures and/or heating or cooling energy requirements given a) building characteristics, b) internal heat generation characteristics, c) thermal environmental conditions, d) building airflow or ventilation flow rates, and e) mechanical system operating characteristics and criteria.

For each of these branches of analysis, system equations may be formulated, in principle, for steady or dynamic conditions of system excitation. For the former, the equations describe the *steady-state* (*stationary* for microscopic flow analysis) response of the system while for the latter, the *dynamic* response of the system. Although possible, dynamic formulations of the macroscopic airflow analysis problem are very seldom considered. Building airflows respond so rapidly to changes in driving pressures that it is commonly considered adequate to compute the steady-state (or stationary) response at each time step of a dynamic excitation record to approximate the dynamic variation of building airflows – an approach that may be identified as *quasi-steady dynamic analysis*. For detailed analysis of the dynamics of stack backdrafting, however, the inertia of the air enclosed within a stack contributes a significant delay to airflow response and, thus, a dynamic formulation of the problem is required (Etheridge 1998).

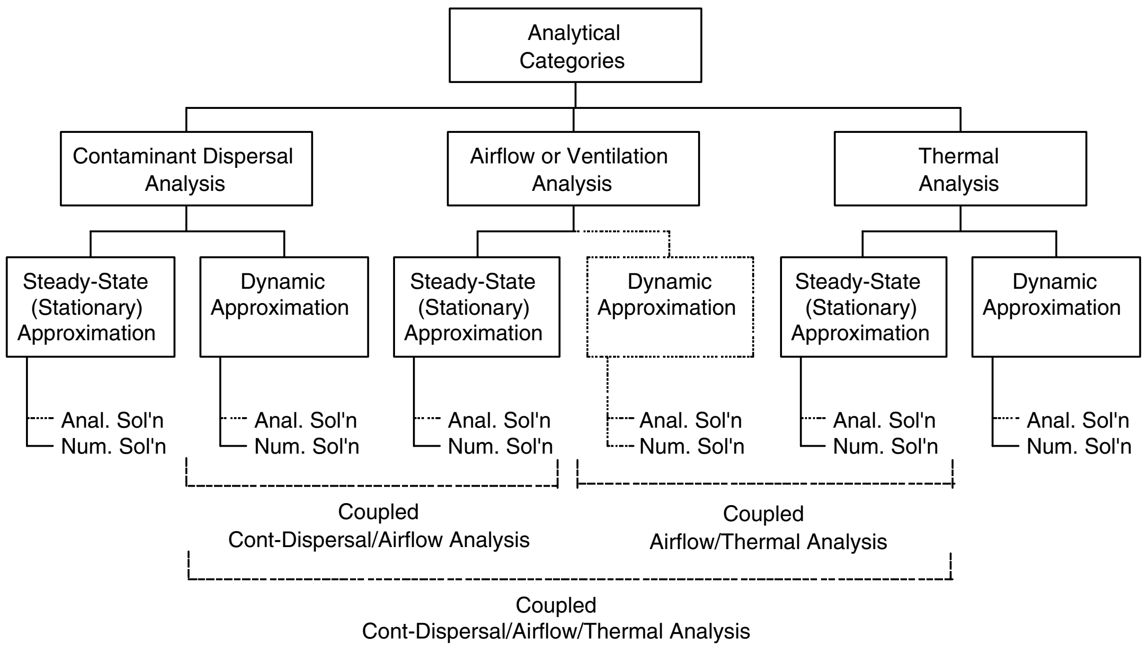


Figure 4.3 Analytical Categories
(Dashed lines indicate options possible in principle but uncommon in practice.)

For semi-empirical macroscopic models, the system equations explicitly provide (an estimate of) the system response. For example, the LBL infiltration model directly provides an estimate of the infiltration volumetric airflow rate of a single-zone representation of a building given building characteristics and environmental conditions. For all other cases, the system equations implicitly define (estimates of) the system response to environmental excitation and thus must be solved. In some simpler cases, formal mathematical *analytical* solutions may be derived but most often, the analyst must accept *numerical* solutions.

Although the theory relating to each of the three major categories of analysis has evolved independently, in the most general and exact formulation all three types of analyses are *coupled*. That is to say, airflow is in part dependent on buoyancy forces which, in turn, are dependent on variations of air density which, in turn, result from both temperature and air contaminant level variations in a building. Conversely, both temperature and air contaminant levels depend on the convective airflows in a building system. Consequently, we may distinguish three additional classes of analysis – *coupled contaminant dispersal/airflow analysis, coupled airflow/thermal analysis,* and *coupled contaminant dispersal/airflow/thermal analysis* – although the coupling of contaminant dispersal analysis is seldom considered significant and is, therefore, generally ignored for air quality analysis. This common assumption may not be appropriate for moisture transport and is certainly inappropriate for smoke spread analysis as moisture changes and smoke generation can produce significant variations in air density and thus buoyancy forces in buildings.

These analytical distinctions are not simply an academic matter – coupling of these theories results in computational complexities that, minimally, demand additional time and resources to address and, possibly, may prove insoluble. In any event, to fully evaluate the efficacy of ventilation system for air quality control, contaminant dispersal analysis should be employed as it directly determines (i.e., predicts) the quality of air and its variation with time. As contaminants disperse primarily due to airflow, airflow analysis would then be required as well (i.e., even if modeled as uncoupled) and in realistic cases this airflow is likely to be significantly coupled to the thermal response of the building system.

Presently, however, the sources and health effects of indoor air pollutants are not yet known with the certainty needed for determinate analysis. Thus for the time being, the evaluation of ventilation systems is most often based on airflow analysis alone – hence the emphasis placed on nominal ventilation airflow rates in current building standards – and less often on the dispersal of an air contaminant (e.g., moisture or carbon dioxide as general surrogates for air quality).

4.1.6 Common Assumptions for Residential Ventilation Analysis

A number of simplifying assumptions is commonly made for residential ventilation analysis.

4.1.6.1 Airflow/Thermal Analysis Coupling

In North America homes, indoor air temperatures are normally maintained within a limited range by heating or cooling systems. Consequently, the coupling between airflow and thermal analysis is often ignored and thermal analysis is not generally considered when investigating air quality and ventilation behavior. This common assumption, however, may not be acceptable during the shoulder seasons when mechanical heating or cooling is not used and indoor air temperatures are allowed to float freely.

4.1.6.2 Contaminant Dispersal/Airflow Analysis Coupling

Indoor air pollutant levels remain at trace levels. Consequently, changes in buoyancy due to density variations in air pollutants can be ignored quite reasonably. Realistic changes in moisture levels can, however, result in air density changes of the same order of magnitude as those due to temperature differences, yet these are also commonly ignored. This common assumption should also be reconsidered.

4.1.6.3 Contaminant Dispersal Analysis

Contaminant dispersal analysis is commonly ignored altogether and consideration is limited to airflow analysis alone for the evaluation of residential ventilation systems for air quality control. While a number of exceptions to this rule can be noted in the research literature – e.g., Persily's multi-zone annual evaluations considering carbon monoxide, volatile organic compounds, particulates, nitrogen dioxide, and moisture (Persily 1998); de Gids' multi-zone evaluations of innovative PSV systems considering carbon dioxide (de Gids 1997), Yuill and Wray's radon mitigation and demand control ventilation studies (Wray and Yuill 1990; Yuill, Jeanson et al. 1991; Wray and Yuill 1992), and other studies considering moisture (Edwards and Irwin 1988; Jardinier and Simonnot 1990; Woolliscroft 1994; Palin, McIntyre et al. 1996) – equations published for practical system evaluation (e.g., (Awbi 1991; ASHRAE 1997; Allard 1998)) are generally limited to airflow analysis. This, as noted above, is due in part to the general indeterminacy of air quality analysis at this time and the emphasis placed on ventilation airflow rates in building ventilation standards.

4.1.7 Outline of Macroscopic Airflow Analysis Theory

Presently, steady macroscopic airflow simulation – single or multi-zone – is the method of choice for research investigations of residential ventilation systems. Macroscopic simulation remains too complex for most practical uses although two exceptions to this general rule should be noted: an internet implementation of LBL's RESVENT program and a new Dutch program NavIAQ (Forowicz 1997; de Gids 1998; de Gids 1998). However, the underlying theory of steady macroscopic airflow analysis may be used to develop a practical design method that will be presented subsequently. Consequently, it will be useful to outline the general features of macroscopic airflow simulation theory here.

Given a macroscopic idealization of a house – for example, the four-zone idealization of the centralized passive stack system shown in Figure 4.4:

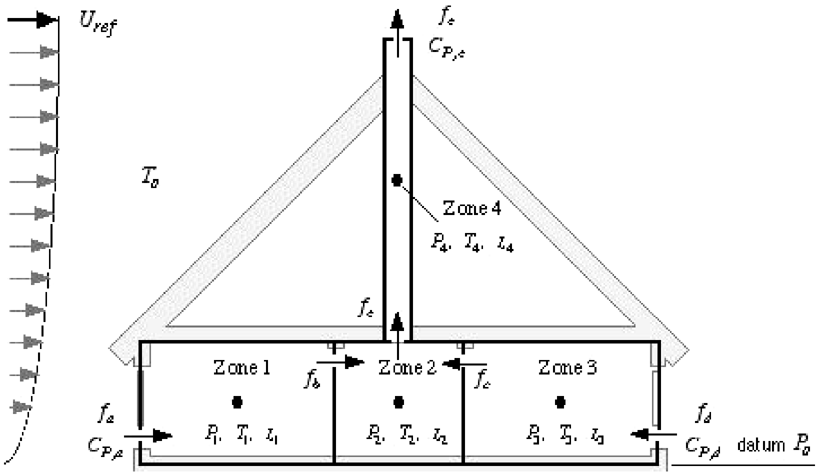


Figure 4.4 A representative four-zone model of a building with a centralized passive stack system.

- each zone is numbered and a specific location is identified within each zone – the zone node,
- an unknown pressure variable P_i and a known temperature T_i variable is associated with the zone nodes *"i"*,
- each discrete airflow path is identified – in Figure 4.4 the flow paths are identified by letters *a, b, c, …* – and a unique mathematical component flow model is associated with each flow path, also identified as a flow component or element,
- environmental conditions are established including an outdoor datum and associated pressure P_o, an outdoor air temperature T_o, and a reference wind speed U_{ref} and direction ϕ, and
- surface-averaged wind pressure coefficients are assumed, given the reference wind direction, for the exterior surface locations of each of the envelope flow components – in Figure 4.4 these are identified as $C_{P,a}, C_{P,d}$, and $C_{P,e}$ for locations *a, d,* and *e*.

4.1.7.1 Component Flow Models

Component flow models associated with each discrete flow path, say *"l"*, relate the volumetric flow rate through the component $\dot{V}_l$ to the pressure drop across the component ΔP_l. Specific flow models will be presented in the following sections. Here, we may represent these flow models with a simple functional notation as:

$$\text{General Form of Component Flow Models}\ \dot{V}_l = f_l\left(\Delta P_l\right) \tag{4.1}$$

Flow models $f_l\left(\Delta P_l\right)$ are most often nonlinear algebraic expressions of the pressure drop across the flow path ΔP_l.

4.1.7.2 Surface Wind Pressures

At surface locations external to envelope flow components, wind-driven pressures $P_{l,o}$ are related to the ambient pressure $P_o(z_l)$ at the component level z_l and the dynamic pressure of the oncoming wind defined in terms of a surfaced-average wind pressure coefficient C_{Pl} and the dynamic pressure of a corresponding reference wind speed U_{ref}:

$$P_{l,o} = P_o(z_l) + C_{Pl}\frac{\rho U_{ref}^{\ 2}}{2} \tag{4.2}$$

where ρ is the density of the outdoor air. Commonly, the reference wind speed is taken as the upwind, undisturbed wind speed at the building height or building eve height (Orme, Liddament et al. 1998; ASHRAE 1999) but this convention is far from universal. The analyst must be careful to determine and apply the reference wind speed actually used to determine any given set of wind pressure coefficients.

At surface locations where the wind pressure acts inward, the wind pressure coefficient is positive with absolute value less than or equal to 1.0. At locations where the pressure acts outward in suction this coefficient is negative with absolute value usually less than 1.0 but possibly greater. The magnitude of surface wind pressure coefficients depends, in general, on the geometric configuration of the building and its details and the sheltering effects of nearby structures and vegetation.

As wind pressure coefficients determine the driving forces behind wind-driven ventilation, these geometric, detail, and sheltering conditions become critical design issues in the design of passive ventilation systems. Passive stack terminal devices are particularly important in this regard. De Gids and Den Ouden at TNO in The Netherlands, concerned with conditions that can lead to backdrafting in both passive ventilation and combustion device stacks, made detailed measurements of wind pressure coefficients of some thirteen different stack terminal devices (de Gids and den Ouden 1974). They accounted for the influence of ascending and descending winds (i.e., relative to the vertical stack axis) and, importantly, the influence of stack flow rates on the wind pressure coefficients which proved to be significant in many instances. Similarly, Welsh at the BRE in Britain measured wind pressure coefficients for twelve different stack terminal devices. As part of this work he developed a useful *wind performance indicator* (WPI) to rate stack terminal devices accounting for wind direction sensitivity of the devices and their tendency to foster backdrafting or not (Welsh 1994; Welsh 1995; Welsh 1995; Welsh 1995). Both TNO and BRE studies clearly indicate subtle differences in stack terminal device design can have a profound impact on their performance. Seemingly similar devices may perform quite differently due to small details in their construction so broad general design recommendations may not be made. Rather, the interested reader should consult the TNO, BRE, and related publications directly when faced with the task of selecting stack terminal devices.

4.1.7.3 Buoyancy Pressures

Pressure changes $\Delta P_e(\Delta z_{i,j})$ due to elevation changes $\Delta z_{i,j}$ from location "*i*" to location "*j*" within a component or zone "*e*" or the outdoor environment are defined by the discrete form of the hydrostatic equation for the usual assumption of uniform temperature distributions:

$$\Delta P_e(\Delta z_{i,j}) = -\rho_e g \Delta z_{i,j} \tag{4.3}$$

or by the integral form for nonuniform temperature distributions:

$$\Delta P_e(\Delta z_{i,j}) = -\int_{z_i}^{z_j} \rho_e(z) g dz \tag{4.4}$$

where, ρ_e is the density of air within the component, zone, or outdoor environment, and *g* is the standard acceleration due to gravity (i.e., 9.8 m/s^2). The air density may be determined from the component, zone or outdoor air temperature using the ideal gas law, which for standard dry air at (or near) atmospheric pressure is:

$$\rho_{dry-air}(T_{\circ C}) = \frac{352.6 \text{ K} \cdot \text{kg/m}^3}{\left(T_{\circ C} + 273.15\right) \text{ K}} \tag{4.5}$$

It will be shown in the design methods section of this Technical Note that these hydrostatic changes determine the driving forces behind buoyancy-driven (stack) ventilation. Furthermore, the stack pressures due to these changes are directly related to stack height and temperature differences between indoors and out. Consequently, to maximize stack pressures the designer can maximize stack height and maintain high stack air temperatures using insulation or, possibly, enhance stack temperatures with heat recovered from chimneys or absorbed from solar radiation.

4.1.7.4 System Equations

For a system idealization of *n* zones (or, more generally, control volumes) *n* equations in the unknown *n* zone pressures may be formed by simply demanding conservation of the mass flow rate of air into and out of each zone. The mass flow rate $\dot{m}_l$ of a given flow component is simply the product of the component volumetric flow rate and the density of the air passing through the component:

$$\dot{m}_l = \rho_l \dot{V}_l (4.6) \tag{4.6}$$

Consequently, the system equations are formed by summing all component mass flow rates for each of the zones and demanding the zone sums each equal zero as:

Zone 1:
$$\sum_{\substack{\text{connected}\\\text{components}}}^{\text{zone 1}} \left(\dot{m}_l\right) = 0 \tag{4.7a}$$

Zone 2:
$$\sum_{\substack{\text{connected}\\\text{components}}}^{\text{zone 2}} \left(\dot{m}_l\right) = 0 \tag{4.7b}$$

etc.

Zone n:
$$\sum_{\substack{\text{connected}\\\text{components}}}^{\text{zone } n} \left(\dot{m}_l\right) = 0 \tag{4.7c}$$

The component mass flow rates or, by Equation 4.6, volumetric flow rates depend on the component pressure drops ΔP_l which may be expressed in terms of the known outdoor pressure P_o, known wind pressure terms, $C_{Pl}\left(\rho U_{ref}^{\;2}/2\right)$, known buoyancy pressure terms, $\rho_e g \Delta z_{i,j}$, and the *n* unknown zone node pressures $\{P_1, P_2, \ldots P_n\}$. Consequently, Equations 4.7 lead to *n* equations in the *n* unknown zone node pressures. This set of *n* equations in *n* unknowns will be identified as the *system equations*.

These system equations may then be solved to determine the zone pressures and finally these values may be used to determine the component flow rates (i.e., using Equations 4.1 through 4.5 above). These flow rates fully characterize the ventilation behavior of the building (i.e., for the assumed environmental conditions) and may be used to determine whole-building as well as individual zone air exchange rates. While the procedure outlined is simple in concept, the details are complex and the solution of the resulting system equations is computationally challenging. That is to say, this procedure must be implemented computationally for all but the simplest cases.

4.1.8 Applications to Passive Ventilation Systems

With all said and done, the practical application of simulation methods to passive ventilation system evaluation has been more or less limited to steady-state or quasi-steady dynamic macroscopic airflow analysis. Of this modeling subset, empirical or semi-empirical methods can not be expected to be useful unless passive ventilation systems are included in the studies used to evaluate the empirical model parameters, hence, these methods have not been used. Calibrated simplified single-zone models extended to included detailed representations of passive ventilation components, detailed single-zone models, and multi-zone airflow models have been applied with reasonable success and will be considered in this section.

4.1.8.1 Calibrated Simplified Single-Zone Models

The simplified single-zone models – e.g., the US Department of Energy LBL Infiltration Model and RESVENT model, the Building Research Establishment BREVENT model, the Canadian models that led to the LOCALLEAKS model, and the model included in the ASHRAE *Handbook of Fundamentals* (Sherman and Modera 1984; Wilson 1992; Cripps and Hartless 1994; Walker and Wilson 1994; ASHRAE 1997; Forowicz 1997) – vary in their details but share the common feature that they account for infiltration using lumped or effective leakage characteristics of building envelope areas rather than explicitly modeling individual infiltration leakage paths.

Effective leakage characteristics of whole-buildings and building components are measured using pressurization tests. In these tests, a controlled air pressure ΔP_l is maintained across the whole-building envelope or component, say component *"l"*, and the volumetric flow rate $\dot{V}_l$ through the envelope or component is measured. This procedure is repeated for a number of imposed pressure differences (e.g., in the range of 1 to 50 Pa) and the resulting data is commonly fitted to pressure-flow relations of the so-called *power-law* form – a two-parameter flow equation:

Power-Law Equation $$\dot{V}_l = C_l \Delta P_l^{n_l} \qquad (4.8)$$

$n_l \approx 0.6 \pm 0.1$ for cracks and joints (Orme, Liddament et al. 1998)

$n_l \approx 0.63 - 0.13 \log_{10} C_l \left(\frac{\mathrm{dm}^3}{\mathrm{s} \cdot \mathrm{m}^2 \cdot \mathrm{Pa}^{-\mathrm{n}}} \right)$ for porous surfaces (Orme, Liddament et al. 1998)

The power-law coefficients C_l and n_l thus determined define the building or component leakage characteristics. A large number of component and whole-building leakage data bases have recently been compiled by the AIVC that may be used directly for practical evaluation of the infiltration characteristics of buildings (Orme, Liddament et al. 1998). The statistical analysis of this data reveals that while the measured flow exponent *n* varied over a range of approximately 0.5 to 1.0 the majority of the data clustered around the value of 0.6 for building cracks and joints and varied inversely with the power-law coefficient *C* for porous surfaces (Orme, Liddament et al. 1998). (These results are consistent with theoretical expectations. The lower flow rates observed through porous media are expected to be more laminar in nature and thus, by the Couette flow equation, may be expected to varying linearly with pressure difference (Awbi 1991)).

The US LBL and ASHRAE infiltration models (Sherman and Modera 1984; ASHRAE 1997) are single-zone macroscopic models that lump enveloped leakage characteristics into a so-called effective leakage area, A_{eff}, distinguishing floor, ceiling, wall, and whole-building leakage areas – $A_{eff\text{-}f}$, $A_{eff\text{-}c}$, $A_{eff\text{-}w}$ and $A_{eff\text{-}b}$ respectively. For a given envelope area, say area *"l"*, the volumetric airflow rate $\dot{V}_l$ due to a pressure drop ΔP_l across the area is estimated using the *effective leakage area equation* – a single-parameter flow equation:

Effective Leakage Area Equation $$\dot{V}_l \approx A_{eff-l} \sqrt{\frac{2\Delta P_l}{\rho}} \qquad (4.9)$$

where ρ is the density of air in the flow path. The geometric and leakage path detail of a building is, in effect, idealized as a simple rectangular box with the composite of all leakage airflow paths in ceiling, walls, or floor represented by single flow relations for each area respectively – i.e., as illustrated in Figure 4.5.

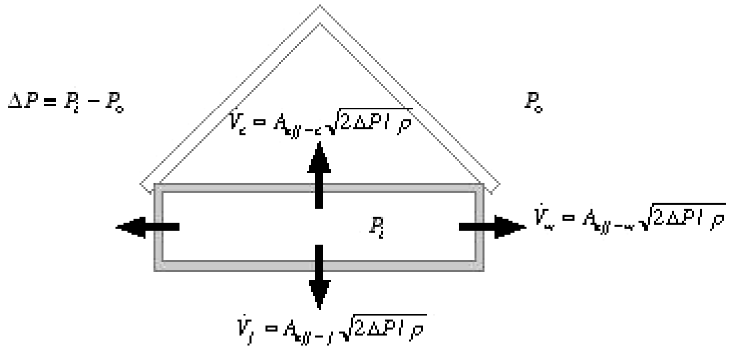


Figure 4.5 US Effective Leakage Area Building Idealization
(Schematic representation during building pressurization test.)

As this idealization establishes the form of the relation between building ventilation rate and indoor-outdoor temperature and wind speed, one may use field-measured ventilation rate data and weather records to establish correlation coefficients between the former and latter. That is to say, one may *calibrate* the idealization to measured data. This was done to establish the LBL and ASHRAE infiltration models. The details of the calibration procedure used will not serve our purposes here. Suffice it to say the resulting models have proven to be reliable tools to quickly estimate likely ventilation rates under representative design conditions for typical dwellings of conventional construction. They are not, however, directly useful for passive ventilation analysis or design without modification.
*
The simple infiltration model developed by the Building Research Establishment (BRE) is similar to the US models in that a simple idealization is used as the basis of the model which then is calibrated against measured data to evaluate model parameters (see (Awbi 1991)). The original BRE infiltration model is based on the power-law flow equation, however, with all envelope leakage lumped into a single whole-building flow relation $\dot{V}_{bldg} \approx C_{bldg}\Delta P^n$ as illustrated schematically in Figure 4.6.

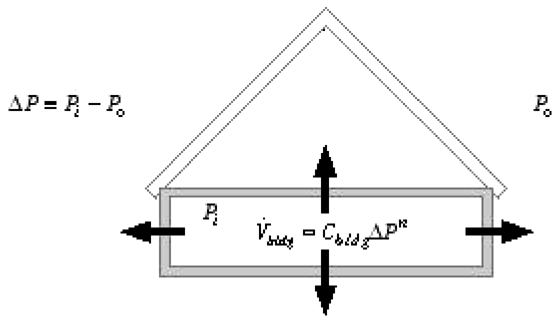


Figure 4.6 BRE Power-Law Building Idealization
(Schematic representation during building pressurization test.)

The simpler single parameter effective leakage area flow equation can not be made equivalent to the power-law equation in general – except for the special case when $n = 0.5$. Equivalency can be established at a single reference pressure difference ΔP_{ref} by equating Equation 4.8 to 4.9 and solving to obtain:

$$A_{eff\text{-}l} = C_l \Delta P_{ref}^{\;n_l - 0.5}\sqrt{\rho/2} \qquad (4.10)$$

In the LBL and ASHRAE infiltration models, a reference pressure of 4 Pa is selected to transform building and component pressurization test results to the simple equivalent leakage form. Figure 4.7 compares the pressure-flow curves for the equivalent leakage area model with A_{eff} = 1.0 and the power law model with a flow exponent, n = 0.6 and C adjusted by Equation 4.10 to provide equal flows at the 4 Pa US standard reference pressure.

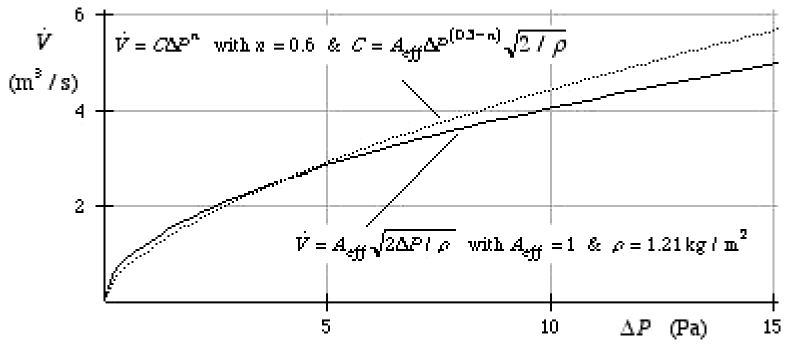


Figure 4.7 Comparison between the effective leakage area and power-law component flow models adjusted to be equal at ΔP=4 Pa.

Building envelope components may be expected to experience pressure differences of up to 10 Pa most usually and higher occasionally. Consequently, it is clear from this figure that the use of the 4 Pa reference pressure in the LBL and ASHRAE simplified infiltration models is well-justified.
The effective leakage area equation is a close-cousin of the classic orifice equation that may be used to describe the pressure flow relations through simple openings (e.g., larger cracks, trickle vents, windows, etc.):

Classic Orifice Equation $$\dot{V}_l = C_d A_l \sqrt{\frac{2\Delta P_l}{\rho}} \quad (4.11)$$

$$C_d \approx 0.6 \text{ for turbulent flow conditions}$$

where C_d is the orifice *discharge coefficient* which has a value of approximately 0.6 for the turbulent flow encountered in all but the smallest building envelope openings and A_l is the cross-sectional area of the opening *"l"*. Comparing Equation 4.11 to 4.9, we conclude that the effective area of an orifice is simply:

$$A_{eff-l} = C_d A_l \approx 0.6 A_l \quad (4.12)$$

A typical two-story detached US home with a useable floor area of 152 m^2 (1636 ft^2) (Woolliscroft 1997) would have a whole-house effective leakage area of 1,400 cm^2 (1.5 ft^2) (i.e., based on the US average normalized leakage value of *NL* = 1.2, a dimensionless measure of building leaskage area, reported by Sherman and Matson (Sherman and Matson 1997)). The same house built to current ASHRAE recommended airtightness standards will have a whole-house effective leakage area a fraction of this value (i.e., with NL values ranging from 0.10 to 1.13 in North America) (ASHRAE 1988). If fitted with a BRE *best-practice* PSV system with 4,000 mm^2 inlets in a kitchen and bath, 8,000 mm^2 inlet vents in each of five additional rooms and two stacks with total cross-sectional area of 20,000 mm^2 an additional effective leakage area on the order of 40,000 mm^2 or 400 cm^2 will be added to the building system. Clearly, the impact of a PSV system must be expected to be significant and even dominant for a tighter home. Consequently, the simplified infiltration models discussed above can not be expected to predict the ventilation behavior of dwellings with passive ventilation systems without modification.

4.1.8.2 Detailed Single-Zone & Multi-Zone Models

Such modifications have been implemented in the Canadian simplified infiltration models (Wilson 1992; Walker and Wilson 1994). The lessons learned from the Canadian work are particularly instructive here. Wilson and Walker modified their simplified power-law infiltration model by explicitly adding flow components for passive inlets and stacks – i.e., as shown schematically in Figure 4.8. In this model individual envelope areas are modeled with the power-law flow equation (e.g., the east *"E"*, west *"W"*, floor *"f"*, and ceiling *"c"* flow relations shown) and passive inlets and the stack were modeled using the orifice flow relation (e.g., with the east *"*$A_{I,E}$*"*, west *"*$A_{I,W}$*"*. and stack *"*A_s*"* areas as shown). The latter is justified as the stack was fitted with a sharp-edged orifice restriction.

In the actual implementation of this model, all distributed leakage was combined by assuming the same flow exponent *n* for all envelope areas and summing the flow coefficients as:

LocalLeaks Infiltration Model

$$\dot{V}_{\text{infil}} = \dot{V}_f + \dot{V}_c + \dot{V}_{w,W} + \dot{V}_{w,E} + \ldots = (C_f + C_c + C_{w,W} + C_{w,E} + \ldots)\Delta P^n \qquad (4.13)$$

A similar approach will be used in a design method to be presented subsequently to account for infiltration, although the distributed leakage will not be combined as presented here.

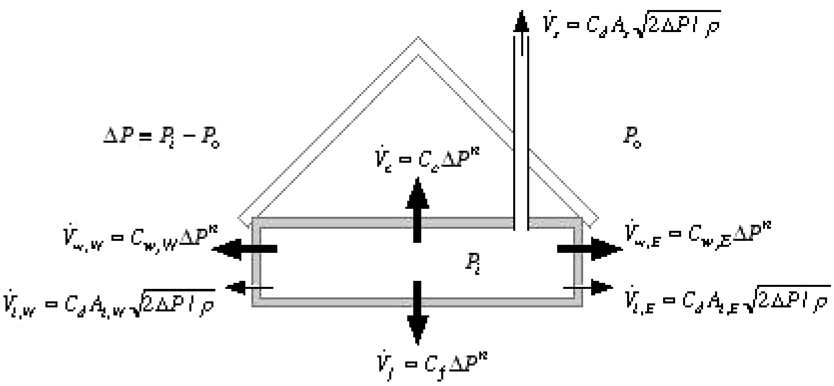


Figure 4.8 Schematic of the LOCALLEAKS building idealization.
(Schematic representation during building pressurization test.)

Using this modified idealization, the Canadian research group found they could reliably simulate measured response if a) wind pressure coefficients, b) wind direction, and c) wind sheltering of adjacent structures were modeled correctly. They used wind pressure coefficients that accounted for upwind built obstacles – specifically for individual units of row houses with wind-flow along the row they argued a side-wall wind pressure coefficient of –0.2 was more appropriate than the value of –0.65 published for isolated buildings. They introduced an empirical relation to provide a continuous model of the directional dependency of wind pressure coefficients C_P with wind direction ϕ using four measured values of these coefficients as:

$$C_P(\phi) = \tfrac{1}{2}\Big[\big(C_P(0°) + C_P(180°)\big)\left|\cos(\phi)\right|^{\frac{1}{2}} + \big(C_P(0°) - C_P(180°)\big)\cos(\phi)\left|\cos(\phi)\right|^{\frac{1}{4}} \\ + \big(C_P(90°) + C_P(270°)\big)\sin(\phi)^4 + \big(C_P(90°) - C_P(270°)\big)\sin(\phi)\Big] \qquad (4.14)$$

Finally, they accounted for wind sheltering effects common to urban sites using a shelter factor $S_{U,}$ with values ranging from 0.0 to 1.0. This factor is used to reduce the estimated wind speed at eve height U_H (i.e., in an unsheltered location) to a more representative value for the sheltered location U:

$$U = S_U U_H \qquad (4.15)$$

The refinement of the single-zone building idealization that is the basis of the LOCALLEAKS model is essentially a transformation from a simplified to a detailed single-zone model. These same conclusions could be drawn, apparently, for modifications made to the BRE model that has been modified to better account for PSV system interaction as well. The recent trend toward detailed multi-zone analysis of building ventilation systems has also emerged out of the need to better account for the detailed configuration of building systems (Bassett 1994; Bassett 1994; Heikkinen and Pallari 1994; Månsson 1998; Persily 1998). Specifically, we may conclude:

- Reliable analysis of passive ventilation systems demand reasonably detailed deterministic models.
- Deterministic analysis of building ventilation systems is particularly sensitive to wind modeling details.

Fürbringer's conclusions, based on an investigation of the sensitivity of both simple and detailed macroscopic models to uncertainty in wind model parameters, underscore the second conclusion above (Fürbringer 1994) as do the reports of Roulet, Santamouris and Bassett (Bassett 1990; Roulet, Fürbringer et al. 1996; Santamouris, Argiriou et al. 1996).

Available general purpose, component-assembly, air-quality and ventilation modeling tools such as the COMIS and CONTAM family of programs may be used directly to investigate single-zone passive ventilation idealizations like that used in the LOCALLEAKS development (Feustel and Raynor-Hoosen 1990; Feustel and Smith 1992; Walton 1994; Pelletret and Keilholz 1997; Walton 1997). They may be used for multi-zone idealizations as well. A number of investigators have done just this, benefiting from the libraries of flow components, wind modeling capabilities, and robust and efficient numerics imbedded in these programs and their abilities to compute annual simulations (Bassett 1994; Bassett 1994; Heikkinen and Pallari 1994; Månsson 1998; Persily 1998).

As these programs demand time and expertise that few building designers can be expected to have, special purpose programs based on the same underlying modeling theory have been developed to provide more practical alternatives (Villenave, Millet et al. 1994; Forowicz 1997; de Gids 1998; de Gids 1998). While it is not appropriate to consider the details of these computational tools here, it will be useful to consider a number of additional pressure-flow relations that may be used in these programs as these same relations may be used to develop a design method for sizing passive ventilation system components.

4.1.9 Passive Ventilation Component Models

Power-law, equivalent-leakage area, and the classic orifice equations have been presented above, Equations 4.8, 4.9 and 4.11. The orifice equation may be reasonably employed to model the pressure-flow behavior of constant area passive inlets and outlets, as well as door and window openings, that act to restrict airflow through openings with cross sectional dimension large in comparison to their length along the flow direction. Alternatively, the power-law relation may be fitted to measured data for each of these elements – again, the AIVC compilation of measured data for such components is available for this very use (Orme, Liddament et al. 1998). In addition, model parameters for each of these relations may be transformed to the alternate forms using Equations 4.10 and 4.12 if desired.

4.1.9.1 Quadratic Component Model

A rigorous and detailed analysis of flow through narrow openings leads to the *quadratic model*:

$$\dot{V}_l = \frac{1}{2}\frac{\sqrt{4\beta\Delta P_l + \alpha^2} - \alpha}{\beta} \tag{4.16}$$

where α and β are model parameters that are directly related, theoretically, to the geometry of the narrow opening being considered or may be estimated by fitting this model to measured data (Etheridge and Sandberg 1996; Orme 1998). The inverse form of the quadratic model reveals its simplicity and essential character:

$$\Delta P_l = \alpha\dot{V}_l + \beta\dot{V}_l^2 \tag{4.17}$$

This model, thus, combines a linear relation, able to better capture low flow conditions through a flow path, with a quadratic relation. Sound theoretical and experimental evidence supports the use of the quadratic relation, especially for modeling adventitious openings in building enclosures, yet the modeling community has been slow to embrace this component model (Etheridge 1998). This illogical and curious state-of-affairs is likely due to the path of historical development in the field and the relative complexity of the forward form of this model, Equation 4.16. The inverse form, Equation 4.17, is, however, well-suited to the design method presented in this Technical Note.

The quadratic model also offers a significant, albeit esoteric, numerical advantage that proves to be important in computational implementations – the derivative of the volumetric flow rate with respect to the pressure difference $d\dot{V}_l/d\Delta P_l$ is bounded for this model at near-zero flow rates - precisely because this model has a physical theoretical base. The power-law, equivalent-leakage area, and the classic orifice models don't offer this advantage.

As useful as the power-law, equivalent-leakage area, orifice and quadratic models may be, they are not suitable for modeling ventilation stack components or newer, innovative self-regulating inlet or outlet vents.

4.1.9.2 Self-Regulating Inlet Vent Components

Knoll and de Gids have developed a detailed macroscopic modeling tool for the evaluation of ventilation systems utilizing the self-regulating vent they developed that, apparently, includes a component model for this vent (Knoll and Kornaat 1991; de Gids 1997; de Gids 1998; de Gids 1998; Knoll and Phaff 1998). Unfortunately, the details of this component model have not yet been published. An alternative, empirical model is proposed here of the following form:

$$\textit{Self-Regulating Vent Model} \quad \dot{V} = \dot{V}_o \left(1 - \exp\left(\frac{-\Delta P}{\Delta P_0} \right) \right) \tag{4.18}$$

where:

$\dot{V}$ is the volumetric flow rate through the self-regulating vent,

$\dot{V}_o$ is an empirical model parameter that corresponds to the maximum possible flow rate through the self-regulating vent – the nominal controlled flow rate,

ΔP is the pressure drop across the self-regulating vent, and

ΔP_o is an empirical model parameter that corresponds, approximately, to the threshold pressure drop above which the vent acts to regulate the flow to $\dot{V}_o$.

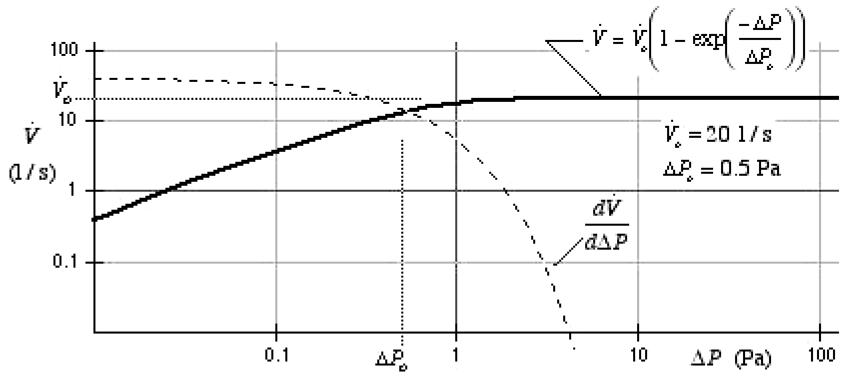


Figure 4.9 Plot of the proposed self-regulating vent model – solid line. The slope of the proposed model is also plotted – dashed line.

Figure 4.9 plots the pressure-flow behavior of this model for $\dot{V}_o$ = 20 l/s and ΔP_o = 0.5 Pa illustrating how the flow asymptotically approaches $\dot{V}_o$ above the effective threshold for control.

This proposed model was fitted to measured data for both the Dutch and French self-regulating vents discussed above. Figure 4.10 illustrates the results of this exercise comparing the proposed empirical model to both measured data and the orifice model for the 4,000 mm^2 and 8,000 mm^2 inlet vents recommended for PSV systems by the BRE.

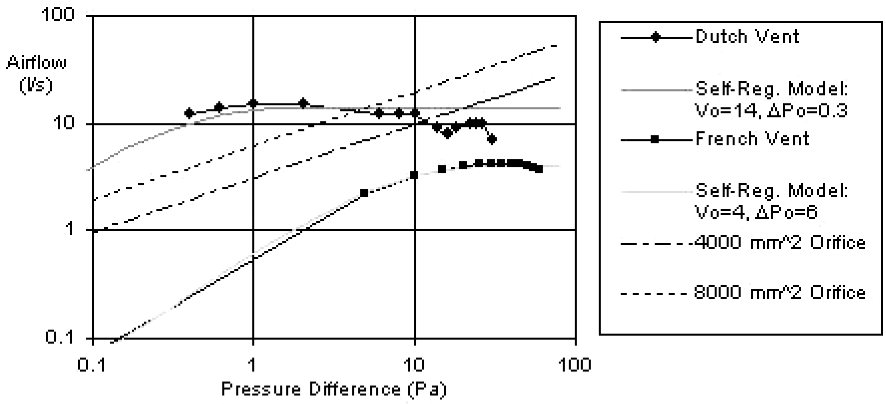


Figure 4.10 Comparison of proposed empirical self-regulating vent model, Equation 4.18, with measured data (de Gids 1997) and orifice models for the 4,000 mm^2 and 8,000 mm^2 inlet vents.

As may be seen, the fit of this model to the measured data is quite good. If implemented in a general multi-zone ventilation modeling program, however, the model would have to satisfy three additional numerical constraints:

4. it would have to have bounded derivatives (i.e., for common numerical methods used to solve the resulting system equations),
5. it would have to be able to model reverse flow that, realistically, would be regulated to values well below the forward-flow control limit $\dot{V}_o$ and
6. it should provide a smooth transition from forward to reverse flow.

The derivative of the empirical model, as illustrated in Figure 4.9, satisfies criteria i). It approaches a constant $\left(\dot{V}_o/\Delta P_o\right)$ as $\Delta P \to 0$ and approaches zero for large values of ΔP:

$$\frac{d\dot{V}}{d\Delta P} = \left(\frac{\dot{V}_o}{\Delta P_o}\right)\exp\left(\frac{-\Delta P}{\Delta P_o}\right) \tag{4.19}$$

The second and third criteria may be satisfied if reverse flow is modeled with the same empirical form (i.e., scaled by negative one) using: a) a limiting reverse-flow rate $\dot{V}_{o,r}$ much smaller that the forward-flow rate $\dot{V}_{o,f}$ and b) selecting the reverse flow threshold pressure so that the slopes are equal at the origin as:

$$\frac{\dot{V}_{o,r}}{\Delta P_{o,r}} = \frac{\dot{V}_{o,f}}{\Delta P_{o,f}} \tag{4.20}$$

For example, if the reverse-flow limit for the self-regulating vent is set to 1/10th the forward flow-limit (i.e., $\dot{V}_{o,r} = \left(1/10\right)\dot{V}_{o,f}$) and the slope criteria, Equation 4.20, is satisfied, then the reverse-flow pressure threshold would have to be set to 1/10th the forward-flow threshold as:

$$\dot{V}_{o,r} = \frac{\dot{V}_{o,f}}{10} \text{ and } \frac{\dot{V}_{o,r}}{\Delta P_{o,r}} = \frac{\dot{V}_{o,f}}{\Delta P_{o,f}} \quad \text{thus} \quad \Delta P_{o,r} = \frac{\Delta P_{o,f}}{10}$$

For the self-regulating vent illustrated in Figure 4.9, then, we would have:

$$\dot{V}_{o,r} = \left(\tfrac{1}{10}\right)\ 20\ \text{l/s} = 2\ \text{l/s} \quad \text{and} \quad \Delta\text{P}_{o,r} = \left(\tfrac{1}{10}\right)\ 0.5\ \text{Pa} = 0.05\ \text{Pa}$$

Figure 4.11 presents plots of both the forward- and reverse-flow relations for this example:

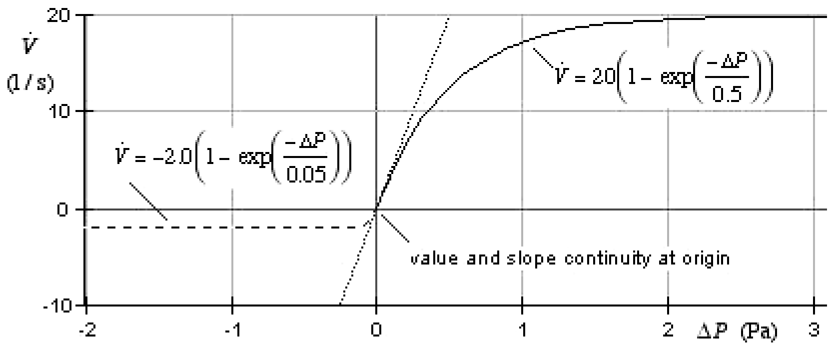


Figure 4.11 Forward- and reverse-flow self-regulating vent models for the example discussed satisfying value and slope continuity at origin.

4.1.9.3 Stack Components

Passive ventilation stacks are assembled from simple duct sections linked together by a variety of *fittings* – entries, exits or stack terminal devices, elbows, transitions, junctions, and obstructions (e.g., insect screens). Each of these fittings and the lengths of ducts connecting them provide some resistance to airflow due either to a) viscous (frictional) losses or b) *dynamic* losses. Dynamic losses are associated, fundamentally, with changes of the kinetic energy of the air flowing through a duct and result from changes in flow direction and/or velocity due to the geometric constraints a fitting places on the flow. The pressure-flow characteristics of duct networks have been thoroughly studied and practical flow models are well-established, although reasonably complex. The modeling approach incorporated in the ASHRAE *Handbook of Fundamentals* will be used here (ASHRAE 1989; ASHRAE 1997).

4.1.9.4 Constant Cross-Section Duct Length Components

Pressure losses in reasonably straight segments of constant cross-section ducts may be shown to be a simple sum of a viscous loss contribution and dynamic loss contributions. By isolating the entry and exit loss contributions as *fitting* losses, we may model the losses or, inversely, model the pressure flow relation for a given duct segment *"l"* as:

$$\dot{V}_l = \sqrt{\frac{D_h}{f\,L}}\,A\sqrt{\frac{2\,\Delta P_l}{\rho}} \tag{4.21}$$

where:

D_h is the so-called *hydraulic diameter* of the duct equal to the area of the cross-section divided by its perimeter,

f is the dimensionless friction factor,

L is the length of the duct segment, and
A is the cross-sectional area of the duct.

As presented, this relation has the same form as the classic orifice equation, Equation 4.11, with the first term on the right hand side $\sqrt{D_h/f\,L}$ equivalent to the orifice discharge coefficient C_d. Unfortunately, the friction factor f contained in this term is not constant - it varies with the intensity of the flow in the duct as measured by the Reynolds number Re (see Equation 4.23). For our purposes, it will be sufficiently accurate to consider two distinct flow regimes – *laminar* and *turbulent* flow – and corresponding friction factor relations for each:

Laminar Flow $f = 16/\mathrm{Re}$ (4.22a)

Turbulent Flow $f \approx 0.0791/\mathrm{Re}^{1/4}$ for smooth ducts (4.22b)

The first relation, Equation 20a, may be derived from the Hagen-Poiseuille formula and the second relation, Equation 4.22b, is known as the Blasius formula (Bird, Stewart et al. 1960).

The Blasius formula is reasonably accurate up to $\mathrm{Re} \approx 10^6$ for smooth ducts. For airflow in passive stacks the flow is likely to vary over two orders of magnitude centered on the intersection of these relations (i.e., Re = 1190*).* Consequently*,* the friction factor may be expected to fall within the range of 0.01 to 0.05 as illustrated in Figure 4.12.

The Reynolds number may be defined in terms of the volumetric flow rate as:

$$\mathrm{Re} = \frac{\rho\, D_h \dot{V}_l}{\mu\, A} \tag{4.23}$$

where μ is the viscosity of air which is approximately 18 Pa·s for dry air at room temperature.

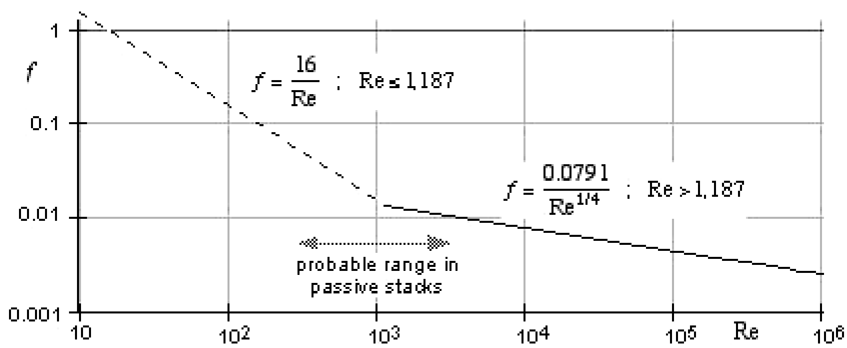


Figure 4.12 Variation of friction factor with Reynolds number for smooth-wall ducts based on the Hagen-Poiseuille and Blasius formulas (Bird, Stewart et al. 1960).

Thus, substituting these friction factor relations into Equation 4.21 we obtain duct flow relations for each flow regime:

Laminar Flow

$$\dot{V}_l = \left(\frac{D_h^{\ 2}}{8 \mu L} \right) A \, \Delta P_l \qquad (4.24a)$$

Turbulent Flow

$$\dot{V}_l \approx 6.33 \left(\frac{D_h^{\ 5}}{\rho^3 \mu L^4} \right)^{1/7} A \, \Delta P_l^{\ 4/7} \qquad (4.24b)$$

For computational analysis, these (and more complete and complex) formulations can be readily implemented. For manual calculations on the other hand the turbulent flow relation is inconveniently complex. Given the range of flow intensities likely to be encountered in passive stacks, however, the laminar flow expression, Equation 4.24a, may prove to be sufficiently accurate. Alternatively, the general form, Equation 4.21, may be employed by assuming a reasonable value for the friction factor (e.g., f • 0.03). If accuracy is critical, one may iteratively assume a friction factor, compute the flow rate, evaluate the Reynolds number and an improved estimate of the friction factor, and recompute.

4.1.9.5 Duct Fitting Components

Duct fitting pressure losses are commonly related simply to the kinetic energy of the flow through the fitting. Consequently, the flow relations for practically all fittings have the simple general form of (for a fitting *"f"*):

Duct Fittings Flow Model

$$\dot{V}_f = \frac{1}{\sqrt{C_f}} A \sqrt{\frac{2 \Delta P_f}{\rho}} \qquad (4.25)$$

where C_f is a dimensionless fitting loss coefficient.
The ASHRAE *Handbook of Fundamentals* (ASHRAE 1997) provides a compendium of fitting loss coefficients. While these coefficients are intended to be used for transitional and turbulent flow, they may be sufficiently accurate for passive ventilation analysis. Furthermore, in the design of passive stacks it is best to reduce the number of fittings and use low-loss fittings to minimize the resistance to airflow. Thus, any error introduced using the ASHRAE data should be small by design.

Stack entries and terminal devices are particularly relevant because they can not be eliminated altogether. Thus low-loss configurations should be employed. For example, a stack entry configured to be flush with the ceiling plane would have a loss coefficient of 0.50 if cut square at the entry while if fitted with a smooth converging bellmouth the loss coefficient can drop to a value of 0.03.

Welsh at the BRE measured flow characteristics of seven stack terminal devices for 110 mm diameter round stacks and six devices for 150 mm diameter round stacks including *H-pots*, self-vaning rotating cowls, two standard gas flue caps, a *mushroom* cap, a *balloon* cap, a *Chinese hat*, and an *aerodynamic* cowl (Welsh 1994; Welsh 1995; Welsh 1995; Welsh 1995). While all had loss coefficients greater than the open stack alone (i.e., with a loss coefficient of 1.0), when backdraft potential and wind suction characteristics are considered the advantages of some of these terminal devices becomes clearer. The low loss of an open stack is useful for buoyancy-driven flow but an open stack can not provide the wind-driven suction that other devices can and is more likely to suffer backdrafting.

4.1.10 Modeling Uncertainties

Simulation involves a) the idealization of a given building system, b) the representation of the excitation that drives the response of the building system, and c) the formation and solution of the system equations derived from the given building idealization and the representation of the system excitation. Consequently sources of uncertainty in simulation may be traced to errors or uncertainties in a) building idealization, b) excitation representation, and c) numerical methods used to form and solve the system equations.

Clearly the amount of system detail represented in a given building idealization determines the level of detail and consequently the spatial certainty of computed response results. Thus if an analyst chooses, for example, to model a room as a well-mixed zone, computed zone air temperatures and contaminant concentrations will, in some sense, represent only spatial average conditions in the zone and the actual spatial variation of these response variables will not be determined. In general, the analyst will elect to employ simpler building idealizations to reduce computational and personal effort and will therefore accept greater spatial uncertainty in the computed results. This source of uncertainty is implicitly defined by the nature of the building idealization used and therefore may be controlled by the analyst to the extent that the building system may be well characterized. Unfortunately, the detailed configuration and distribution of building leakage airflow paths will generally be difficult to characterize forcing the analyst to employ approximate, lumped-parameter idealizations of these important airflow paths as presented above.

There are limits here however. While microscopic idealizations offer greater spatial resolution, current computational limits limit their application to the analysis of relatively simple spatial domains that inevitably correspond to a small part of a whole building system (e.g., one or two well-connected rooms). For this reason macroscopic analysis remains the only choice for whole-building system simulation. Within this limitation the analyst may still elect to form macroscopic models of the whole-building system with greater or lesser detail and, as a consequence, obtain, in principle, greater or lesser uncertainty in computed response results.

Beyond the spatial resolution of the building idealization, the analyst must consider the appropriateness of the component or element models used. Regrettably, all component models in macroscopic airflow analysis are empirically based and thus even in ideal circumstances can model component behavior only approximately. In microscopic airflow analysis empirical *turbulence models* that are generally required for building studies similarly introduce uncertainty. In both micro and macroscopic simulation the skill and experience of the analyst is of paramount importance as an inappropriate selection of component models in macroscopic analysis or turbulence model in microscopic analysis can result in unacceptable errors in computed response results.

Beyond problems of system idealization, error may be introduced via the numerical methods used to form and solve the resulting system equations. Airflow analysis proves to be particularly challenging in this regard – coupled airflow and thermal analysis even more so – for both microscopic and macroscopic simulation studies. Again the skill and experience of the analyst is critical, although most simulation programs provide warnings of some numerical difficulties that may be expected to produce completely erroneous results.

At this point in the development of building simulation methods, however, representation of the system excitation is likely to introduce the greatest uncertainty in computed results. Most importantly, detailed time histories of wind speed and direction are generally not available for most sites and the nature of site sheltering by nearby buildings and vegetation will introduce additional uncertainty. Likewise, while ambient levels of many air contaminants may be estimated with certainty for all but catastrophic events, indoor source locations and emission rates are far more difficult to characterize. On the thermal side, however, solar radiation and ambient air temperatures and relative humidities are likely to be known with greater certainty.

Beyond the uncertainty introduced by lack of detailed knowledge of site wind speed and direction, wind turbulence is known to add to the problem. The theory outlined above, which is presently the basis of most airflow analysis simulation tools, tacitly assumes wind-driven airflows are due only to mean wind pressures exerted on external surfaces of buildings. Yet it is well-known that wind-induced turbulence can also induce ventilation and infiltration airflows (Knoll and Phaff 1998). Sirén has estimated that infiltration exchange rates may be underestimated by as much as 20% due to this oversight and has formulated an approach to correct macroscopic modeling methods for it (Sirén 1997). DeGidds and Phaff also present a method to correct for turbulence-induced airflow by adding an (effective) turbulence-induced pressure to that due to the mean wind speed (see § 3.1.1.4 of (Allard 1998)).

While these several sources of error may prove fatal for a given simulation analysis, a number of validation studies have demonstrated that macroscopic analysis may be relatively reliable under ideal circumstances (i.e., for well-defined systems subjected to well-characterized excitations), but uncertainty in system idealization and environmental excitation often prove critical (Liddament 1983; Bassett 1990; Fürbringer, Dorer et al. 1993; Fürbringer, Roulet et al. 1996; Fürbringer, Roulet et al. 1996; Roulet, Fürbringer et al. 1996). Likewise, for well characterized spaces idealized with a sufficiently fine mesh and carefully modeled boundary conditions (i.e., the system excitations of microscopic analysis) microscopic analysis has also been proven to produce accurate results.

4.2 Design Methods

Ventilation design methods must be understood in terms of both purpose and process. As to purpose, design methods are used to size ventilation system components to satisfy some reasonable design criteria (e.g., an air quality measure or a nominal ventilation rate) for a given design condition (e.g., representative or critical weather conditions and, possibly, pollutant source emission rates).

4.2.1 Ventilation Design Tasks

As to process, five distinct design tasks can be identified:

- Establish Global Geometry The designer must initially set the global geometric configuration of the system – e.g., siting of the building and landscape configuration, overall building form, and positions of fresh air inlets and stale air exhausts.
- Establish System Topology The designer must layout the airflow paths from inlet to outlet that will achieve the desired airflow objective – e.g., cooling, moisture control, CO2 control, etc. – and select the types of airflow components – e.g., windows, doors, vents, etc. – that will provide the control of airflow desired.
- Component Sizing The designer must then size the components of the airflow system considering reasonable and relevant climatic conditions – the design conditions – and appropriate design criteria.
- Control Strategy The designer must develop a strategy to control the ventilation flow rates to achieve the objective design criteria and select hardware and, possibly, software to implement the strategy.

- Detail and Assembly Finally, the designer must develop detail and assembly drawings so that the system can actually be built.

This Technical Note addresses only Task 3 component sizing – the task often identified as "design" in the engineering community. Up-to-date guidance for Tasks 1 and 2 for both domestic and non-domestic buildings may be found in the recent publications of the Building Research Establishment (BRE) and the Chartered Institution of Building Services Engineers (CIBSE) (Stephen, Parkins et al. 1994; Irving and Uys 1997; CIBSE 1998).

In addition to these publications, Annex 27 of the International Energy Agency has recently published a useful handbook for ranking domestic ventilation systems – the *Simplified Tools for Evaluation of Domestic Ventilation Systems Handbook* – that offers guidance relating to system configuration, component sizing, and several related design issues (Månsson 1998). The semi-quantitative procedures offered in this handbook offer a natural complement to this Technical Note.

4.2.2 Component Sizing Methods

In the allied fields of building technology, component sizing methods are invariably based on the same theoretical principles used to simulate building behavior. Recognizing the time and financial constraints designers face, however, these methods are generally simplified inversions of the mathematical methods used for whole system simulation. Three classes of system simulation are relevant here:

- simple (uncoupled) airflow analysis where ventilation airflows are predicted for selected climatic conditions given temperature distributions within the building,
- airflow integrated with contaminant dispersal analysis where building air quality distributions and airflows are predicted for selected climatic conditions,
- coupled airflow/thermal analysis integrated with contaminant dispersal analysis where building air quality and temperature distributions and airflows are predicted for selected climatic conditions.

The first class of simulation, when inverted would lead to design methods based on simple ventilation criteria. The second class, on the other hand, would lead to design methods based on air quality criteria, while the third would lead to design methods using both air quality and thermal comfort criteria. For the purposes of the discussion here, these design methods will be classified as:

- *first order design methods* where ventilation system components are sized for ventilation criteria (i.e., to achieve a specified ventilation rate),
- *second order design methods* where ventilation system components are sized for air quality criteria (i.e., to maintain pollutant levels below some threshold level), and
- *third order design methods* where ventilation system components are sized for air quality and thermal comfort criteria.

As these simulation classes are ordered in terms of both theoretical and computational complexity, design methods based on them will reflect this complexity. It will be demonstrated that *first order* design methods based on ventilation criteria alone are quite tractable and may be approached using existing, relatively simple and intuitively direct theory. When air quality is directly related to ventilation rate (e.g., for cases where steady-state dilution dynamics govern), then *second order* design methods based on air quality criteria will be no more difficult than the first order methods. For dynamically cases, *second order* design methods may have to be based on trial and error iteration using the parent simulation method. While this is more complex, it has been done in recent research investigations (Wray 1993; Bassett 1994; Bassett 1994; Persily 1998).

As the third class of simulation demands consideration of the complex dynamically coupled interaction of a building's airflow system and its thermal characteristics – a challenge that only the most advanced simulation programs have been able to address and one where few, if any, can claim real expertise – *third order* design methods based on dual thermal comfort and air quality criteria have yet to be addressed. In dwellings, however, indoor air temperatures are controlled during the heating and cooling seasons, effectively uncoupling the thermal response of the building from the airflow and contaminant dispersal response. Thus it may seem that the *third order* design challenge is not particularly relevant in dwelling design. Yet during the so-called *shoulder seasons,* homeowners are likely to let indoor air temperatures float freely and thus thermal and airflow behavior become coupled. It is at these very times when stack-driven flows may be expected to be small and thus under-ventilation may result. Thus, as ventilation system design methods mature we may expect the need for *third order* design to rise.

4.2.2.1 Existing First-Order Methods

A new, theoretically rigorous and general *first order* design method for sizing components of passive ventilation systems of arbitrary complexity is presented in the next chapter of this Technical Note. Before proceeding to this chapter, however, it is useful to briefly review the current state of the art of existing *first order* design methods.

Santamouris reviews six semi-empirical methods for sizing natural openings in (Allard 1998) – two methods developed by the Florida Solar Energy Center, a method proposed by Aynsley (Aynsley, Melbourne et al. 1977), the Simplified Method of the University of Athens, one promoted by ASHRAE (ASHRAE 1997), and the British Standards Institution (BSI) Method. As all of these methods are approximate methods that can be considered to be simplified special cases of the general method presented in the Chapter 5 hence their details will not be considered here.

Suffice it to say that all six methods employ a building idealization consisting of a single control volume with a single inlet and outlet with all internal resistances assumed negligible. Furthermore, in all of these methods the inlets and outlets are (tacitly) modeled as simple orifices and are thus only appropriate for sizing windows, doors, trickle vents and other openings where the cross-sectional dimension of the opening is large relative to the dimension along the airflow path. The first four methods are limited to wind-driven airflow. The ASHRAE method considers either wind or buoyancy-driven effects but offers no means to consider both combined. The BSI method offers an ad hoc approach to account for the combined effects of wind and buoyancy forces.

Santamouris suggests that network airflow analysis may be used in an iterative manner to size natural ventilation openings providing a more rigorous approach to the problem. Using the six methods listed above and network airflow analysis to both size components and simulate system performance, Santamouris compares the reliability of the methods in achieving a specified design ventilation flow rate (i.e., 6 ACH) for a reasonable design condition (i.e., a mean wind speed of 6 m/s) (Allard 1998). For the building considered, which due to the limitations of the models was simply a box with inlet and outlet openings, only two of the methods produced designs that achieved the design ventilation flow rate exactly. The four other, more empirical methods produced designs that exceeded the design ventilation flow rate by as much as 60%.

The *loop equation* design method presented in Chapter 5 may be applied to the design problem considered by Santamouris with no more effort than that required by any of these six methods and quite directly provide a design solution that is "exact" (i.e., relative to a network airflow simulation evaluation). Furthermore, it will be shown that this method a) is not limited to orifice-type component sizing, b) can directly and rigorously account for the combined effects of wind and buoyancy forces, c) can be applied to problems of sizing an arbitrary assembly of components that make up the passive ventilation system, and d) provides the means to consider annual variations of design conditions.

Awbi provides a component relation to size inlet openings, stack duct, and stack duct fittings of the form of the orifice equation, Equation 4.11, noting the discharge coefficient C_d may be set equal to published loss coefficients for fittings (i.e., equivalent to Equation 4.25) (Awbi 1991). He notes that stack and wind pressures may be estimated using equations equivalent to Equations 4.2, 4.3, and 4.4 and implies that duct pressure losses may be modeled by the classic duct loss equation, Equation 4.21. However, he does not explicitly outline how these relations may be combined to form design equations or how these nonlinear equations may be solved to size system components. Again, the elements of Awbi's approach are contained in the method presented in Chapter 5. Consequently, Awbi's approach may be considered a somewhat incomplete special case of the more general method to be presented in the next chapter.

Etheridge and, independently, Andersen have taken a different approach to the problem of sizing openings for natural ventilation systems – they offer solutions to ‘‘exact’’ formulations of the component sizing problem (Etheridge and Riffat 1997; Andersen 1998). Again the same fundamental relations presented above for wind-induced pressures, buoyancy-induced pressures, and flow through orifices, ducts, and duct fittings were employed. Etheridge applied these relations to form system equations for representative building idealizations and then solved the resulting equations to develop non-dimensional graphs for rapid manual calculations of component sizes. Andersen, on the other hand, provides an iterative analytical procedure to solve the system equations to determine component sizes but limits consideration to buoyancy-driven airflows. Both limited considerations to building idealizations having negligible internal resistances but with multiple openings distributed over the height of multiple story buildings. Etheridge also included a passive stack in some of his studies.

The generation of design graphs to allow rapid estimation of component sizes is, of course, the strong point of Etheridge's approach. The solution of the nonlinear equations that are associated with natural ventilation component sizing may be expected to be challenging at best for most analysts. The design graphs avoid this difficulty but, of course, so will a well-designed computational tool. On the other hand, limiting designs to one of a few selected building idealizations and ignoring internal resistances is a major shortcoming of these ‘‘exact’’ analytical approaches as it is for the semi-empirical approaches discussed above. To be practically useful, a design method must provide the means for a designer to size or account for all components of an arbitrarily configured natural ventilation system including internal resistance components.

5 A First-Order Design Method Based on *Loop Equations*

Without a theoretically rigorous and general method of sizing passive ventilation system components, researchers limited their attention to one system, the PSV system, and applied empirical trial and error procedures to improve its performance over nearly a century of development. As useful as recent research into the behavior of these systems has been, it too remains empirically-based, differing from the past only in that computational simulation is now sometimes used to conduct experiments (Bassett 1994; Heikkinen and Pallari 1994; Villenave, Millet et al. 1994; Harris and Webb 1996; Månsson 1998; Persily 1998). The development of such a design method is key to not only the successful application and adaptation of the PSV system to the North American context but to the development of alternative passive ventilation systems here and abroad.

This section will present an ‘‘exact’’ approach to the *first-order* design problem defined above. It may be considered to be a more complete formulation of the approximate approach recently published in the CIBSE Application Manual: *Natural Ventilation in Non-domestic Buildings* AM10:1997 (Irving and Uys 1997). The approach presented is based on so-called *loop equations* that are commonly used in flow network simulation in the hydraulics field (Jeppson 1976) but have been largely ignored in the building ventilation field (with the notable exception of the work by Li (Li 1993) and Wray (Wray and Yuill 1993)). It allows direct sizing of a variety of airflow components and the direct and unambiguous consideration of both stack-driven and wind-driven flows without resorting to simplifying approximations. Yet, the approach is developed to allow building designers to identify a full range of feasible design configurations so that other, non-technical design constraints and operational strategies may be included in the process of seeking a design solution. Several examples of passive ventilation design scenarios will be presented.

Irving and his colleagues have developed an approach to the problem of sizing components of natural ventilation systems in buildings (Irving and Concannon 1995; Irving, Concannon et al. 1995; Irving and Uys 1997). This approach is coincidentally similar to an approach presented by the author to architectural design students at MIT and Yale based on the fundamental principles of the macroscopic airflow analysis theory reviewed above. Irving's approach, however, has been formulated as simplified *loop equations* for the macroscopic idealizations considered. For the professional building designer, there is little penalty and much to be gained through the use of the more general and complete method based on an exact formulation of *loop equations*.

The basic theory underlying this approach and the method of its application to the design of natural ventilation systems will be presented in this chapter. A series of example applications of increasing complexity will elaborate the details of the method and demonstrate the means to account for infiltration and annual variations of climatic conditions. Conventional PSV systems, innovative PSV systems utilizing self-regulating vents, and alternative passive ventilation systems will be considered.

5.1 Basic Theory

The theory is formulated using macroscopic idealizations of building ventilation systems. That is to say, the building system is idealized as a collection of control volumes (e.g., rooms, zones, or duct segments of a duct network system) linked by airflow paths or *components* (e.g., windows, doors, cracks, or duct segments of a duct network). For example, in the representative macroscopic idealization illustrated in Figure 5.1, a single room is modeled with an inlet opening at *a*, a room outlet grill at *b*, and a ducted stack running from *b* to *c* terminated by a stack terminal device at *c*. The control volumes considered in this case are the room and the stack duct, and the flow components: the inlet, outlet, and stack terminal devices.

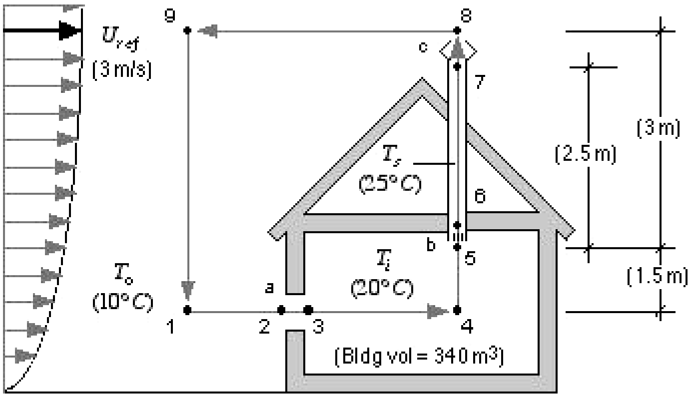


Figure 5.1 Representative macroscopic idealization of a building. (Numerical values in parenthesis will be used in a subsequent example application.)

Discrete locations or nodes are identified (e.g., the black dots in Figure 5.1) with which values of temperature and pressure are specifically defined. The form of the variation of temperature and pressure within each control volume is assumed and directly related to the node values – e.g., most commonly, but not necessarily, the temperature is assumed to be uniform and the pressure assumed to vary hydrostatically within the control volume. Finally, one of several pressure-flow relations is then associated with each flow component to complete the building idealization task.

With a macroscopic idealization in hand, one may approach the task of forming system equations by summing expressions describing the changes of pressure as one traverses a continuous flow *loop* through the building. A *loop* follows possible airflow paths from node to node in the building idealization returning to the original starting node. With reference to Figure 5.1, one such loop is possible following the node path 1-2-3-4-5-6-7-8-9 and back to 1. These changes of pressure must add up to zero upon completion of the loop – a condition that essentially satisfies the fundamental principle of the conservation of momentum. While these *loop equations* are tricky to form automatically (i.e., for simulation analysis), they are well-suited to design development since the designer must, literally, layout the loops as a first step in the design process.

5.1.1 Wind & Hydrostatic Pressures

Wind and buoyancy effects due to hydrostatic changes, of course, play a central role in the formulation of the loop equations. Wind-induced pressure changes may be modeled using Equation 4.2 presented above that relates wind-induced surface pressures to a reference wind speed U_{ref} and surface wind pressure coefficients C_p. Wind pressure coefficient data for buildings may be found in the ASHRAE Handbook of Fundamentals (ASHRAE 1997) and in the compilation of data assembled by the AIVC (Orme, Liddament et al. 1998). While these references also provide some guidance for stack terminal devices more detailed information can be found in the TNO and BRE publications discussed earlier (de Gids and den Ouden 1974; Welsh 1995; Welsh 1995).

Buoyancy effects may be modeled using the familiar hydrostatic equations presented above for uniform, Equation 4.3, or nonuniform, Equation 4.4, temperature distributions within each control volume and the exterior environment as well. For most design situations the ideal gas law for dry air, Equation 4.5, should be sufficiently accurate for estimating air densities. Loop Flow Component Relations

Flow component relations used in loop equations relate the pressure change ΔP_l across a flow component *"l"* to the volumetric airflow rate $\dot{V}_l$ through the component and a characteristic design variable associated with the component ϕ_l as – in general, functional notation:

$$\Delta P_l = g(\dot{V}_l, \phi_l) \tag{5.1}$$

These relations are simply inverse forms of the component flow relations used in the more conventional macroscopic analytical theory (outlined above).

We may directly create a *library* of loop component relations from the macroscopic flow component relations presented earlier (i.e., Equations 4.8, 4.9, 4.11, 4.17, 4.18 and 4.21 or 4.24) to obtain the following specific flow components:

$$\textit{Power Law Component}\ \Delta P_l = \frac{\dot{V}_l^{1/n_l}}{C_l^{1/n_l}} \ \textit{where}\ \phi_l = C_l^{1/n_l} \tag{5.2}$$

$$n_l \approx 0.6 \pm 0.1 \ \text{ for cracks \& joints } \ \text{(Orme, Liddament et al. 1998)}$$

$$n_l \approx 0.63 - 0.13\log_{10} C_l \left(\frac{\mathrm{dm^3}}{\mathrm{s{\cdot}m^2{\cdot}Pa^{-n}}}\right) \ \text{ for porous surfaces } \ \text{(Orme, Liddament et al. 1998)}$$

$$\textit{Effective Leakage Area Component}\ \Delta P_l = \frac{\rho \dot{V}_l^2}{2\left(A_{eff-l}\right)^2} \ \textit{where}\ \phi_l = A_{eff-l}{}^2 \tag{5.3}$$

$$\textit{Orifice Component} \quad \Delta P_l = \frac{\rho \dot{V}_l^2}{2 C_d{}^2 A_l{}^2} \ \textit{where}\ \phi_l \equiv A_l{}^2 \tag{5.4}$$

$$\textit{Quadratic Component} \qquad \Delta P_l = \alpha \dot{V}_l + \beta \dot{V}_l^2 \tag{5.5}$$

$$\textit{Self-Regulating Vent Component}\ \Delta P_l = -\Delta P_o \ln\left(1 - \frac{\dot{V}_l}{\dot{V}_o}\right) \textit{where}\ \phi_l \equiv \dot{V}_o \tag{5.6}$$

$$\textit{Duct Component: General Form}\ \Delta P_l = \frac{f\,L}{2}\frac{\rho\,\dot{V}_l^2}{D_h\,A_l{}^2} \ \textit{where}\ \phi_l \equiv D_h\,A_l{}^2 \tag{5.7}$$

$$\textit{Duct Component: Laminar Flow}\ \Delta P_l = \frac{8\mu L\dot{V}_l}{D_h^{\ 2} A_l} \quad \textit{where} \quad \phi_l \equiv D_h^{\ 2} A_l \tag{5.8}$$

$$\textit{Duct Component: Turbulent Flow}\ \Delta P_l = 0.04\frac{\rho^{3/4}\mu^{1/4}\dot{V}_l^{7/4}}{D_h^{5/4} A_l^{7/4}} \quad \textit{where} \quad \phi_l \equiv D_h^{5/4} A_l^{7/4} \tag{5.9}$$

$$\textit{Duct Fitting Component}\ \Delta P_l = \frac{\rho C_l \dot{V}_l^2}{2A_l^2}\ \textit{where}\ \phi_l = A_l^2 \tag{5.10}$$

To these component relations a fan component relation may be added using available theory. Measured pressure-flow data for fans – so-called *fan performance curves* – may be approximated by polynomial expressions of the following form:

$$\textit{Base Fan Performance Curve}\ \Delta P_l = a_0 + a_1\dot{V}_l + a_2\dot{V}_l^{\ 2} + a_3\dot{V}_l^{\ 3} + \ldots \tag{5.11}$$

where the leading coefficients, a_0, a_1, a_2, a_3, …, are empirically determined constants.

Fan laws provide the means to estimate fan performance of geometrical similar fans (ASHRAE 1997). Given a base fan performance curve for a fan of size D_o operating at a rotational speed of N_o, a geometrically similar fan of size D operating at rotational speed of N would have a derived fan performance curve of:

Derived Fan Performance Curve

$$\frac{\Delta P_l}{\left(\frac{D}{D_o}\right)^2\left(\frac{N}{N_o}\right)^2} = a_0 + a_1\frac{\dot{V}_l}{\left(\frac{D}{D_o}\right)^3\left(\frac{N}{N_o}\right)} + a_2\frac{\dot{V}_l^{\ 2}}{\left(\frac{D}{D_o}\right)^6\left(\frac{N}{N_o}\right)^2} + a_3\frac{\dot{V}_l^{\ 3}}{\left(\frac{D}{D_o}\right)^9\left(\frac{N}{N_o}\right)^3} + \ldots \tag{5.12}$$

Limiting consideration to the quadratic form for convenience, a fan component relation for use in pressure loop equations may be formulated from Equation 5.12 as:

$$\textit{Fan Component}\ \Delta P_l = a_0\left(\frac{D}{D_o}\right)^2\left(\frac{N}{N_o}\right)^2 + a_1\frac{\dot{V}_l}{\left(\frac{D}{D_o}\right)}\left(\frac{N}{N_o}\right) + a_2\frac{\dot{V}_l^{\ 2}}{\left(\frac{D}{D_o}\right)^4} \tag{5.13}$$

Here the design variables may be taken as either the base fan performance curve coefficients, a_0, a_1, a_2, a_3, …, the rotational speed ratio, N/N_o, or the size ratio, D/D_o, depending on the context of the design problem.

To provide an example of the application of Equation 5.13, a base fan performance curve (i.e., for $N/N_o = 1.0$ and $D/D_o = 1.0$) representative of a relatively high capacity fan that might be used in a residential centralized mechanical exhaust system is plotted in Figure 5.2. Derived fan performance curves, based on Equation 5.13, for a geometrically similar fan operated at 75% of the base rotational speed and another similar fan 75% of the size of the base fan are also plotted on this figure.

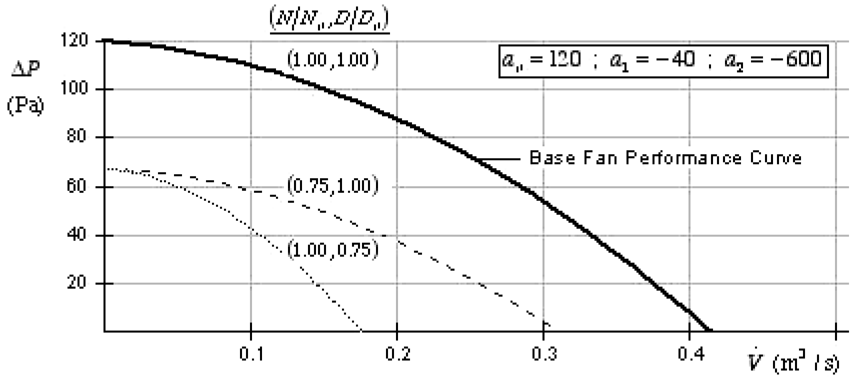


Figure 5.2 Base fan performance curve and derived fan performance curves based on Equation 5.13 representative of a high capacity fan for a central residential exhaust system.

The general fan-component relation, Equation 5.13, made specific using a particular base fan performance curve may be included in pressure loop equations as required. Suffice to say here, the multiple terms of the polynomial relation will, however, introduce complexities not encountered with the single term relations of the other flow components.

5.1.2 Boussinesq Approximation

The subscript on the air density variable ρ in the flow component relations and the wind pressure relation, Equation 4.2, has been omitted to acknowledge that the uncertainty associated with the use of these equations does not warrant precision in the specification of air density. When applying these relations sufficient accuracy, especially for design calculations, may be obtained by using an air density representative of the range of values associated with the problem at hand (e.g., a mean value). On the other hand, precision is required in the hydrostatic equations, Equations 4.3 and 4.4, thus the subscript is used. When applying the hydrostatic equations air densities should be accurate to three or four significant figures – the ideal gas law for dry air, Equation 4.5, may be used to estimate these air densities for most practical situations. This approximation – a discrete form of the Boussinesq assumption used in *microscopic* analysis – is not theoretically required but simply eases the burden of computation.

5.1.3 Pressure Loop Equations

Using the flow component relations enumerated above and the hydrostatic and wind pressure relations presented earlier, we may directly form the pressure loop equations for a given building idealization. For the representative idealization shown in Figure 5.1, we begin at the ambient pressure node 1, P_{1o}, and move forward around the loop adding first the increase due to the wind acting on the wall at node 2, then the pressure drop ΔP_a through the window component a, the hydrostatic decrease resulting from the elevation change from 4 to 5, and so on to obtain:

$$+C_{P2}\frac{\rho U_{ref}{}^{2}}{2} - \overbrace{\Delta P_a}^{\text{window}} - \rho_i g \Delta z_{45} - \overbrace{\Delta P_b}^{\text{room outlet}} - \overbrace{\Delta P_{bc}}^{\text{stack duct}} - \rho_s g \Delta z_{58} - \overbrace{\Delta P_c}^{\text{stack terminal}} - C_{P8}\frac{\rho U_{ref}{}^{2}}{2} + \rho_o g \Delta z_{91} = 0$$

(5.14)

where ρ_i and ρ_s are the air densities in the room and the stack respectively. Substituting the flow relations for each of the components yields the final explicit form of the loop equation. For a more complex building additional flow paths and, hence, loop equations may be formed.

Presented as in Equation 5.14, the loop equations may seem rather formidable. On closer examination, it may be seen that these equations involve simply:

- a summation of hydrostatic changes that define a stack-driven pressure difference ΔP_s,
- windward and leeward pressures that define the wind-driven pressure difference ΔP_w, and
- a summation of flow component pressure drops ΔP_l along the loop.

Thus the loop equations assume the simple general form:

$$\textit{Loop Equation General Form} \sum \overbrace{f(\dot{V}_l)}^{\Delta P_l} = \overbrace{g\sum \rho_{ij}\Delta z_{ij}}^{\Delta P_s} + \overbrace{(\Delta C_P)\frac{\rho U_{ref}{}^{2}}{2}}^{\Delta P_w} \quad (5.15)$$

A consistent set of sign conventions must be applied in the formulation of this equation:

- $f(\dot{V}_l)$ is positive for *forward* flow along the loop (although the fan component terms must be added algebraically),
- Δz_{ij} is positive for drops and negative for rises in elevation along the loop,
- ρ_{ij} is the air density corresponding to the elevation change Δz_{ij}, and
- ΔC_p is the algebraic sum of pressure coefficients with C_P summed positively when traversing the loop from the exterior to a wall surface location and negatively other wise.

For loops involving orifice flow components only, the loop equations assumes the following form:

$$\textit{Orifice-Only Loop Equation} \sum \overbrace{\frac{\rho \dot{V}_l^{2}}{2C_d{}^{2}A_l^{2}}}^{\Delta P_l} = \overbrace{g\sum \rho_i \Delta z_{ij}}^{\Delta P_s} + \overbrace{(\Delta C_P)\frac{\rho U_{ref}{}^{2}}{2}}^{\Delta P_w} \quad (5.16)$$

This orifice-only relation is an exact form of the simplified and approximate expression developed by Irving and his colleagues by limiting building idealizations to control volumes with single inlets and outlets and accounting for internal resistances only approximately (Irving and Concannon 1995; Irving, Concannon et al. 1995; Irving and Uys 1997).

At any given stage of design all variables in a loop equation will be known, with the exception, of course, of the flow component design parameters (e.g., the opening areas A_l for orifice components), and may be substituted directly. In general, however, one will not be able to define a sufficient number of loop equations to determine a unique solution to the design problem. This dilemma – i.e., that many alternative *feasible design solutions* are possible in natural ventilation design – provides the designer with the opportunity to include additional and often non-technical constraints in the process of selecting a design solution.

5.1.4 Design Example 1: Sizing Components of a Traditional PSV System

As a first example, consider the problem of sizing the components of the PSV system for the single-zone building represented in Figure 5.1.

5.1.4.1 Design Conditions

We will assume a reference approach wind speed of U_{ref} = 3 m/s, a windward pressure coefficient of C_{P2} = +0.7, a stack terminal pressure coefficient of C_{P8} = –0.5, and an outdoor air temperature of T_o = 10 °C (50 °F), an indoor air temperature of T_i = 20 °C (68°F) and a stack temperature of T_s = 25 °C (77 °F). Applying the ideal gas law for dry air, Equation 4.5, the air densities for these three temperatures are ρ_o = 1.245, ρ_i = 1.203, and ρ_s = 1.183 kg/m^3 respectively.

5.1.4.2 Design Criteria

To provide the ASHRAE recommended ventilation rate of 0.35 ACH, for a representative dwelling of 140 m^2 (1,500 ft^2) floor area and an enclosed volume of 340 m^3 (12,000 ft^3), the air flow rate through the PSV system would have to be 119 m^3/h or 0.033 m^3/s. This ventilation airflow rate establishes the objective criteria that the passive ventilation system must satisfy.

5.1.4.3 Loop Equation

To apply the general loop equation, Equation 5.15, component models need to be selected for each of the components of the ventilation system. In general, this will require judgement based on experience and familiarity with the fluid dynamics underlying each component model relation. In this case, however, the selection choices are relatively obvious – the window may be modeled using the orifice model Equation 5.4; the room outlet and stack terminal fittings may be modeled with the duct fitting model Equation 5.10; and the stack duct may be modeled with the duct component model Equation 5.7. Formally, then, the loop equation specific to the building of Figure 5.1 becomes:

$$\overbrace{\frac{\rho \dot{V}_a^2}{2C_d^2 A_a^2}}^{\text{window}} + \overbrace{\frac{\rho C_b \dot{V}_b^2}{2A_b^2}}^{\text{room outlet}} + \overbrace{\frac{f L_s \rho \dot{V}_s^2}{2D_{hs} A_s^2}}^{\text{stack duct}} + \overbrace{\frac{\rho C_c \dot{V}_c^2}{2A_c^2}}^{\text{stack terminal}} = \overbrace{g\left(-\rho_i \Delta z_{4,5} - \rho_s \Delta z_{5,8} + \rho_o \Delta z_{9,1}\right)}^{\Delta P_S} + \overbrace{\left(C_{P2} - C_{P8}\right)\frac{\rho U_{ref}}{2}}^{\Delta P_W} \tag{5.17}$$

where, the design parameters are A_a, A_b, A_s, and A_c, the unobstructed cross-sectional areas of the window, room outlet, stack duct, and stack terminal respectively, and D_{hs}, the hydraulic diameter of the stack duct.

To transform this loop equation for sizing members numerical values must be substituted for all parameters except, of course, the component design parameters, including:

- ρ , the nominal air density will be taken as the average of design condition values or ρ = 1.21 kg/m^3,
- $g\Delta z$, the elevation differences are shown in Figure 5.1; g = 9.8 m/s^2,
- C_d, the orifice discharge coefficient will be assumed to be 0.60 – a value associated with transitional or turbulent flow in sharp-edged openings,
- C_b, the room outlet fitting loss coefficient will be assumed to be 0.10 – a value characteristic of flush-fitting bell-mouth entries to circular ducts (ASHRAE 1997),
- *fL*, the length of the duct *L* is shown in Figure 5.1; *f* will be assumed to be 0.03 – a value associated with transitional-to-turbulent flow in ducts,
- C_c, the stack terminal device fitting loss coefficient will be assumed to be 1.0 – a value associated with transitional-to-turbulent flow from a simple open circular duct (Welsh 1994; Welsh 1995; Welsh 1995; Welsh 1995; ASHRAE 1997),

The component volumetric airflow rates must satisfy continuity. Here, the airflow rate through the window $\dot{V}_a$, room outlet $\dot{V}_b$, stack $\dot{V}_s$, and stack terminal device $\dot{V}_c$ must all equal 0.033 m3/s (i.e., within the limits of the Boussinesq approximation discussed above). Hence, substituting numerical values for all variables except the design parameters:

$$\frac{\overbrace{0.00183}^{\text{window}}}{A_a^2}+\frac{\overbrace{0.0000659}^{\text{room outlet}}}{A_b^2}+\frac{\overbrace{0.0000961}^{\text{stack duct}}}{D_s^5}+\frac{\overbrace{0.000659}^{\text{stack terminal}}}{A_c^2}=\overbrace{2.44\text{ Pa}}^{\Delta P_S}+\overbrace{6.53\text{ Pa}}^{\Delta P_W} \qquad (5.18)$$

where, for a circular duct cross-section $A_s = \pi D_s^2/4$ and $D_{hs} = D_s$, the duct diameter.

It is important to emphasize that driving pressures due to wind and buoyancy effects are accounted for directly and unambiguously in this formulation. They may be considered acting together or separately (i.e., to size components for the with-wind and without-wind conditions). Nevertheless, the resulting single equation is not only algebraically nonlinear but contains, in this case, four unknown design parameters. This underscores the fact that natural ventilation design is usually an under-determined problem – many feasible solutions to the design problem may be formulated.

5.1.4.4 Feasible Design Surfaces & Asymptotes

Loop equations, in general, and Equation 5.18, in particular, define *feasible design surfaces* in terms of the design parameters (i.e., here, the denominators of the left hand side of Equation 5.18). Fortuitously, these surfaces have a useful form. They are hyperbolic with limiting asymptotes corresponding to minimum values for each of the design parameters that will still achieve the ventilation airflow objective. Consequently, these limiting asymptotes are directly useful to the designer – they establish the minimum feasible sizes of each of the flow components along the loop.

Quantitative values for each of the limiting asymptotes may be determined quite readily by considering the limiting cases where all other design parameters are allowed to approach large values (i.e., all other flow components are assumed to have negligible resistance). Thus for example, the limiting asymptote for the window inlet vent in this first example would be determined as:

$$\lim_{\substack{A_a\to\infty \\ D_s\to\infty \\ A_c\to\infty}}\left(\overbrace{\frac{0.00183}{A_a^{\;2}}}^{\text{window}}+\overbrace{\frac{0.0000659}{A_b^{\;2}}}^{\text{room outlet}}+\overbrace{\frac{0.0000961}{D_s^{\;5}}}^{\text{stack duct}}+\overbrace{\frac{0.000659}{A_c^{\;2}}}^{\text{stack terminal}}\right)=\overbrace{\frac{0.00183}{A_a^{\;2}}}^{\text{window}}=\overbrace{2.44\text{ Pa}}^{\Delta P_S}+\overbrace{6.53\text{ Pa}}^{\Delta P_W}$$

(5.19a)

or

$$A_a \geq \begin{Bmatrix} 0.027\ \text{m}^2\ \text{ without-wind} \\ 0.014\ \text{m}^2\ \text{ with-wind} \end{Bmatrix} \tag{5.19b}$$

That is to say, to achieve the airflow objective of 0.35 ACH, the window opening area would have to be greater than 0.027 m^2 (27,000 mm^2) for the without-wind case and 0.014 m^2 (14,000 mm^2) for the with-wind case. The other limiting asymptotes may be determined in a similar manner. For the room outlet A_b • 0.0052 m^2 for the without-wind case and A_b • 0.0027 m^2 for the with-wind case. For the stack terminal A_c • 0.016 m^2 for the without-wind case and A_c• 0.0086 m^2 for the with-wind case. The values for the stack duct are D_s • 0.132 m for the without-wind case and D_s • 0.101 m for the with-wind case.

5.1.4.5 Additional Constraints & Design Rules

To narrow the possibilities the designer may introduce additional constraints. Here it is reasonable to assume the room outlet and stack terminal fittings have the same nominal cross-sectional area that the duct has. In fact, for these specific fittings, the loss coefficients are defined in terms of the duct cross-sectional area.

Additional Constraint $\qquad A_b = A_c = \pi D_s / 4$ (5.20)

Substituting this constraint into the governing loop equation, Equation 5.18, establishes a more-narrowly defined set of feasible solutions – now defined in terms of single curves in the "A_a–D_s space" as:

$$\overbrace{\frac{0.00183}{A_a^2}}^{\text{window}}+\overbrace{\frac{0.00117}{D_s^4}}^{\substack{\text{room outlet}\\ \text{stack terminal}}}+\overbrace{\frac{0.0000961}{D_s^{\;5}}}^{\text{stack duct}}=\overbrace{2.44\text{ Pa}}^{\Delta P_S}+\overbrace{6.53\text{ Pa}}^{\Delta P_W} \qquad (43)(5.21)$$

Again, we can directly determine the numerical values of the limiting asymptotes for this transformed loop equation. In this case, however, it is more compelling to plot the resulting curves for the with-wind and without-wind cases as shown in Figure 5.3.

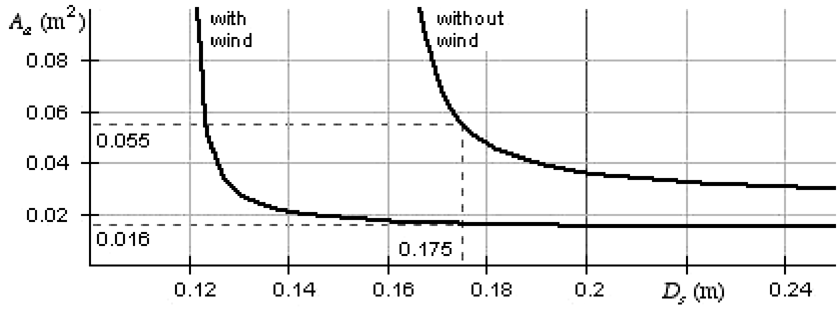


Figure 5.3 Feasible design curves for sizing window and duct diameter to achieve 0.35 ACH for the example 140 m² dwelling considered.

Using these design curves, we may finally specify candidate design solutions.
Most often designers choose to select off-the-shelf components that, in effect, establish technically arbitrary but practically necessary *design rules.* Here we will assume duct diameters are available in 25 mm increments off-the-shelf. For example, if we choose to specify a stack duct diameter of 0.175 m (175 mm) then the window free opening area for the critical without-wind condition would be 0.055 m^2 (55,000 mm^2) – i.e., as indicated in Figure 5.3. To maintain the desired 0.35 ACH these openings would have to be constricted to 0.016 m^2 (16,000 mm^2) for the with-wind case – also indicated in Figure 5.3.

5.1.4.6 Operational Strategies

The solution obtained above implicitly defines an *operational strategy*. During still conditions, the window opening would be set to 0.055 m^2. For the moderate breezes (i.e., corresponding to the design wind speed used of 3 m/s) it would be constricted to 0.016 m^2. Alternatively, we could reduce the stack terminal opening with a damper to limit airflow while leaving the window opening at 0.055 m^2. In this case, the loss coefficient for the stack terminal C_c would become a design variable and the window opening area would be fixed at 0.055 m^2. A reformulated loop equation could then be used to determine the required loss coefficients (i.e., damper settings) for the with- and without-wind conditions. If this is done, we will discover C_c will have to equal 1.00 for the without-wind case and 6.73 for the with-wind case. The former value corresponds to a wide-open damper and the latter to a damper set at about 36° to the axis of the duct (ASHRAE 1997).

In general, by considering the probable range of design conditions, the designer may establish not only critical sizes of system components, but operational strategies and settings for a variety of environmental conditions.

5.1.4.7 Conclusion to Design Example 1

We will consider the impact of each of these two operational strategies on the impact of infiltration in a third design example considered subsequently. Here, it is interesting to compare the results of this design exercise to the BRE *best practice* recommendations. For a dwelling of the size considered, one might expect to find two service rooms (i.e., a bathroom and kitchen) and four habitable rooms (e.g., two bedrooms, a living room, and a dining room). Following BRE guidelines for PSV systems, such a dwelling would have room inlet vents totaling 2(4,000 mm^2) + 4(8,000 mm^2) = 40,000 mm^2, a kitchen stack of 125 mm and a bathroom stack of 100 mm. Consequently, the values determined above – a single stack of 175 mm with an inlet opening of 55,000 mm^2 for still conditions and 16,000 mm^2 for windy conditions – correspond quite favorably to the BRE guidelines.

5.1.5 Outline of the Loop Equation Design Method

Generalizing from this first example, a multi-step design methodology – the *Loop Equation Design* Method– may be outlined:

- Layout the global geometry and topology (i.e., flow component types and connectivity) of the passive ventilation flow loops for each habitable room in the house.
- Identify an ambient pressure node and additional pressure nodes at entries and exits of each flow component along each loop.
- Establish design conditions: envelope node wind pressure coefficients, ambient temperature, wind speed and direction, interior temperature conditions, and evaluate ambient and interior air densities (e.g., using Equation 4.5).
- Establish the design criteria, here, a ventilation objective (e.g., 0.35 ACH) and by applying continuity determine the objective design flow rates required for each flow component.
- Form the forward loop equations for each loop selected in i) above by systematically accounting for all pressure changes while traversing the loop. (It is useful here to keep the stack and wind pressure contributions separate so that the with-wind and without-wind cases can be more easily evaluated.)
- Determine the minimum feasible sizes for each of the flow components by evaluating the asymptotic limits of the loop equation (e.g., Equation 5.18) for the with-wind and without-wind cases separately.
- Develop and apply a sufficient number of technical or non-technical design rules or constraints to transform the under-determined design problem defined by each loop equation into a determined problem.
- Develop an appropriate operational strategy to accommodate the regulation of the passive ventilation system for variations in design conditions.

In the example considered above, the constraining relation between the stack diameter and the room outlet and stack terminal cross-sections established two of the three needed equations. The practical specification of a 175 mm stack established the third. These two constraints and one design rule combined with the loop equation provided the four equations needed to uniquely determine the four unknown component sizes. The operational strategy used involved regulating the window inlet vent opening area in response to wind pressures.

Defining design rules, design constraints, and operational strategies may seem to be the most elusive part of this methodology. It is important to emphasize that this is not a flaw in the methodology but intrinsic to the design of natural ventilation systems – systems that inevitably may be designed in a number of different ways. The methodology offers and supports the designer’s need to introduce practical design considerations (e.g., off-the shelf component sizes) that necessarily constrain design in the real world. In a more complex application, these design rules could define, for example, objective functions to be minimized in search of a design solution.

5.1.6 Design Example 2: Sizing Components of an Innovative PSV System

The first design example demonstrated the application of the Loop Method to the problem of sizing components of a traditional PSV system. The design of an innovative PSV system may be realized by simply replacing the inlet orifice flow component *a* with a self-regulating inlet vent component using Equation 5.6. Here, it will be assumed that the house is to be fitted with the Dutch self-regulating inlet vent, discussed above, that provides a threshold control pressure of $\Delta P_o = 0.30$ Pa. The pressure drop across this self-regulating device will then be:

$$\Delta P_a = -\Delta P_o \ln\left(1 - \frac{\dot{V}_a}{\dot{V}_o}\right) = -(0.3 \text{ Pa}) \ln\left(1 - \frac{0.033 \text{ m}^3/s}{\dot{V}_o}\right) \tag{5.22}$$

Recall that $\dot{V}_o$ is the objective design parameter for this component – the nominal air flow rate of the self-regulating inlet vent. Also, the air flow rate through *a* needed to achieve the ASHRAE recommended ventilation rate of 0.35 ACH is $\dot{V}_a = 0.033$ m³/s.

5.1.6.1 Loop Equation & Limiting Asymptotes

Replacing the window orifice term of Equation 5.18 with Equation 5.22 we obtain the governing loop equation for the passive stack system with a self-regulating inlet vent:

$$\overbrace{-(0.3)\ln\left(1 - \frac{0.033}{\dot{V}_o}\right)}^{\text{window self-regulating inlet}} + \overbrace{\frac{0.00117}{D_s^4}}^{\substack{\text{room outlet}\\ \text{stack terminal}}} + \overbrace{\frac{0.0000961}{D_s^5}}^{\text{stack duct}} = \overbrace{2.44 \text{ Pa}}^{\Delta P_S} + \overbrace{6.53 \text{ Pa}}^{\Delta P_W} \tag{5.23}$$

In evaluating the asymptotic limits of this equation, note that the limit of the self-regulating inlet term as $\dot{V}_o \to \infty$ is simply zero. Consequently, the limiting asymptotes of this loop equation may be determined by solving the following two equations for $\dot{V}_o$ and D_s, respectively:

$$\dot{V}_o \text{ Asymptote} \quad \overbrace{-(0.3)\ln\left(1 - \frac{0.033}{\dot{V}_o}\right)}^{\text{window self-regulating inlet}} = \overbrace{2.44 \text{ Pa}}^{\Delta P_S} + \overbrace{6.53 \text{ Pa}}^{\Delta P_W} \tag{5.24}$$

$$D_s \text{ Asymptote} \quad \overbrace{\frac{0.00117}{D_s^4}}^{\substack{\text{room outlet}\\ \text{stack terminal}}} + \overbrace{\frac{0.0000961}{D_s^5}}^{\text{stack duct}} = \overbrace{2.44 \text{ Pa}}^{\Delta P_S} + \overbrace{6.53 \text{ Pa}}^{\Delta P_W} \tag{5.25}$$

Solving, for the asymptotes associated with each of these equations, we find that the duct diameter must be larger than 0.164 m for the without-wind case and 0.122 m for the with-wind case while the nominal control flow rate of the self-regulating inlet device must exceed $\dot{V}_o = 0.03301 \text{ m}^3/\text{s}$ for the without wind case and $\dot{V}_o = 0.03300 \text{ m}^3/\text{s}$ for the with wind case. That is to say:

Passive Ventilation Design Strategy

The nominal controlled flow rate $\dot{V}_o$ of the self-regulating inlet vent must be slightly larger than the required ventilation rate.

This is, of course, one of the benefits of self-regulation – rough sizing of the self-regulating flow device is trivial. Final sizing of the device and sizing of the rest of the system components demands, however, the same effort as before.

It may be useful to note in passing here, that if the total driving pressure falls close to or below the self-regulating device threshold pressure ΔP_o, then rough sizing of self-regulating vents would not be as trivial. For example, if we arbitrarily set the stack pressure to, say, 0.25 Pa in Equation 5.23 and solve, we would discover that the limiting asymptote would be $\dot{V}_o = 0.0584$ m^3/s for the without-wind case. This value is substantially greater than the required ventilation rate of 0.033m^3/s. Hence we conclude:

Passive Ventilation Design Strategy

The threshold pressure of a self-regulating vent must fall well below <u>probable</u> driving pressures acting across the vent if regulation is to be achieved.

5.1.6.2 Non-Technical Constraints & Design Rule

Figure 5.4 presents plots of the loop equation, Equation 5.23, for the with-wind and without-wind cases. These curves define feasible combinations of stack duct diameter D_s and the nominal control flow rate $\dot{V}_o$ of the self-regulating vent that will achieve the ventilation airflow rate objective. The limiting asymptotes determined numerically above are also evident in these plots.

Using these feasible design curves we may now move in on a specific design solution by imposing additional constraints, design rules, and/or operational strategies. From these curves it is clear that if we select a stack duct diameter sufficiently large the nominal control flow rate of the self-regulating vent will be nearly equal for both the with-wind and without-wind cases. For example, we could select, as indicated in this figure, an "off-the-self" duct of D_s = 0.20 m and thereby determine the nominal volumetric flow rate of the self-regulating device would have to be 0.0333 m^3/s for the without-wind case. To maintain precise control of ventilation rate for the with-wind case, the self-regulating device would have to be throttled to 0.0330 m^3/s. That is to say, the theoretical adjustment needed is negligibly small thus, in use, the device would not need to be adjusted.

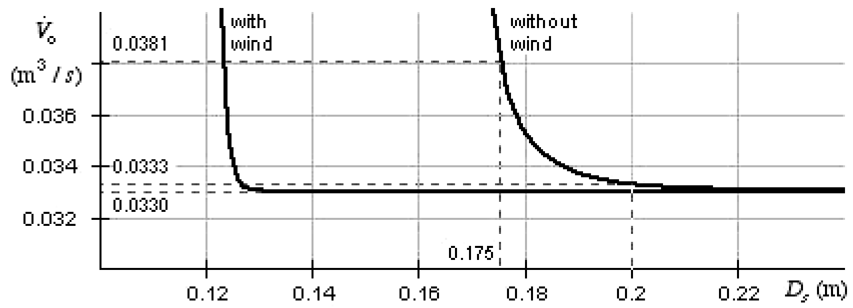


Figure 5.4 Feasible design curves for sizing stack diameter D_s and self-regulating inlet vent nominal control flow rate $\dot{V}_o$ to achieve 0.35 ACH for the example 140 m^2 dwelling considered. Dashed lines indicate appropriate nominal control flow rates $\dot{V}_o$ for a 0.175 and 0.200 m stack diameter.

5.1.6.3 Conclusion to Design Example 2

In conclusion, if we specify a 0.20 m diameter stack and a self-regulating inlet with $\dot{V}_o$ = 0.0333 m^3/s, then the system will provide the desired ventilation rate for the without-wind condition and only slightly more during windy conditions without adjustment. To evaluate the amount of over-ventilation that will occur, we may return to the loop equation, with all design parameters now set, and solve for the ventilation rate. If this exercise is carried out, we will discover the ventilation rate will simply approach the inlet control flow rate $\dot{V}_o$ = 0.0333 m^3/s (i.e., less than 1% over the desired flow rate) as we increase wind pressure. More precisely, as the pressure difference across the self-regulating vent exceeds its threshold control pressure the airflow rate will be limited to the control value.

On the other hand, if we had elected to use a 0.175 diameter stack duct, then the self-regulating inlet would have to be sized to $\dot{V}_o$ = 0.0381 m^3/s to achieve the desired 0.0330 m^3/s ventilation flow rate for the without-wind condition (see Figure 5.4). Under windy conditions the ventilation rate would again approach the control value, thus this design would suffer up to (0.0381 – 0.0330) ÷ 0.0330 = 15% over-ventilation if not adjusted for windy conditions. We conclude then:

> *Passive Ventilation Design Strategy*
>
> *To avoid adjustment of self-regulating vents and thereby realize their full benefit, other system components must be sized so that the self-regulating vent maintains authority over the ventilation flow rate. This may be achieved by reducing other system component resistances sufficiently so that practically the full driving wind and stack pressures act across the self-regulating vent.*

5.2 Accounting for Infiltration

In the first two design examples presented above the possibility of air infiltration has been ignored altogether, yet, if sufficient, infiltration can not only result in over-ventilation but can limit the ability of a passive ventilation system to control ventilation rates. Clearly, in the design of passive ventilation systems it would be best to avoid infiltration altogether, yet this is not practically possible.

One may account directly for infiltration using the macroscopic simulation techniques outlined above, but even this analytical approach is uncertain because the airtightness characteristics of any dwelling can never be known with great certainty. One may also account for infiltration using a loop equation approach, but here the procedure demands an iterative approach.

We begin by recognizing the "design solution" provided by the loop equation method is exact for the specified design flow rate through the loop. The design flow rate determines all pressure changes along the loop thus it may be used to directly estimate pressure differences across each element of the building envelope. Infiltration rates may then be estimated using these envelope pressure differences following an equivalent leakage area approach, Equation 4.9, or a power-law approach, Equation 4.8, for each surface of the building envelope.

Having estimated infiltration rates following this approach, three outcomes are possible:

- if the estimated infiltration flow rates are found to be negligibly small then the ventilation rate will be provided by the PSV system as desired,
- if the estimated infiltration flow rates are not negligibly small, yet together with the PSV flow rates satisfy continuity within a reasonable error of approximation, then the objective PSV system air flow rates will be known to be an admissible solution to the complete building airflow problem, or
- if the estimated flow rates are not negligibly small and together with the PSV flow rates do not satisfy continuity, then the objective PSV system air flow rates will be known to be an inadmissible solution to the complete building air flow problem.

The third, and most objectionable, possibility is likely to occur but, fortunately, this incorrect solution may be used to formulate a modified design solution that will satisfy continuity. In both case (ii) and the corrected case (iii) the total ventilation rate will exceed the desired value and the PSV system and/or the envelope airtightness will have to be modified accordingly to achieve the ventilation design objective.

These possibilities underscore the compelling fact that, in general, the success of a PSV system design depends critically on the control of infiltration.

5.2.1 Design Example 3: Accounting for Infiltration

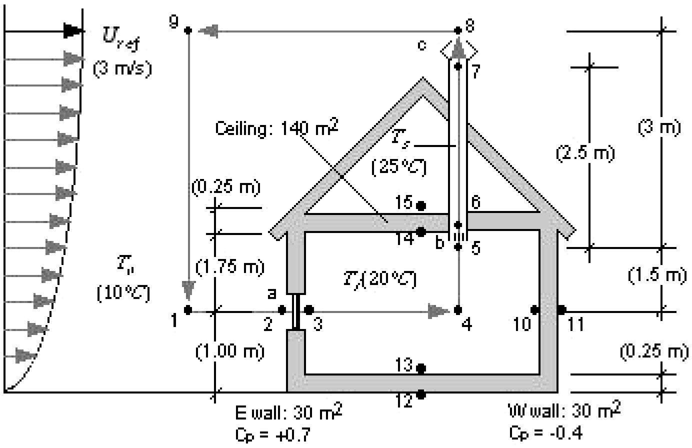


Figure 5.5 Passive stack and building (140 m²) configuration considered in Design Example 3.

To demonstrate this approach, consider again the 140 m² building studied in Design Example 1. For the purposes of this example, we will assume an east wall of 30 m² and a west wall of 30 m² are exposed to the wind with wind pressure coefficients of +0.7 and –0.4, respectively, as illustrated in Figure 5.5. Side-walls, floor and ceiling will be assumed to be protected from wind exposure (e.g., for a mid unit in a row-house with a fully vented attic space). Consequently, we will only need to account for stack driven flow through the ceiling and floors for these envelope surfaces. Finally, to carry out the analysis, additional nodes and vertical dimensions have been included to account for the pressure drop across each of the envelope surfaces.

For this building idealization, the envelope <u>infiltration</u> airflow rates may be estimated using the following effective leakage area relations:

E Wall $$\dot{V}_{Ewall} = A_{eff-Ewall}\sqrt{\frac{2\Delta P_{2-3}}{\rho}} \tag{5.26}$$

W Wall $$\dot{V}_{Wwall} = A_{eff-Wwall}\sqrt{\frac{2\Delta P_{11-10}}{\rho}} \tag{5.27}$$

Floor $$\dot{V}_{Floor} = A_{eff-Floor}\sqrt{\frac{2\Delta P_{12-13}}{\rho}} \tag{5.28}$$

Ceiling $$\dot{V}_{Ceiling} = A_{eff-Ceiling}\sqrt{\frac{2\Delta P_{15-14}}{\rho}} \tag{5.29}$$

Each of the pressure drops in these equations may be determined using the component flow relations, the ‘‘design solution’’ and the hydrostatic and the wind pressure equations. Two design solutions were considered in Design Example 1 – one in which the inlet vent was reduced to limit airflow during windy conditions and the second where the outlet was throttled with a damper to limit airflow. It will be useful to consider both.

Table 5.1 Alternative design solutions from Design Example 1 for windy conditions to achieve a passive stack system ventilation rate of 0.35 ACH.

	Window Inlet A_a (m^2)	Room Outlet Loss Coef. C_b	Stack Duct D_s (m)	Stack Terminal Loss Coef.[a] C_c	Pass. Vent. $\dot{V}$ (m^3/s)
Design I Inlet Throttled	0.016	0.10	0.175	1.00 (no damper)	0.033
Design II Outlet Throttled	0.055	0.10	0.175	6.73 (36° damper)	0.033

[a] Loss coefficients from (ASHRAE 1997).

For convenience, computational details are given below for Design I but only results for Design II (i.e., in square brackets).

5.2.1.1 Estimated Envelope Pressure Drops

For the east wall, the pressure drop is simply equal to the pressure drop in component *a*:

E Wall

$$\Delta P_{2-3} = \frac{\rho \dot{V}_a^{\,2}}{2C_d^{\,2} A_a^{\,2}} = \frac{\left(1.21 \text{ kg/m}^3\right)\left(0.033 \text{ m}^3/s\right)^2}{2\left(0.60\right)^2\left(0.016 \text{ m}^2\right)^2} = 7.15 \text{ Pa } \textit{for I} \; [0.60 \text{ Pa } \textit{for II}] \quad (5.30)$$

For the west wall, we note first that P_3, P_4, and P_{10} are equal and P_{11} differs from the ambient outdoor pressure P_1 due to the wind pressure action on the west wall. Systematically working from node 1 then:

W Wall

$$\Delta P_{11-10} = \left\{ P_1 + C_{P-Wwall}\left(\frac{\rho U_{ref}{}^2}{2}\right)\right\} - \left\{ P_1 + C_{P-Ewall}\left(\frac{\rho U_{ref}{}^2}{2}\right) - \Delta P_{2-3} \right\}$$
$$= \left(C_{P-Wwall} - C_{P-Ewall}\right)\left(\frac{\rho U_{ref}{}^2}{2}\right) + \Delta P_{2-3}$$
$$= \left(-0.4 - 0.7\right)\left(\frac{(1.21\ \text{kg/m}^3)\left(3\ \text{m/s}\right)^2}{2}\right) + 7.15\ \text{Pa} = 1.16\ \text{Pa}\ \textit{for I}\ [-5.38\ \text{Pa}\ \textit{for II}]$$

(5.31)

The pressure differences across the floor and ceiling may also be determined by systematically working from node 1 to 12, 13, 14 and 15 respectively as:

Floor

$$\Delta P_{12-13} = \left\{ P_1 + \rho_o g \Delta z_{1,12} \right\} - \left\{ P_1 + C_{P-Ewall}\left(\frac{\rho U_{ref}{}^2}{2}\right) - \Delta P_{2-3} + \rho_i g \Delta z_{3,13} \right\}$$
$$= \rho_o g \Delta z_{1,12} - C_{P-Ewall}\left(\frac{\rho U_{ref}{}^2}{2}\right) + \Delta P_{2-3} - \rho_i g \Delta z_{3,13}$$
$$= 1.245\frac{\text{kg}}{\text{m}^3}9.8\frac{\text{m}}{\text{s}^2}1.00\ \text{m} - 0.7\left(\frac{(1.21\ \text{kg/m}^3)(3\ \text{m/s})^2}{2}\right) + 7.15\ \text{Pa} - 1.203\frac{\text{kg}}{\text{m}^3}9.8\frac{\text{m}}{\text{s}^2}0.75\text{m}$$
$$= 12.20 - 3.81 + 7.15 - 8.84 = 6.70\ \text{Pa}\ \textit{for I}\ [0.15\ \text{Pa}\ \textit{for II}]$$

(5.32)

Ceiling

$$\Delta P_{15-14} = \left\{ P_1 - \rho_o g \Delta z_{1,15} \right\} - \left\{ P_1 + C_{P-Ewall}\left(\frac{\rho U_{ref}{}^2}{2}\right) - \Delta P_{2-3} - \rho_i g \Delta z_{3,14} \right\}$$
$$= -\rho_o g \Delta z_{1,15} - C_{P-Ewall}\left(\frac{\rho U_{ref}{}^2}{2}\right) + \Delta P_{2-3} + \rho_i g \Delta z_{3,14}$$
$$= -1.245\frac{\text{kg}}{\text{m}^3}9.8\frac{\text{m}}{\text{s}^2}2.00\text{m} - 0.7\left(\frac{(1.21\ \text{kg/m}^3)(3\ \text{m/s})^2}{2}\right) + 7.15\ \text{Pa} + 1.203\frac{\text{kg}}{\text{m}^3}9.8\frac{\text{m}}{\text{s}^2}1.75\text{m}$$
$$= -24.40 - 3.81 + 7.15 + 20.63 = -0.43\ \text{Pa}\ \textit{for I}\ [-6.98\ \text{Pa}\ \textit{for II}]$$

(5.33)

5.2.1.2 Estimated Envelope Airflows

For this 140 m^2 house and an assumed normalized whole-house leakage of 0.35 – an airtightness required of new construction in the mid-latitudes of North America by ASHRAE Standard 119 (ASHRAE 1994) – the effective whole-house leakage would be about 0.049 m^2. Of this we will assume 10% is distributed to each of the two exposed walls, 20% is distributed to ceiling, and another 20% to the floor (see Chapter 23 of (ASHRAE 1997) or (ASHRAE 1994) for guidance here). For these assumptions and the pressure differences computed above, the envelope infiltration rates would then be:

$$\dot{V}_{Ewall} = 0.0049\ \mathrm{m}^2 \sqrt{\frac{2(7.15\ \mathrm{Pa})}{1.21\ \mathrm{kg/m}^3}} = 0.017\ \mathrm{m}^3/s\ for\ I\ [0.005\ \mathrm{m}^3/s\ for\ II] \tag{5.34}$$

$$\dot{V}_{Wwall} = 0.0049\ \mathrm{m}^2 \sqrt{\frac{2(1.16\ \mathrm{Pa})}{1.21\ \mathrm{kg/m}^3}} = 0.007\ \mathrm{m}^3/s\ for\ I\ [-0.015\ \mathrm{m}^3/s\ for\ II] \tag{5.35}$$

$$\dot{V}_{Floor} = 0.0098\ \mathrm{m}^2 \sqrt{\frac{2(6.70\ \mathrm{Pa})}{1.21\ \mathrm{kg/m}^3}} = 0.033\ \mathrm{m}^3/s\ for\ I\ [0.005\ \mathrm{m}^3/s\ for\ II] \tag{5.36}$$

$$\dot{V}_{Ceiling} = -0.0098\ \mathrm{m}^2 \sqrt{\frac{2(0.43\ \mathrm{Pa})}{1.21\ \mathrm{kg/m}^3}} = -0.008\ \mathrm{m}^3/s\ for\ I\ [-0.033\ \mathrm{m}^3/s\ for\ II] \tag{5.37}$$

These results are illustrated in Figure 5.6 along with the net continuity error for each case expressed as an infiltration over-estimation error (i.e., the sum of all infiltration flow rates minus the sum of all exfiltration flow rates). In both cases, there is a significant violation of continuity so, strictly speaking, the combined passive ventilation and infiltration airflow solutions are inadmissible – that is to say, these estimated airflows violate continuity and must be assumed to be in error.

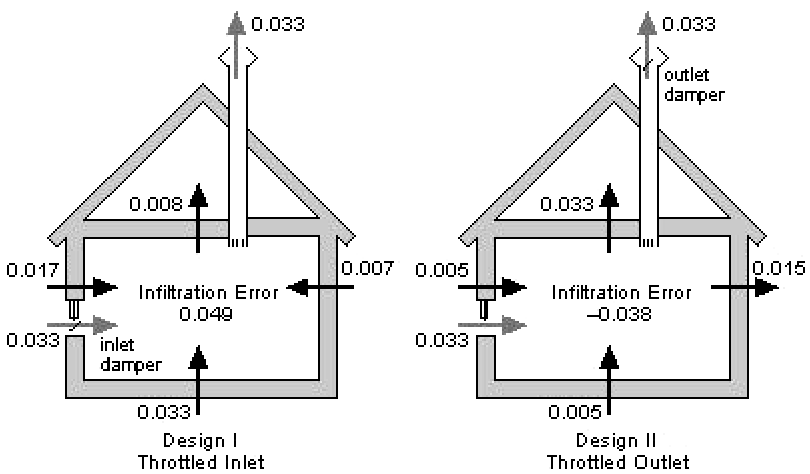


Figure 5.6 Estimated envelope airflows (m^3/s) based on pressure distributions corresponding to the "throttled inlet" Design Solution I and the "throttled outlet" Design Solution II.

5.2.1.3 Iterative Correction for Infiltration

Although in error, the estimated infiltration rates and the method used to determine them suggest a means to adjust the PSV system component sizes so that the infiltration error will become negligibly small. All estimated envelope pressure drops and airflows were based on the pressure drop in component a which, in turn, was dependent only on the squared ratio of the flow rate to the components unobstructed free area $(\dot{V}_a / A_a)^2$ – see Equation 5.30. Observing that the infiltration error computed above changes sign as the area A_a increases from 0.016 to 0.055 m^2 a value intermediate between these two values may be determined that will result in a negligible infiltration error. Using this value, the PSV components may then be resized and, hence an admissible solution to the combined infiltration/ventilation problem will have been determined.

Maintaining the objective ventilation rate of $\dot{V}_a$ = 0.033 m^3/s and completing the calculations required to estimate the infiltration error for trial values of A_a (i.e., evaluating Equations 5.30 through 5.37) we will find that a value of A_a = 0.0224 leads to a negligibly small infiltration error. The resulting envelope airflow rates are illustrated in Figure 5.7 for this solution.

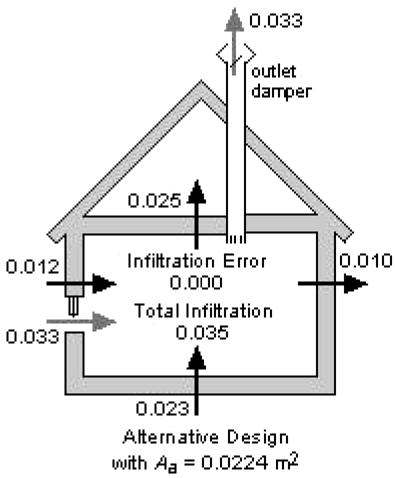


Figure 5.7 Estimated envelope airflows (m^3/s) based on pressure distributions corresponding to an alternative design with A_a = 0.0224 m^2.

To complete the correction, the value A_a = 0.0224 m^2 determined may be substituted into the governing loop equation, Equation 5.17, to determine the stack terminal loss coefficient C_c needed to maintain the objective flow rate of 0.033 m^3/s in the stack. Completing this exercise, we will discover C_c will have to be adjusted to 4.06. This could be accomplished by setting a damper to approximately 31° to the axis of the duct (ASHRAE 1997) – a setting slightly different from that determined earlier (see the discussion of *Operational Strategies* for *Design Example 1* above). The final design specification is presented in Table 5.1.

Table 5.2 Final design solution from Design Example 1 for windy conditions to provide a passive stack system ventilation rate of 0.35 ACH and compatible infiltration.

	Window Inlet A_a (m^2)	Room Outlet Loss Coef. C_b	Stack Duct D_s (m)	Stack Terminal Loss Coef. C_c	Pass. Vent. + Infiltration (m^3/s)
Final Design	0.0224	0.10	0.175	4.06 (31° damper)	0.033 + 0.035

Although we have managed to adjust the PSV system components so that the combined ventilation and infiltration problems are now admissible (i.e., both satisfy continuity) the computed infiltration rate is significant. Indeed, as indicated in Figure 5.7 it is slightly larger than the design ventilation rate and can, therefore satisfy the ventilation requirement alone!

This over-ventilation problem may be mitigated by further adjustments to the PSV system components to reduce the PSV system ventilation flow rate. To maintain the computed infiltration solution illustrated in Figure 5.7 while reducing the PSV system ventilation rate we need only maintain a constant ratio of $(\dot{V}_a / A_a)$ – i.e., a constant pressure drop across *a*. Following this strategy to the limit, if the PSV system is removed the envelope flow rates will remain as illustrated in Figure 5.7 and the total ventilation rate will be determined by the infiltration rate alone.

Alternatively, the PSV system components may be adjusted to actually reduce both the infiltration and PSV system ventilation so that the total ventilation rate approaches the design objective. It should be clear from the calculations completed thus far, however, that these adjustments will be subtly small and likely to be within the error of uncertainty of the effective leakage area of the whole house (i.e., on the order of ± 0.010 m^2 or equivalent). Consequently, while these fine adjustments may be made in principle the leakage characteristics of the house can not be expected to be known with sufficient accuracy *a priori* to justify the approach.

5.2.1.4 Conclusion to Design Example 3

We may generalize from this example and assert that an admissible solution to the combined passive ventilation and infiltration problem may be found by adjusting inlet resistances of the passive ventilation system so that continuity is achieved for the envelope infiltration and exfiltration flows. Furthermore, the loop equation design method may then be applied to adjust the sizes of the remaining components of the passive ventilation system to also achieve continuity of flows in this system. If, however, the estimated infiltration rate proves to be large relative to the objective (total) ventilation rate then further adjustments to the passive ventilation system to reduce the infiltration and, thus, the total ventilation rate are likely to be small relative to the uncertainty in the building leakage and thus not practically viable.

Again, one must conclude that the success of a passive ventilation system depends critically on the control of infiltration. Conversely, houses built to less stringent airtightness standards are not likely to require passive ventilation systems since infiltration alone may be expected to be sufficient. Indeed ASHRAE Standard 62-89 acknowledged (i.e., for past residential construction standards): *"The ventilation is normally satisfied by infiltration and natural ventilation."* (ASHRAE 1989). In the present context, however, this statement must be understood to apply only to houses that will incur no significant energy penalty because of the infiltration. Based on these observations a tentative passive design strategy may be formulated:

Passive Ventilation Design Strategy

Passive ventilation systems may not be warranted nor easily implemented in houses built to lower airtightness standards (e.g., houses with a normalized leakage greater than 0.35 or thereabouts).

5.3 Design Criteria & Design Conditions

Ventilation design criteria are most reasonably based on air quality objectives – e.g., maintaining indoor relative humidity levels between 30 and 70% or indoor carbon dioxide concentrations below 1,000 ppm. While, in principle, we can determine the ventilation rate time history needed to achieve a specified air quality objective, in practice the pollutant generation rate characteristics needed to make this determination may not be known with much certainty. Furthermore, even if these characteristics are known with certainty, the determination of ventilation rates will demand the solution of the difficult *inverse* contaminant dispersal problem (i.e., the determination of ventilation rates given objective indoor air concentrations, pollutant generation rate histories, building characteristics, and climatic conditions).

In dwellings, however, pollutant generation is often assumed to be either intermittent in character (e.g., due to cooking or bathing) or more or less constant (e.g., due to general "background" sources). Consequently, practical ventilation design and design criteria may be formulated by directing separate design strategies to each pollutant generation class. Thus ASHRAE Standard 62-89 specifies local ventilation be provided mechanically or using openable windows in those rooms subjected to intermittent generation of air pollutants (i.e., kitchens and bathrooms) while for living and sleeping areas a constant ventilation rate be provided for background sources (i.e., the lesser of 0.35 ACH or 15 cfm per person) (ASHRAE 1989).

This Technical Note takes the position that in North America, local ventilation for intermittent pollutant control should continue to be provided by local mechanical systems or openable windows and passive systems should be considered for general background ventilation. Furthermore, design criteria for this background ventilation should be based on prescribed ventilation rates, rather than air quality objectives, as air pollutant generation rates in residences are not likely to be known with sufficient certainty to warrant an air quality approach to design.

As airflow in passive ventilation systems is directly driven by wind and stack pressures, which are subject to variations in wind speed and outdoor air temperatures, passive ventilation systems can never provide truly constant ventilation rates. However, maintaining nearly constant air quality conditions – i.e., satisfying an air quality objective without the energy penalty of over-ventilation – will not, in general, demand constant ventilation as long as the average ventilation rate is acceptable. Thus, in establishing ventilation design criteria based on specified ventilation rates it is necessary to establish guidelines for time averaging ventilation rates and the wind speeds that drive them.

5.3.1 Wind Speed Time Averaging Period

To establish design criteria regarding appropriate wind speed averaging time periods, consider the behavior of a simple single-zone idealization of a dwelling subjected to a constant pollution generation rate G_i (mg/h) and a time varying air exchange rate $ACH(t)$ illustrated in Figure 5.8.

The time variation of the indoor air pollutant concentration $C_i(t)$ (mg/m^3) is related to the ventilation rate $\dot{V}(t)$ (m^3/h), the pollutant generation rate G_i, and the volume enclosed in the dwelling V (m^3) by demanding conservation of pollutant mass in the dwelling as:

$$V\frac{dC_i}{dt} = G_i - \dot{V}(t)\,C_i \quad or \quad \frac{dC_i}{dt} = \frac{G_i}{V} - ACH(t)\,C_i \tag{5.38}$$

where the air exchange rate is simply equal to the ventilation rate divided by the volume $ACH(t) = \dot{V}(t)/V$ (h^{-1}), outdoor air concentrations have been assumed to be zero for convenience, and the air within the dwelling is assumed to be well-mixed.

To investigate the influence of time varying ventilation rates on indoor air concentrations, the air exchange rate may be modeled as a harmonic function of time with a mean value equal to the ASHRAE prescribed ventilation rate (0.35 ACH) and an amplitude arbitrarily set to half this value as:

$$ACH(t) = 0.35 + \frac{0.35}{2}\sin\left(\frac{2\pi t}{T}\right) \tag{5.39}$$

where T (h) is the period of the variation.

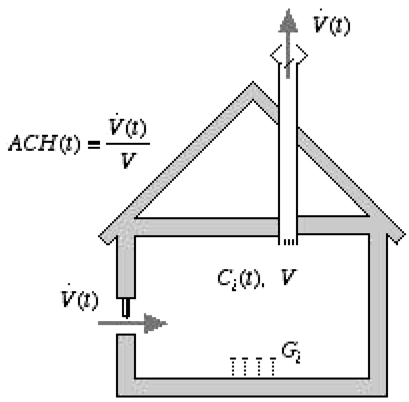


Figure 5.8 Single-zone idealiation of a dwelling subjected to a constant pollutant generation rate G_i and time varying ventilation rate ACH(t).

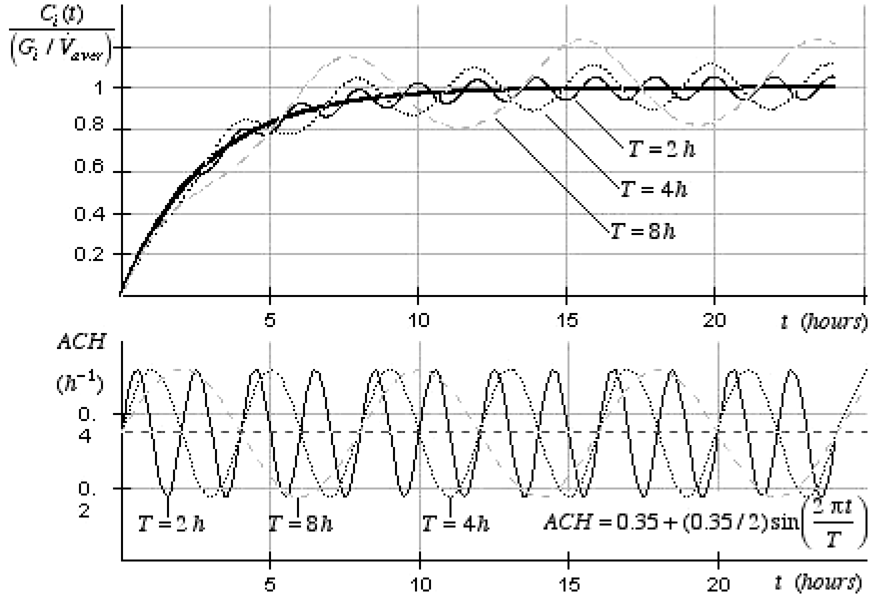


Figure 5.9 Indoor air concentration time histories for harmonically varying ventilation rates with mean values of 0.35 ACH.

Solutions to Equation 5.38 using the air exchange rate model of Equation 5.39 are plotted in Figure 5.9 along with plots of $ACH(t)$ for four cases: a steady exchange rate of 0.35 and harmonically varying exchange rates with periods of 2, 4, and 8 hours. For the first case, the indoor air concentration builds up gradually to a steady concentration equal to $G_i / \dot{V}$. The indoor air concentrations for the other cases oscillate about the steady-ventilation solution, yet stay within 10% of the steady-ventilation solution for exchange rates with periods of 4 hours or less – even for this relatively high amplitude flow variation.

In general, it may be shown that pollutant concentration time histories are insensitive to variations of ventilation rates as long as the period of variation of the ventilation rates are relatively small. That is to say, as long as the period of variation is smaller than approximately twice the mean ventilation rate *time constant* – the inverse of the mean ventilation rate (i.e., here 1/0.35 h^{-1} = 2.86 h). Given ventilation rates respond practically instantaneously to changes in wind speed, these results suggest that two to four-hour average wind speeds may be acceptable for wind pressure evaluation when applying the loop equation design method.
It is important to emphasize the link that is being made here between design criteria and design conditions. To maintain indoor air pollutant concentrations nearly constant at an acceptable level (i.e., the design criteria), wind speed (i.e., the design conditions) may reasonably be based on two to four-hour averages of actual wind speed records.

5.3.2 Accounting for Annual Variation

While the above conclusion allows the designer to ignore short-term wind speed variation, longer term wind speed variation must be addressed. Indeed, if a passive ventilation system is to maintain air quality for general background sources it must do so throughout the year. Thus, the annual variation of both wind speed and outdoor air temperatures must be accounted for.
The wind and stack terms of the loop equation, Equation 5.15:

Stack Pressure $$\Delta P_s \equiv g \sum \rho_i \Delta z_{ij} \tag{5.40}$$

Wind Pressure $$\Delta P_w = (\Delta C_P)\frac{\rho U_{ref}^{\ 2}}{2} \tag{5.41}$$

are directly and unambiguously determined by wind speed and temperature design data. Importantly, while these terms are related to the global geometry and topology of the proposed passive ventilation system they remain independent of system component sizes. Thus, given annual records of wind speed and temperature data, one may directly evaluate the detailed time histories of these driving pressures, $\Delta P_s(t)$ & $\Delta P_w(t)$, and use this information to size system components.

One could, for example, compute $\Delta P_s(t)$, $\Delta P_w(t)$ and their sum $\Delta P(t) \equiv \Delta P_s(t) + \Delta P_w(t)$, the total driving pressure, and extract minimum and maximum values to be used to size passive ventilation components. Given the stochastic nature of wind and outdoor air temperatures, it is more reasonable to use these time histories to extract probable minimum and maximum values to establish design conditions.

5.3.3 Design Example 4: Accounting for Annual Design Conditions

To make these ideas more concrete, reconsider the design of the innovative PSV system of Design Example 2 above, now, for a suburban site in the Boston area. The annual hourly records of wind speed, wind direction, and outdoor air temperature for the Boston meteorological station extracted from the ASHRAE WYEC2 tapes are illustrated in Figures 5.10, 5.11, and 5.12 (ASHRAE 1997).

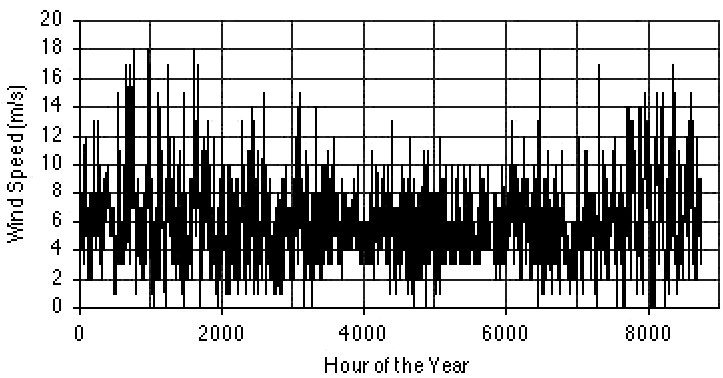


Figure 5.10 Annual hourly record of windpeeds for Boston from ASHRAE WYEC2 data (ASHRAE 1997).

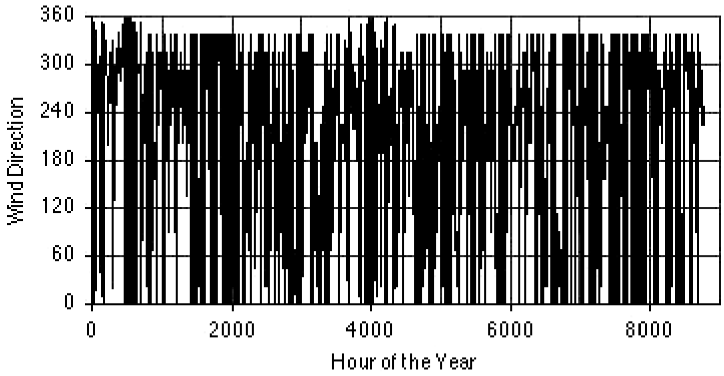


Figure 5.11 Annual hourly record of wind direction for Boston from ASHRAE WYEC2 data (ASHRAE 1997).

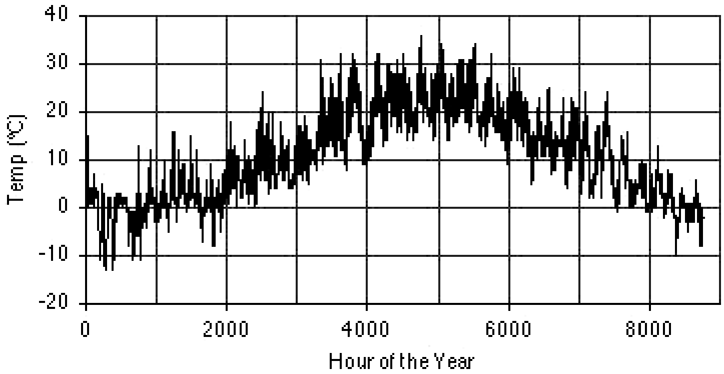


Figure 5.12 Annual hourly record of outdoor air temperatures for Boston from ASHRAE WYEC2 data (ASHRAE 1997).

Represented as time histories, the wind speed and direction data appears to somewhat random with speeds varying erratically from 1 m/s to 12 to 16 m/s and directions swinging wildly around the compass. Histograms of this data, Figures 5.13 and 5.14, better characterize the probable wind speeds and directions – both 1-hour and 3-hour average data are represented in these figures.

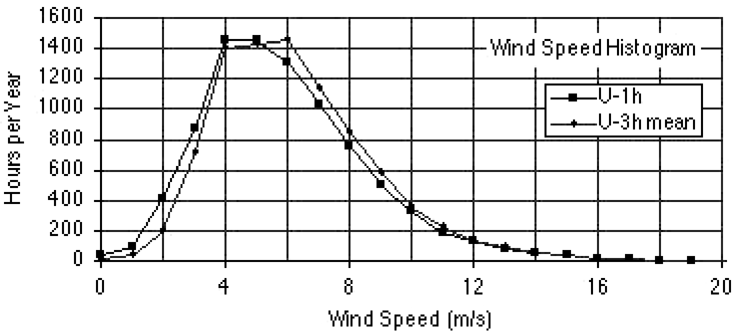


Figure 5.13 Histograms of the Boston wind speed data for 1-hour and 3-hour averaging time periods.

Figure 5.13 indicates wind speeds will fall between 2 m/s and 10 m/s for 90 % of the time (i.e., the portion of the histograms above a line through 438 hours or 5% of the year at both the low end and high end of the histogram). Figure 5.14 reveals that while winds from the western quadrant (i.e., from 240 ° to 300°) are most likely, winds from other directions may be expected for 5% or more of the time. The weak peak at 90° to 120° in the wind direction histogram, being diametrically opposed to the strong peak at 240° to 330°, might indicate a diurnal change in wind direction (e.g., on-shore and off-shore breezes) is at play. Finally, it is interesting to note that for this particular data set there is little difference between the 1-hour and 3-hour averaging periods.

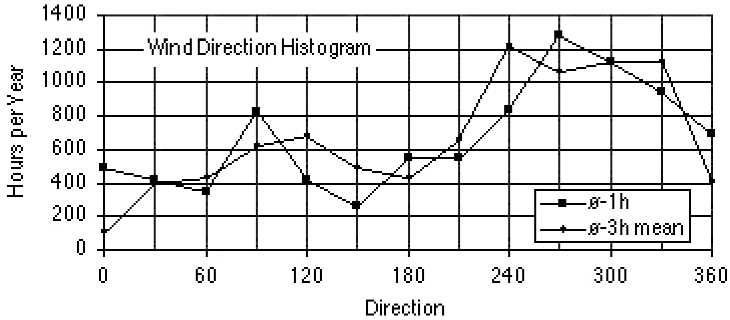


Figure 5.14 Histograms of the Boston wind direction data for 1-hour and 3-hour averaging time periods.

The global geometry of the building/PSV system will remain as specified earlier but, here, the self-regulating flow component will be placed in the stack terminal as a means to minimize infiltration – as illustrated in Figure 5.15. In this example, however, we will account for the annual variation in the reference wind speed data, $U_{ref}(t)$ and $\phi(t)$, and outdoor temperatures, $T_o(t)$, to establish *design conditions*.

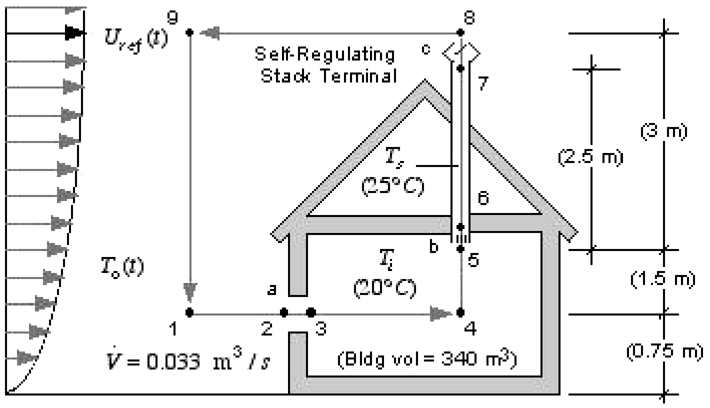


Figure 5.15 Macroscopic idealization of the building/innovative-PSV system to provide 0.35 ACH for Design Example 4.

The stack and wind pressure terms of the loop equation for this idealization are:

$$\Delta P_s = g\left(-\rho_i \Delta z_{45} - \rho_s \Delta z_{58} + \rho_o \Delta z_{91}\right) = 9.8\ \frac{\text{m}}{\text{s}}\left(-1.203\ \frac{\text{kg}}{\text{m}^3}1.5\ \text{m} - 1.183\ \frac{\text{kg}}{\text{m}^3}3\ m + \rho_o\left(T_o(t)\right)4.5\ \text{m}\right) \tag{5.42}$$

$$\Delta P_w = +C_{P2}\frac{\rho U_{ref}{}^2}{2} - C_{P8}\frac{\rho U_{ref}{}^2}{2} = \left(C_{P2}\left(\phi(t)\right) + 0.5\right)\frac{1.2\ \frac{kg}{m^3}U_{ref}(t)^2}{2} \tag{5.43}$$

where it has been assumed that the wind pressure coefficient of the stack terminal device (i.e., –0.5) is invariant with wind direction.

The air densities used in the above equation may be evaluated using the ideal gas relation presented earlier, Equation 4.5. Thus the indoor and stack air densities, ρ_i and ρ_s, for the given temperatures of 20 °C and 25°C are 1.203 kg/m^3 and 1.183 kg/m^3 respectively. The outdoor air density will vary with outdoor air temperature which, in turn, varies with time, $\rho_o\left(T_o(t)\right)$, but again the ideal gas relation may be used to directly compute the time variation of the outdoor air density using the WYEC2 data plotted above.

To apply Equations 5.42 and 5.43 one additional adjustment must be made. The WYEC2 wind speed data *U(t)* is for the Boston meteorological station – this data must be adjusted for the actual site conditions. A variety of techniques have been developed to estimate reference wind velocities at specific sites from meteorological data (Orme, Liddament et al. 1998) – here we will use the *terrain* adjustment relation favored by the ASHRAE *Handbook of Fundamentals* (ASHRAE 1997):

$$U_{ref} = UKz_h{}^a \tag{5.44}$$

where K and a are constants that characterize a given *terrain* and z_h is the height of the building being studied. In the present case the building height is z_h = 5.25 m and it is located in a suburban terrain which is characterized by K = 0.32 and a = 0.28. (A shelter factor, Equation 4.15, could also be applied at this point to correct for local sheltering of the building. Here it will be assumed that sheltering does not occur.)

5.3.3.1 Design Conditions

The annual hourly variation of stack and wind pressure may be directly determined by adjusting the WYEC2 wind speed data for the local terrain and substituting this data, the wind direction data, and the outdoor air temperature data into Equations 5.42 and 5.43. The results of this exercise are plotted in Figure 5.16 for the case where the inlet wind pressure coefficient is assumed to have a value of +0.7 that is invariant with wind speed.

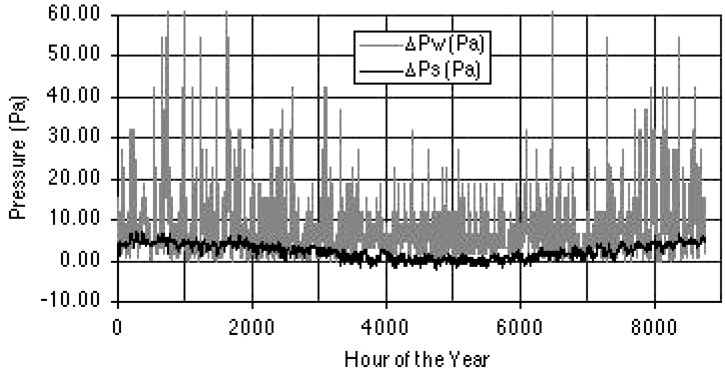


Figure 5.16 Annual hourly estimates of stack, ΔP_s, and wind pressure ΔP_w, for the given building global geometry and topology for constant inlet and outlet wind pressure coefficients of +0.7 and –0.5 respectively.

For the given building (global geometry and topology) located in a suburban site in the Boston area, then, it is seen that the wind pressure contribution, although erratic, is generally greater than the stack pressure contribution. In fact, during the summer months the stack pressure is negative at times (i.e., for the assumed indoor air temperature, presumably maintained by air conditioning or dynamic natural cooling strategies). Furthermore, it appears that truly calm conditions seldom prevail so design for the hypothetical no-wind condition considered in the earlier design examples is no longer justified. This specific example suggests a general principle:

> *Passive Ventilation Design Strategy*
>
> *Design for a hypothetical no-wind condition may not be justified for many sites if local site wind conditions are realistically evaluated. Instead a probable low-wind condition should be used.*

Ventilation flow rates in the passive ventilation system are driven by the sum of the stack and wind pressure contributions, $\Delta P = \Delta P_s + \Delta P_w$, thus it is most valuable to consider the variation of this sum. Figure 5.18 presents a histogram of this total pressure difference for two cases:

- Case 1: the constant inlet pressure coefficient case used to generate the data in Figure 5.16 and
- Case 2: a second case where the inlet wind pressure coefficient was assumed to vary from +0.7 to –0.3 for incident wind directions of 0° and 180° respectively.

The variation of the inlet wind pressure coefficient for the second case was modeled using the Walker/Wilson equation presented earlier, Equation 4.14, and is plotted in Figure 5.17.

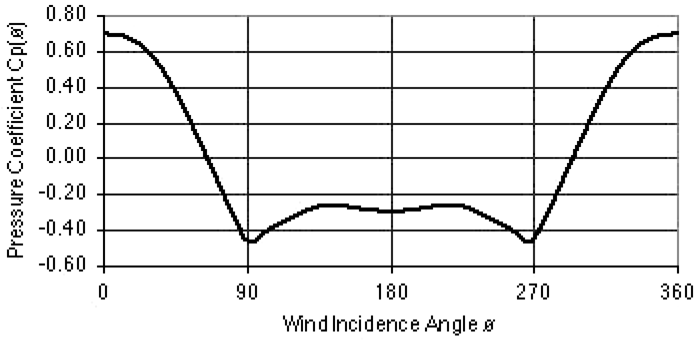


Figure 5.17 Assumed inlet wind pressure coefficient variation with wind direction for Design Example 4 – Case 2 generated using the Walker/Wilson model, Equation 4.14.

Figure 5.18 indicates that the wind and stack pressure sum will vary between 2 and 15 Pa for 90% of the time (i.e., values delimited by the intersection of a line at 438 hours or 5% of the year with the histogram) for the PSV system designed to be insensitive to wind direction (i.e., Case 1). Again, for this data set, the wind speed time averaging period is not significant. For the second, wind direction sensitive case the wind and stack pressure sum will vary between approximately 0.5 and 12 Pa for 90% of the time.

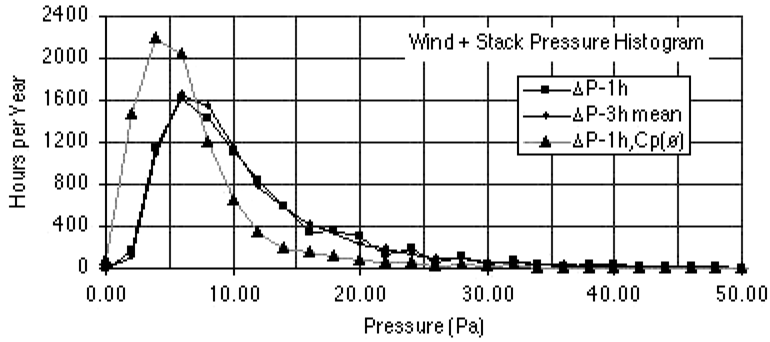


Figure 5.18 Histograms of stack and wind pressure for the building of Design Example 4 located on a suburban site in Boston for two cases. Case 1: inlet and outlet wind pressure coefficients assumed constant at +0.7 and –0.5 (for both 1-h and 3-h averaged wind speeds). Case 2: a variable inlet pressure coefficient ranging from +0.7 to –0.3 and a constant outlet pressure coefficient of –0.5 (for 1-h averaged wind speeds).

It is interesting to note at this point that Skaret and his colleagues computed similar histograms of the combined wind and buoyancy natural driving forces for both the spring and fall heating seasons of five northern European countries (Skaaret, Blom et al. 1997). For an assumed stack height of 10 m, an effective wind pressure contribution of 80% of the approach dynamic pressure, and normal indoor air temperatures – values appropriate for office buildings of moderate scale rather than residences – their results were broadly similar to those shown in Figure 18. Significantly, their results also indicate that truly calm conditions were unlikely.

5.3.3.2 Sizing System Components

Accepting a 90% probability as a reasonable basis for design conditions, we may move on to form the loop equations for each case so that system components may be sized. For this example, we will model the fixed area inlet, room outlet, stack duct as before in Design Example 2. The stack terminal will be modeled with the self-regulating vent model based on the Dutch device characteristics (i.e., with ΔP_o = 0.30 Pa). Formally, then, the loop equation assumes the following form:

$$\overbrace{\frac{\rho \dot{V}_a^2}{2C_d^2 A_a^2}}^{\text{window}} + \overbrace{\frac{\rho C_b \dot{V}_b^2}{2A_b^2}}^{\text{room outlet}} + \overbrace{\frac{f L_s \rho \dot{V}_s^2}{2D_{hs} A_s^2}}^{\text{stack duct}} \overbrace{- \Delta P_o \ln\left(1 - \frac{\dot{V}_c}{\dot{V}_o}\right)}^{\text{stack terminal}} = \Delta P_s + \Delta P_w \tag{5.45}$$

To this equation we will again impose the continuity condition that $\dot{V}_a = \dot{V}_b = \dot{V}_s = \dot{V}_c$ and set these ventilation flow rates equal to the objective rate of 0.033 m^3/s to achieve the desired 0.35 ACH. Finally, we will substitute the component parameter values and the geometric relations used in Design Example 2 (i.e., $A_b = A_s = \pi D_s^2/4$ and $D_{hs} = D_s$) to obtain:

$$\overbrace{\frac{0.00183}{A_a^2}}^{\text{room inlet}} + \overbrace{\frac{0.000107}{D_s^4}}^{\text{room outlet}} + \overbrace{\frac{0.0000961}{D_s^5}}^{\text{stack duct}} \overbrace{- (0.3)\ln\left(1 - \frac{0.033}{\dot{V}_o}\right)}^{\text{self-regulating stack terminal}} = \begin{cases} \overbrace{2\ \text{Pa}}^{\Delta P_{\min}} & \overbrace{15\ \text{Pa}}^{\Delta P_{\max}} & \text{Case 1} \\ 0.5\ \text{Pa} & 12\ \text{Pa} & \text{Case 2} \end{cases} \tag{5.46}$$

Again, we may evaluate the limiting asymptotes of each design variable in turn for the driving pressure terms. These limiting values are presented in Table 5.3 for the minimum combined wind and stack pressure differences – design parameter values needed to realize the 0.35 ACH (0.033 m^3/s) ventilation objective when natural driving forces are at their (probable) minimum.

Table 5.3 Minimum values of design parameters to achieve 0.35 ACH for Design Example 4.

	Room Inlet A_a (m^2)	Stack Duct D_s (m)	Self-Reg. Term. $\dot{V}_o$ (m^3/s)
Case 1	0.030	0.141	0.033
Case 2	0.060	0.188	0.041

Based on these results we could, for example, choose to use a stack duct diameter of 0.150 m (6 in) for the Case 1 design and 0.200 m (8 in) for Case 2. With these design decisions made (and substituted into Equation 5.46) feasible design solutions are now more narrowly defined by the following two equations:

$$\textit{Case 1:}\ \overbrace{\frac{0.00183}{A_a^2}}^{\text{room inlet}} + \overbrace{1.477}^{\substack{\text{room outlet}\\ \text{stack duct}}} \overbrace{- (0.3)\ln\left(1 - \frac{0.033}{\dot{V}_o}\right)}^{\text{self-regulating stack terminal}} = \overbrace{2\ Pa}^{\Delta P_{\min}} \text{ or } \overbrace{15\ Pa}^{\Delta P_{\max}} \tag{5.47}$$

$$\textit{Case 2:} \overbrace{\frac{0.00183}{A_a^2}}^{\text{room inlet}} + \overbrace{0.367}^{\substack{\text{room outlet}\\\text{stack duct}}} - \overbrace{(0.3)\ln\left(1-\frac{0.033}{\dot{V}_o}\right)}^{\text{self-regulating stack terminal}} = \overbrace{0.5\ Pa}^{\Delta P_{min}} \text{ or } \overbrace{12\ Pa}^{\Delta P_{max}} \tag{5.48}$$

Each of these relations define feasible design solutions to achieve the 0.35 ACH ventilation objective – they are plotted in Figure 5.19.

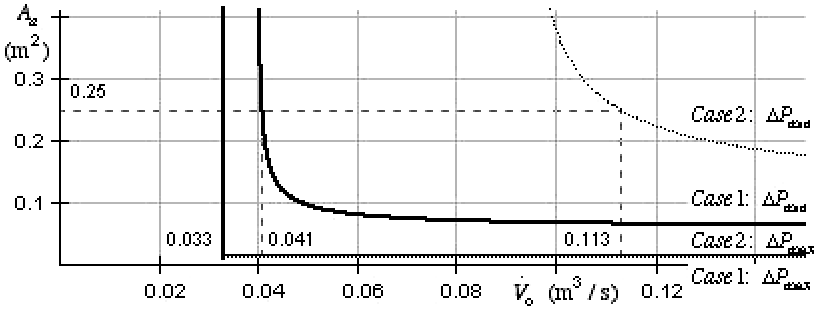


Figure 5.19 Feasible design curves for Cases 1 and 2 of Design Example 4 (Case 1 bold lines; Case 2 dotted lines).

From these curves, a final design specification may be selected for each case. For example, if we choose to set the room inlet area to 0.25 m^2, then for Case 1 the self-regulating terminal device would be set, ideally, to $\dot{V}_o$ = 0.033 m^3/s for maximum driving pressures and $\dot{V}_o$ = 0.041 m^3/s for minimum driving pressures. For Case 2 the self-regulating terminal device would be set, ideally, to $\dot{V}_o$ = 0.033 m^3/s for maximum driving pressures and $\dot{V}_o$ = 0.113 m^3/s for minimum driving pressures. These values are seen to be the intersections of the A_a = 0.25 line and the each of the four design curves of Figure 5.19.

In practice, however, we will not be able to regulate the self-regulating stack terminal device as the driving pressures change! Instead, a reasonable design value for $\dot{V}_o$ must be selected and over- or under-ventilation accepted as a consequence. Thus, for example, we could size the self-regulating terminal device based on the minimum driving pressure (i.e., 0.037 m^3/s for Case 1 and 0.111 m^3/s for Case 2) and accept the over-ventilation that will result at higher driving pressures. Given the greater spread in the design values for Case 2, it is clear that it will suffer proportionately greater over-ventilation.

The actual amount of over-ventilation that would result could be determined from the loop equations by solving for the airflow rate for the maximum driving pressure, given the selected design parameter values. If this is done it will be discovered that the ventilation rates for both cases will approach the nominal ventilation rate of the self-regulating device at the maximum driving pressure. For the target ventilation rate of 0.033 m^3/s (0.35 ACH), the wind-insensitive design, Case 1, will over-ventilate by approximately (0.041 – 0.033)/0.033 = 24% while the wind-sensitive design, Case 2, will over-ventilate by approximately (0.113 – 0.033)/0.033 = 242%. Clearly, the success of the self-regulating passive ventilation system depends on the proper matching of the self-regulating device characteristics with design conditions.

The Case 2 system, due to its wind direction sensitivity, operates at driving pressure differences that fall below the range for which the self-regulating device has authority (i.e., at or below the threshold pressure difference ΔP_o of the device). A general principle may be extracted from this observation and the cases considered:

> *Passive Design Strategy*
>
> *The global geometry, topology, and wind direction sensitivity of a passive ventilation system utilizing self-regulating flow control devices should be configured to produce design conditions (e.g., histograms of the sum of stack and wind pressures) such that minimum total pressures exceed the threshold pressure of the self-regulating device.*

Evaluation of the combined wind and stack pressure histogram not only provides an unambiguous means to establish design conditions for sizing components of passive ventilation systems, it provides the means to quantitatively evaluate the *controllability* of the system being proposed.

5.4 Conclusion

This chapter has introduced a design method for sizing components of passive ventilation systems based on the formation of pressure loop equations for macroscopic idealizations of building ventilation systems. The loop equations are formulated by summing system component relations that describe the pressure drops across each component in terms of volumetric airflow rates through the component. By directly specifying these volumetric flow rates – i.e., to achieve a specific ventilation objective and to satisfy continuity of flow – the resulting loop equations define feasible combinations of component design parameters that may be used to size all system components.

5.4.1 Validation, Error & Uncertainty

The approach has been presented without validation, beyond a few favorable comparisons with existing passive ventilation systems developed empirically over time. Consequently, questions of reliability and certainty of the method must be addressed. To the extent that the ventilation airflows being modeled are stationary, the conservation principles underlying the formulation of the loop equations – i.e., continuity and momentum (pressure) conservation – are truly fundamental and, therefore, are not likely to introduce error by themselves.

The component relations introduced are, however, empirical. Thus, these relations can be expected to introduce error. As these relations are simply inverse forms of the component equations used in multi-zone macroscopic simulation, it is likely that the reliability demonstrated by the use of these relations in simulation studies will be similar to that to be expected here. A number of validation studies have demonstrated that macroscopic analysis may be relatively reliable under ideal circumstances (i.e., for well-defined systems subjected to well-characterized excitations), but uncertainty in system characteristics and environmental excitations often prove critical (Liddament 1983; Bassett 1990; Fürbringer, Dorer et al. 1993; Fürbringer, Roulet et al. 1996; Fürbringer, Roulet et al. 1996; Roulet, Fürbringer et al. 1996).

Uncertainty in wind speed and direction at a given site and uncertainty in envelope leakage characteristics are generally considered most import in this regard. An approach to account for the annual variation of wind and outdoor temperature conditions has been presented in this Chapter to address the former source of error. The latter source of error – envelope leakage and the infiltration that results from it – is particularly important in the design of passive ventilation systems for air quality control. Infiltration can easily overwhelm a passive ventilation system and render it unable to control ventilation flow rates. Consequently, infiltration is not only a source of analysis error but limiting it is a design imperative in passive ventilation design – the control of infiltration through tight building construction is a prerequisite to effective passive ventilation design.

Regrettably the assumption of stationary flow is strictly not valid and is known to be the source of additional uncertainty. Wind is, in fact, characteristically turbulent and nonstationary, consequently the conservation of (mean) pressure changes while circuiting a loop must be expected to suffer some error. That turbulence introduces ventilation air exchange not accounted for by macroscopic analysis is well known (Sirén 1997; Knoll and Phaff 1998) yet the amount of additional air exchange introduced has not yet been well characterized. Sirén has estimated that infiltration exchange rates may be underestimated by as much as 20% due to this simplification and has formulated an approach to correct macroscopic modeling methods for it. Following Sirén's example it may be possible to extend the design method presented in this chapter to account for turbulence induced ventilation. In the meantime, designed ventilation rates must be expected to fall below actual values by as much as 20% for site subjected to particularly turbulent wind conditions – even when infiltration is controlled using tight construction.

5.4.2 Computational Implementation

The loop equation design method has been presented as a manual procedure demanding only algebraic and arithmetic manipulations of the governing pressure loop equations for macroscopic passive ventilation system idealizations. Yet even for the relatively simple yet practically useful examples considered the arithmetic manipulations proved to be tedious at best and the algebraic evaluation of asymptotic values sometimes involved the solution of challenging nonlinear algebraic equations. More complex yet equally practical passive ventilation systems may easily be conceived that may greatly exacerbate these problems. Thus the practical application of the proposed method clearly demands computational tools.

In fact, while all examples in this chapter (and those presented in the Appendix) have been presented in a step-by-step manner that could have been accomplished by hand, all calculations and algebraic manipulations were actually completed using symbolic math-processing programs. A number of such programs, including MathView™, MathCAD®, MATLAB®, Mathmatica™, and Maple V among others, are available that could be used but these and other math-processing programs are not particularly easy to learn or use.

Alternatively, specialized computational tools could be developed. Here, logically, these tools should be developed using the interface conventions that have emerged to support macroscopic airflow and air quality modeling – e.g., the interface conventions established by the CONTAM and COMIS programs (Feustel 1990; Feustel and Raynor-Hoosen 1990; Feustel and Smith 1992; Pelletret and Keilholz 1997; Walton 1997). These macroscopic modeling interfaces offer the user graphic *sketch pads* with which diagrams of macroscopic idealizations may be prepared that define the global geometry and topology of a building airflow network. For simulation, flow component model parameters and sizes are then defined via data input windows linked to the graphic representation of the building airflow network and component airflow rates are computed for selected environmental conditions. For design, the user would define flow component model parameters and objective design airflow rates instead. The user would then work interactively with the program, introducing additional design constraints, to determine candidate component sizes that will achieve the design airflow objectives.

This approach offers the advantage that the loop equation design method could be implemented within the structure of a macroscopic simulation program so that once a candidate design solution is selected annual simulations could be directly computed to evaluate the performance of the solution. Changes could then be made directly to the candidate solution to improve annual performance or, alternatively, the designer could turn once again to the loop equation design tools to modify or add design constraints and search for an alternative design solution. While it is clear that the existing interface conventions can support the necessary definition of global geometry and topology required to initiate the loop equation method and can greatly ease the burden of component model parameter definition, it is not at all clear how one might develop an interface that would support the interactive design search approach presented in this chapter. The definition of design constraints would necessarily be similar to equation definition found in symbolic math processing programs if truly general design constraints are to be supported. Alternatively, the interface could limit design constraints to equations defined in terms of simple linear combinations of design parameters.

In any event, for the time being symbolic math-processing programs should be considered the computational tools of choice for implementing the loop equation design method. Using these tools design solutions may be found that are far more site and building specific, complete, and detailed than possible with existing simplified design equations or approximate design methods. By extending existing network airflow simulation programs and borrowing interface conventions and computational algorithms from symbolic math-processing programs a new generation of building airflow tools may be developed that will provide integrated design and simulation tools. Tools that will support the design development search for system component sizes using the loop equation method and allow the direct and immediate simulation of candidate design solutions to evaluate annual performance.

6 North American Construction & Dwelling Codes

Residential construction in North America is governed by the provisions of building codes adopted by local jurisdictions. In the U.S., these local building codes are invariably based on *model building codes* developed by one of three organizations – the Building Officials and Code Administrators International, Inc. (BOCA), the International Conference of Building Officials (ICBO), or the Southern Building Code Congress International, Inc. (SBCCI). In recent years these three code-writing organizations have collaborated to establish a standard, uniform format for the model codes they publish and through a joint organization, the International Code Council, Inc., have published a model code directed specifically to residential construction – the *International One- and Two-Family Dwelling Code* (ICC 1998). This new model code incorporates the provisions and amendments of its immediate predecessor, the *CABO One and Two Family Dwelling Code* (CABO 1995).

Presently the *International One- and Two-Family Dwelling Code* has no specific provisions for passive ventilation systems beyond minimal window opening regulations that are intended to provide ventilation for rooms not mechanically ventilated. As passive ventilation systems influence air pressure distributions in dwellings they must be expected to interact with other natural and mechanical ventilation systems (e.g., vented combustion devices, fireplaces, and clothes dryer, bathroom and kitchen exhaust systems). The practical implementation of passive ventilation systems therefore demands consideration of this possible interaction. Furthermore, passive ventilation devices – most obviously passive stacks – have the potential to impact fire spread and smoke containment strategies that are central to the fire safety of dwellings. Again, this impact must be considered if passive ventilation systems are to be used. Finally, a number of provisions of the *Dwelling Code* relate to new air-tightness requirements that have been put into place to conserve energy used to heat or cool residences. In some instances these air-tightness regulations will have to be reconciled with the need to provide ventilation for air quality control when the latter is achieved using passive ventilation methods.

This chapter will review the provisions of the new *International One- and Two-Family Dwelling Code* that are likely to relate to the installation of passive ventilation systems for residential air quality control. This dwelling code is organized into chapters that address a) the general administration of the code and definition of terms, b) issues relating to building planning, c) specific construction requirements, and d) mechanical equipment requirements. Here, the building planning, construction, and mechanical equipment requirement provisions will be systematically reviewed and new or modified provisions for passive systems that may be added to this or other model dwelling codes will be formulated as appropriate.

6.1 Building Planning Provisions

The building planning provisions of the *International One- and Two-Family Dwelling Code* relate primarily to the life safety and health of the occupants and secondarily to the durability of the building itself. Two classes of provisions are relevant here – those relating to ventilation for health and those relating to fire safety.

6.1.1 Location on Lot

Section 302 *Location on Lot* of the *International One- and Two-Family Dwelling Code* establishes fire-safety provisions for exterior walls intended to mitigate the spread of fire from one dwelling to an adjacent dwelling. These provisions establish fire-resistive ratings of exterior walls and limit projections and openings in exterior walls when located less than 3 feet from the property line. Passive ventilation inlets and possibly outlets certainly fall within the scope of these provisions as they may introduce openings in exterior walls that could conceivably permit fire spread. Consequently, to satisfy the intent of these provisions the existing provisions could be amended to include:

> *Proposed Passive Ventilation Provision – Location on Lot*
>
> *Passive inlets and outlets should not be permitted in exterior walls of dwellings located less than 3 feet from building property lines unless they can be fabricated to provide a fire-resistive rating of one-hour or less. (The wording of this proposed provision is based on that contained in (ICC 1998).)*

6.1.2 Light and Ventilation

As currently written, Section 303 *Light and Ventilation* stipulates ventilation requirements for habitable rooms and bathrooms offering one of two choices for ventilation – ventilation by openable windows or by mechanical means. These provisions allow ventilation by openable windows to be replaced by mechanical systems but offer no other possibilities. To extend these provisions to allow passive ventilation as a third option a performance standard based on that offered for mechanical ventilation may be added:

> *Proposed Passive Ventilation Provision – Light and Ventilation*
>
> *Habitable Rooms: As an alternative to mechanical ventilation, an approved local passive ventilation system may be provided capable of producing 0.35 air changes per hour in the room or a whole-house passive ventilation system capable of supplying 15 cubic feet per minute per occupant computed on the basis of two occupants for the first bedroom and one occupant for each additional bedroom. These required passive ventilation rates may be based on 8-hour or less averages and may be adjusted to the actual occupancy of the dwelling by the dwelling occupants.*
>
> *Bathrooms: As an alternative to mechanical ventilation, an approved local passive ventilation system may be provided capable of producing a ventilation rate of 20 cubic feet per minute, averaged over an 8 hour period or less, without mechanical assistance and 50 cubic feet per minute on an intermittent basis with mechanical assistance. (The wording of this proposed provision is based on that contained in (ICC 1998).)*

This section also stipulates, in section 303.5 *Required Glazed Openings*, that ventilation air via openable windows be provided from outdoor sources – i.e., from yards, courtyards, streets, alleys, or open porches. These provisions may be directly extended to include passive ventilation systems by simply replicating the requirements for ‘‘glazed openings’’ to requirements for ‘‘glazed openings and passive ventilation inlets’’ as:

> *Proposed Passive Ventilation Provision – Required Openings & Inlet*
>
> *Required glazed openings and passive ventilation inlets shall open directly onto a street or public alley, or a yard or court located on the same lot as the building. (The wording of this proposed provision is based on that contained in (ICC 1998).)*

A similar modification to section 305.5.1 for openings and inlets onto roofed porches should also be made.

6.1.3 Dwelling Unit Separation

The provisions of Section 320 *Dwelling Unit Separation* stipulate fire-resistive and acoustical characteristics of walls and floor assemblies that separate dwelling units of two-family dwellings or town houses. These provisions are implicitly applicable to passive ventilation system installations and as such will limit the placement or penetration of ducts of passive ventilation systems in separating walls or floors.

6.1.4 Protection Against Radon

Section 324 *Protection Against Radon* stipulates that new dwellings in high radon level areas shall be constructed using the radon-control methods described in Appendix D of the dwelling code. Three methods are considered: a *sub-membrane depressurization system* for buildings built over crawl spaces and *active* (i.e., fan-powered) or *passive sub-slab depressurization systems* for buildings with slab-on-grade or basement slabs. As passive ventilation systems will normally act to depressurize a house, passive ventilation systems may act to defeat or compromise the sub-slab depressurization methods described in Appendix D. While the problem of building depressurization is addressed in section D103.11 of this appendix, the provisions outlined, which relate to duct sealing and infiltration control, are sketchy at best. Suffice it to say the interaction of passive ventilation and sub-slab depressurization systems demands further study.

6.2 Building Construction Provisions

The building construction provisions of the *International One- and Two-Family Dwelling Code* lay out specific requirements for construction assemblies and construction subsystems of the building proceeding from the building foundation up through the roof construction.

6.2.1 Wall Construction

Fireblocking is a fundamental requirement for wood-framed hollow cavity wall construction. It is installed to limit the extent of horizontal and vertical chases and thereby the spread of smoke and fire within walls. While the regulations set out in the *Dwelling Code* are quite general and need no special modifications for passive ventilation systems it is important to stress that these regulations will necessarily govern the construction of passive stacks and vents. Specifically, section 602.8 of the *Dwelling Code* requires noncombustible materials be used for the fireblocking around pipes, vents, and ducts and that the wall cavity around these components be packed with unfaced fiberglass batt insulation.

The air infiltration limits set out in section 609 *Windows* and 610 *Sliding Glass Doors* of the *Dwelling Code* presents a potential conflict between air-tightness and ventilation regulations that must be reconciled. These sections stipulate that *"no window (sliding glass door) shall be selected whose air infiltration exceeds 0.50 cubic feet per minute per lineal foot (0.774 L/s per m) of crack when tested in accordance with ASTM E 283 at a pressure differential of 1.56 psf (0.075 kN/m^2)."* Clearly, this provision would have to be modified to accommodate windows or doors fitted with trickle vents that are to be used as part of a passive ventilation system. A possible modification is:

Proposed Passive Ventilation Provision – Air Infiltration

No window (sliding glass door) shall be selected whose air infiltration exceeds 0.50 cubic feet per minute per lineal foot (0.774 L/s per m) of crack when tested in accordance with ASTM E 283 at a pressure differential of 1.56 psf (0.075 kN/m^2. Windows (sliding glass doors) fitted with purpose provided vents that are intended to function as part of a passive ventilation system should satisfy this criteria when the vent is completely blocked. When the vent is set to its intended operating position, however, the window-vent (sliding glass door-vent) system may provide natural ventilation that exceeds this limit providing this ventilation meets the design ventilation objectives of the dwelling. (The wording of this proposed provision is based on that contained in (ICC 1998).)

6.2.2 Chimneys & Fireplaces

Fireplace chimneys and passive ventilation stacks both utilize the natural driving forces provided by buoyancy and wind effects to develop the positive draft that serves to exhaust combustion gases in the former case and stale air in the latter. As these systems will normally act to depressurize a house each can act to reverse the stack flows of the other. Obviously, chimney backdrafting poses a health hazard that can not be tolerated. Less obviously, backdrafting or even a reduction in the airflow of a passive stack may reduce ventilation rates or alter ventilation air distribution adversely. Section 1006 *Exterior Air Supply* provides construction details to direct combustion air directly to fireplaces that will reduce the tendency of fireplace/chimney systems to depressurize a house. Without additional study however one can not be certain that these provisions will inhibit adverse interaction between passive ventilation and fireplace/chimney systems in all instances.

As illustrated in Design Example 5, multiple stack subsystems may be analyzed as a complete interacting system using the loop equation approach presented in Chapter 5 and measures may be taken to assure the relative independence of the respective subsystems. Using the loop equation method and/or multizone simulation methods based on the same fundamental theory, one may certainly study the interaction between passive ventilation and natural draft chimney systems. In this way one may, possibly, develop general guidelines for their design and construction that will mitigate the problems of competition that they are certain to present.

6.3 Building Mechanical Equipment Provisions

The building mechanical equipment provisions of the *International One- and Two-Family Dwelling Code* lay out specific requirements for mechanical equipment and systems used in dwellings.

6.3.1 Exhaust Systems

Passive ventilation systems must be expected to interact with mechanical exhaust systems for clothes dryers, range hoods, bathrooms, and the like. Again, this interaction may adversely affect the operation of the passive ventilation system but it has yet to be studied. Specific systems may be designed and analyzed to minimize adverse interactions using either the loop equation method presented in Chapter 5 or multizone simulation methods. Using these methods, it should also be possible to develop general guidelines for passive ventilation system design to inhibit adverse interaction. If passive ventilation systems are to become a viable alternative, such guidelines would have to be developed.

6.3.2 Duct Systems

The *Duct Systems* provisions of Chapter 19 of the *Dwelling Code* may be applied directly to duct construction for passive ventilation systems and, as such, should provide useful guidelines and construction details.

6.3.3 Combustion Air and Chimneys and Vents

Chapter 20 *Combustion Air* of the *Dwelling Code* and Chapter 21 *Chimneys and Vents* set out specific provisions to provide combustion air to fuel-burning equipment and to exhaust combustion gases to safeguard the health and safety of occupants. These provisions are, apparently, based on the assumption that the dwelling will be of conventional North American construction with the possibility that the construction may be unusually air-tight (i.e., in section 2001.1.1 *Buildings of unusually tight construction)*. The introduction of passive ventilation systems into North American residential construction demands careful reconsideration of these provisions.

It is likely that fuel-burning equipment supplied with combustion air from outdoors satisfying the provisions of section 2003 *All Air From Outdoors* will not be adversely affected by passive ventilation systems. Combustion devices receiving their combustion air from indoors and especially those devices utilizing natural draft vents, however, must be expected to act in competition to passive ventilation systems. Consequently, the potential interaction of these devices and passive ventilation systems demand careful consideration. It should be noted, however, the interaction of passive ventilation systems with these combustion devices is not likely to be more problematic than the interaction with mechanical exhaust and ventilation systems commonly used in residences. Clearly, the best practice in all circumstances would be to use combustion devices with outdoor air supply and sealed direct vents for combustion gases.

6.3.4 Plumbing Vents & Traps

Chapter 36 *Vents* of the *Dwelling Codes* provides detailed prescriptive provisions for the installation of plumbing vents in domestic waste water systems. These vents act to inhibit the removal of water in plumbing traps by the suction potentially created by the flow of waste water from plumbing fixtures. The water in plumbing traps seals gases within the plumbing system to inhibit entry into the dwelling. Again passive ventilation systems will act, in general, to depressurize the house and thus have the potential to influence the operation of plumbing vents or, more specifically, plumbing traps. The influence on water traps must, however, be expected to be negligibly small as even an extremely large building depressurization of 50 Pa (0.20 in of water) can be expected to offset the water in a trap by only 5 mm (0.20 in) – well within the minimum trap seal of 2 inches stipulated in Chapter 37 *Traps* of the *Dwelling Code*. Presumably, the influence on air admittance valves – that are allowed to be used in some limited circumstances in place of water traps – will also be negligible but here the manufacturer’s specifications should be checked to insure that this will be the case.

6.4 Conclusion

From this review of the *International One- and Two-Family Dwelling Code* it appears that few changes would have to be made to include passive ventilation systems as an option to provide ventilation for air quality control. First, of course, the ventilation provisions of the code would have to be extended to allow passive ventilation but this amounts to little more than an editorial revision of the code. Such a revision was presented above. Beyond this questions relating to fire safety and the impact of house depressurization on other building systems would have to be addressed.

To the extent that passive ventilation systems would be constructed of pipes, vents, and ducts the code provisions relating to the installation of this hardware – especially the fireblocking provisions – should serve to provide an acceptable level of fire safety for passive ventilation system installations. The modification of the dwelling unit separation provisions presented above may be expected to safeguard dwellings from the spread of fire from adjacent structures to the same or a similar level of protection provided for conventional dwelling construction by the current *Dwelling Code*.

The use of transfer grills, undercut doors, or other openings between rooms demands, however, further study and consideration. Passive ventilation necessarily requires continuity of airflow from inlet to outlet thus such openings will be required, in general. Unless equipped with smoke-activated dampers, these openings will allow smoke to move from room to room and thereby jeopardize the safety of occupants. In residences smoke can readily spread from room to room if doors are left open yet occupants can limit the spread of smoke (and fire) by simply closing doors. Transfer grills and similar devices could be equipped with operable shutters to achieve the same end providing a simple solution to the problem. Given the importance of the issue, however, the problem should be addressed by fire safety experts and studied with care and deliberation in an effort to formulate workable code provisions that will maintain fire safety yet provide for passive ventilation in residences.

The impact of house depressurization by passive ventilation systems on venting systems for combustion devices also demands expert review and careful, deliberate consideration. Although the degree of depressurization may be expected to be slight the consequences of backdrafting can be significant. The problem may be mitigated by directly supplying combustion devices with combustion air, by sealed venting systems, and by designing passive ventilation systems to minimize building depressurization and system interaction. The latter may be achieved using the loop equation design method introduced in Chapter 5. House depressurization is not a new phenomenon nor is it a phenomenon without advantages (e.g., condensation mitigation in wall construction in the colder North American climates), consequently it is likely that reasonable design guidelines and workable code provisions may be developed to address this issue using available knowledge and expertise.

Finally, the interaction of passive ventilation systems and mechanical exhaust systems (e.g., clothes dryers and kitchen and bath exhausts) warrants some consideration although, here, these mechanical systems are more likely to reduce the efficacy of the passive ventilation system rather than create a hazard. The control or minimization of this interaction, especially from the point of view of long term performance, must be considered however a matter of design rather than regulation. Again, the use of the loop equation design method coupled with annual system simulation provides the tools needed to address this particular design problem.

7 Conclusion

It may fairly be said to *ventilate* – *"to admit fresh air into in order to replace stale air"* – is as old as the wind. The historic etymology of ventilate may be traced to the Middle English word *ventilaten* – *"to blow away"* – that is a direct descendent from the Latin *ventilare* – *"to fan"* – and the Latin *ventus* – *"wind."* Prehistorically, these roots may be traced to the Norse word *vindr* – *"wind"* – and the Sanskrit *vati* – *"he blows"* (Morris 1979). Certainly the desire to ventilate is as ancient as the words used to describe the desire and thus the means devised to ventilate dwellings most certainly predates any written history of it.

The general purpose of ventilation is contained within its definition, which again suggests this purpose has been understood for quite some time. While the specific reasons for ventilation have been and still remain controversial, medical and building scientists easily agree that we seek to ventilate buildings to provide *"air in which there are no known contaminants at harmful concentrations … and with which a substantial majority … of the people exposed do not express dissatisfaction"* (ASHRAE 1989). More positively, we also ventilate buildings for the pleasures of fresh air.

The means that should be used to ventilate has, however, become quite controversial. During the seasons we condition air within dwellings energy is consumed in providing ventilation. Thus we seek to a) avoid unnecessary *over-ventilation* at all costs and b) *ventilate efficiently* otherwise. In addition, to avoid health and moisture-related building problems we seek to provide sufficient ventilation – i.e., we try to avoid *under-ventilation.* The controversy arises because building technologists are not sure what ventilation systems will reliably provide the control to avoid both *under-* and *over-ventilation* and yet minimize energy use both within reasonable first and operating cost considerations.

Commonly we provide ventilation in dwellings by a) unintended natural ventilation – *infiltration*, b) purpose-provided natural ventilation – *passive ventilation*, and/or c) mechanically induced ventilation. All agree that infiltration can not provide the control needed to avoid both *under-ventilation* and *over-ventilation*. Thus, notwithstanding the potential heat recovery that *dynamic insulation* may offer, we now seek to minimize infiltration but accept the fact that buildings can not be made completely airtight. Beyond the (hopefully) minimal contribution provided by infiltration, the contenders in the ventilation controversy are, therefore, *passive ventilation* and *mechanical ventilation*.

It is often tacitly assumed that *mechanical ventilation* can provide the control desired – even though field studies to support this assumption are few (see (Lubliner 1999) for one exception) – and *passive ventilation* can not. This Technical Note has reviewed recent research work that has led to the development of *self-regulating vents* that appear to be able to provide the control needed at modest cost. Given the central importance of these devices, an empirical pressure-flow model for these self-regulating vents has been proposed that may be used in network airflow analysis programs to investigate the performance of ventilation systems, passive or mechanical, using these vents (see section 4.1). The use of the inverse form of this component model to size this and other components of passive ventilation systems has also been demonstrated (see Chapter 5).

It is also often assumed that *mechanical ventilation* systems can be outfitted with heat recovery devices while *passive ventilation* systems can not. While much remains to be done, recent research and development work completed initially by the SAVEHEAT research project (see section 2.2) offers great promise. Both innovative air-to-air heat recovery schemes using heat-pipe technology providing heat recovery rates of up to 55% and schemes using dynamic insulation are being considered (see section 2.2). The prospect of a passive ventilation system with self-regulating inlet and/or outlet vents that also provides acceptable levels of heat recovery (i.e., on the order of 40% or more) is, of course, very attractive as it may offer well-controlled, energy efficient, and reliable ventilation at modest first cost and negligible operating costs. For both passive and mechanical ventilation systems, however, infiltration acts parasitically to reduce net heat recovery rates. Thus again, effective ventilation system design demands minimization of infiltration and thus airtight building construction.

Air-to-air heat recovery devices offer resistance to airflow. Consequently, in mechanical ventilation systems they must be expected to increase fan-power consumption while in passive ventilation systems the added resistance will simply act to reduce ventilation airflow rates. In addition, these devices, by design, reduce exhaust air temperatures and thus the buoyancy forces induced by these temperatures. To the extent that a given passive ventilation system relies on buoyancy forces to drive airflow, then, air-to-air heat recovery devices are doubly problematic. To compensate for reduced airflows resulting from heat recovery, the SAVEHEAT research project has investigated two means to assist passive ventilation – using a solar chimney and a mechanical fan driven by a wind-turbine (see section 2.2).

Alternatively, passive ventilation systems fitted with heat recovery devices may be devised to rely more heavily on wind forces. Strategies have been identified in this Technical Note that will reduce the wind-direction sensitivity of passive ventilation systems and thereby improve the ability of these systems to control ventilation airflow rates. These strategies relate to the passive stack configuration and termination and fresh air inlet configuration and details (see Chapter 2). Chapter 3 presents two system strategies that may be expected to contribute to wind direction insensitivity – a subfloor inlet plenum configuration and a balanced wind stack system that is also well suited to heat recovery. Design calculations completed in Chapter 5 demonstrate the value of self-regulating vents in eliminating over-ventilation and mitigating under-ventilation due to wind speed variation in both systems.

Beyond this, a designer must consider the possibility of still wind conditions. An analytical procedure, introduced in Chapter 5, provides the means to account for annual variations of wind speed and direction in a rigorous and complete manner that may be used to determine if still conditions are likely to be significant at a given site. In spite of the nearly universal concern for still conditions in passive ventilation system design, it is likely that truly still conditions will not be a significant design condition at many sites.

Passive ventilation systems must be seen as assemblages of components that define a ventilation airflow path. Following this path from inlet to exhaust and returning to the inlet defines a ventilation *loop*. A deeper understanding and, in fact, a complete design method for sizing the components of a passive ventilation system may be developed by simply accounting for pressure changes due to airflow as one proceeds around a given ventilation loop. This is done in Chapter 5. Some nine component relations are presented that may be used to account for pressure changes in passive system components. These include power law, effective leakage area, orifice, self-regulating vent, duct, duct fitting, and fan component relations. The hydrostatic equation and correlations of surface wind pressure with the dynamic pressure of the approach wind (i.e., using surface averaged mean wind pressure coefficients) may be used to account unambiguously for buoyancy and wind induced pressure changes. The resulting *pressure loop equations* have been, uncommonly, used in the past to predict system airflow rates given wind and temperature conditions. Chapter 5 demonstrates how these equations can be used to size components of passive ventilation systems defining a design method that is not only ‘‘exact’’ but also far more complete than other methods presented in the literature.

With even a handful of components, a seemingly endless variety of systems may be (and have been) assembled. One system – the *passive stack ventilation (PSV)* system – has, however, been the focus of nearly all European research being tacitly accepted as the most effective choice for residential applications. This system is configured of limiting resistance inlets (e.g., *trickle ventilators*), one or more vertical stacks terminated, ideally, above the roof ridge with a properly detailed terminal device, and room-to-room *transfer grills* or other openings to complete the ventilation loop(s). The traditional PSV system has, however, many shortcomings. It tends to over-ventilate in windy or cold conditions, under-ventilate in still and moderate temperature conditions, provides no heat recovery, provides poor air distribution, and may produce cold drafts and back draft on occasions. The development of self-regulating vents have greatly improved the performance, it appears, of the PSV system but much of this development and the research preceding it has been illogically directed toward developing a single specification for PSV systems that may be applied to all dwellings regardless of dwelling size, climate, or site conditions.

To adapt this system to the North American context – and, for that matter, to better utilize it in the European context – the "one-size-fits-all" approach taken to-date must be replaced with a design method that accounts for building size and configuration, local climate and site conditions. The *loop equation design method* presented in Chapter 5 offers such a method and examples of its application to the design of traditional and innovative PSV systems are included. As the basis of this design method is general, it may also be applied to size components of an endless variety of other passive ventilation systems. Examples of its application to a centralized passive stack system, a cross-ventilation system, a balanced wind stack system, and a fan-assisted balanced stack system are also included. One approach has not been included in these examples – the use of so-called "single-sided ventilation" – i.e., ventilation provided through a single opening. This approach has been omitted primarily because it is not an appropriate strategy for general background ventilation although it may be a strategy of last resort for smaller, site-constrained dwellings.

Beyond sizing components *pressure loop equations* may be used in design to impose constraints relating to both technical matters (e.g., to limit cross-flow between stacks of a dual stack system) and non-technical objectives (e.g., to limit component sizes to off-the-shelf sizes) and to predict system performance once a design has been specified. Again, examples of these uses are also included in Chapter 5.

As central as a design method may be to promote the use or at least the consideration of passive ventilation in North America it alone is not sufficient to achieve this objective. A variety of associated design issues related to comfort, health and safety and interaction with other building systems must be addressed. Of these, perhaps the most problematic relate to potential combustion gas spillage that can conceivably occur due to house depressurization caused by (some) passive ventilation system configurations and the problem of controlling smoke and fire spread while still allowing free movement of ventilation air through a dwelling (see section 2.3). While these problems are not unique to dwellings with passive ventilation they demand careful consideration. IEA Annex 27 – Energy Conservation in Buildings and Community Systems Program – recently addressed the problem of the interaction of residential ventilation systems with combustion appliances and radon mitigation strategies (Månsson 1995) but additional research is warranted.

In spite of an extremely long history of use in the United Kingdom, British building codes have only recently endorsed passive ventilation systems. The 1995 edition of the British building regulation *Approved Document F* allows the use of the traditional PSV system and provides <u>prescriptive</u> details and specifications for trickle ventilators, "air bricks" with "hit or miss" ventilators, partial window openings, room-to-room openings, and stack ducts that are intended to provide necessary background ventilation for British homes (Department_of_the_Environment 1995). Ironically, British researchers among others have demonstrated the shortcomings of the traditional PSV system (i.e., discussed above) endorsed by these regulations and have, at least implicitly, identified the need to approach passive ventilation system design from a <u>performance</u> perspective.

With this in mind, Chapter 6 of this Technical Note has reviewed the very recent *International One- and Two-Family Dwelling Code* that is intended to become the model residential building code for North America (ICC 1998). At this time this code includes no provisions whatsoever for passive ventilation. Chapter 6 systematically identifies those provisions of the new *International One- and Two-Family Dwelling Code* that are likely to relate to the installation of passive ventilation systems and formulates proposals for new or modified provisions that would be needed to support a performance approach to passive ventilation system design in North America.

The position of this Technical Note should be clear. Passive ventilation should be considered as a viable option to provide background ventilation for dwellings in all climates of North America. Unlike the European context, however, these passive ventilation systems must be designed to be compatible with mechanical exhaust systems used commonly in bathrooms and kitchens, air-conditioning systems used in the hotter climates, and the variety of heating systems used throughout North America. This coupled with the wider range of climate types in North America will require a variety of passive ventilation systems – i.e., the nearly universal PSV system of northern Europe is not appropriate for all North American climates – thus demanding a performance-based design approach and a design method to support it. Pressure-dependent self-regulation of passive systems provides the key to reliable control thus there is a clear need to direct further research and development to these key components. Likewise, heat recovery for passive ventilation systems – the "missing link" to energy efficiency in passive ventilation – demands further research and development if the full potential of passive ventilation is to be realized.

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A1 Additional Design Examples

Three additional design examples are presented in this appendix to demonstrate the application of the loop equation design method to more complex systems – a multiple loop system, a cross-ventilated system, a wind-driven balanced stack system, and a fan-assisted balanced stack system. As a general method, the loop equation design method can be applied to an endless variety of systems - these examples may serve to help build the skill needed to do so.

A.1 Multiple Loop Systems

So far, the design examples presented have been limited to passive ventilation systems defined by single loops yet passive ventilation system in any real situation will involve multiple loops. In a centralized passive stack system, for example, air will flow in through a variety of inlet vents, windows, and doors to the central stack. Each inlet flow will be associated with a separate flow loop. If the inlets are designed, however, to be insensitive to wind direction (e.g., if baffled or linked by a plenum so that they all have the same effective wind pressure coefficient) then they may be *lumped* together into a single inlet and analysis may proceed as in the design examples considered so far.

A *lumped* idealization may not be reasonable for passive ventilation systems with multiple stacks. For these systems, and other systems where the designer wishes to retain a detailed system idealization, the designer must consider multiple loop equations. This becomes particularly relevant for multiple passive stack systems if it is important to design the systems so that each stack serves a local ventilation purpose and cross-flow between the stacks is to be avoided.

The procedures used to design ventilation systems modeled by multiple loops are identical to those used for single-loop idealizations. With interlaced multiple loops, however, the designer must be careful to assign ventilation flow rates to each component of the loop so that continuity of airflow is satisfied.

A.1.1 Design Example 5: Sizing Components of a Dual Stack System

To provide a first example of sizing passive ventilation components for a multiple loop situation consider the design of the dual stack system illustrated in Figure A.1. In this case, the left stack is to be designed to provide local ventilation to, say, a kitchen and the taller central stack is to provide general background ventilation for the rest of the house. Both ventilation loops will be topologically similar – inlet vents will admit air to the rooms and a passive stack will provide the exhaust air path. The stacks will be configured of a flush, square inlet at the ceiling level, a straight circular duct, and a stack terminal device with a self-regulating damper.

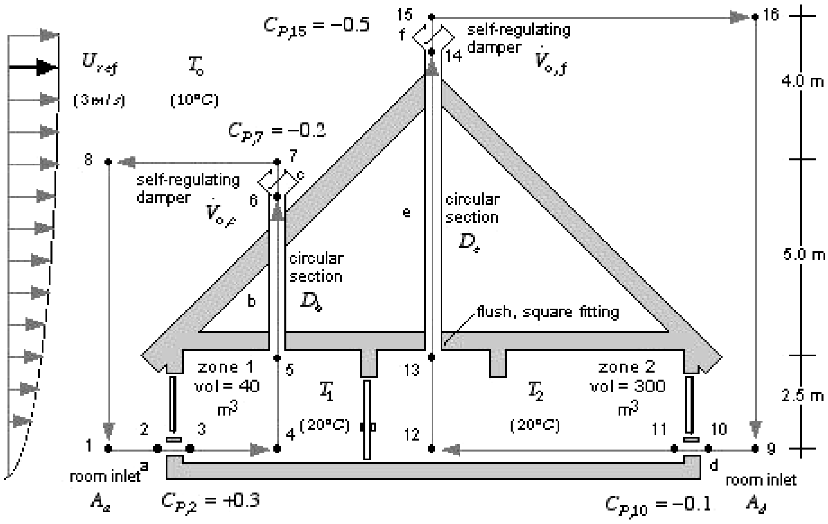

Figure A.1 Dual stack passive ventilation system for Design Example 5.

The system topology will be modeled as:

- the inlets will be modeled using the orifice model with opening areas of A_a and A_d,
- the ceiling fitting will be modeled by the duct fitting model (Equation 5.10) with the cross-sectional area taken to be equal to each stack area $A = \pi D^2/4$ and the fitting loss coefficient $C = 0.50$ (ASHRAE 1997),
- the stack will be modeled by the duct model (Equation 5.7) using circular sections of diameters D_b and D_e with *f* taken as 0.03, and
- the stack terminal device will be modeled using the self-regulating component model (Equation 5.6) with design variables $\dot{V}_{o,c}$ and $\dot{V}_{o,f}$ as indicated in Figure A.1, and a threshold pressure ΔP_o taken as 0.30 (i.e., based on the Dutch device discussed above).

System geometry is shown in Figure A.1 – the local ventilation stack is 5 m high, the central ventilation stack 9 m high and the effective room height is 2.5 m.

Design conditions will be as indicated in Figure A.1 – outdoor air temperature will be assumed to be 10 °C, indoor air temperatures will be 20 °C, and the reference wind speed will be 3 m/s. Applying the ideal gas law, Equation 4.5, the air densities at these two temperatures may be evaluated to be 1.245 and 1.203 kg/m^3 respectively. The wind pressure coefficient of the windward inlet is assumed to be +0.3, the leeward inlet – 0.1, the local stack – 0.2, and the central stack –0.5.

Finally, the objective design criteria will be to provide a background ventilation rate of 0.35 ACH for the larger volume (300 m^3) of the building while providing local ventilation at the rate of 2 ACH for the smaller volume (40 m^3) room. In terms of component volumetric flow rates, these ventilation rates become:

$$\textit{Local Loop}\; V_a = V_b = 2.0\,\mathrm{h}^{-1}(40\,\mathrm{m}^3)(1\,\mathrm{h}/3600\,s) = 0.022\,\mathrm{m}^3/s \qquad \text{(A.1)}$$

$$\textit{Central Loop}\; V_d = V_e = 0.35\,\mathrm{h}^{-1}(300\,\mathrm{m}^3)(1\,\mathrm{h}/3600\,s) = 0.029\,\mathrm{m}^3/s \qquad \text{(A.2)}$$

With these modeling decisions made and design criteria and conditions established, the loop equations may be directly formulated. In consistent units of m, s, and kg these equations are:

Local Loop 1-2-3-4-5-6-7-8-1

$$\frac{1.203\times 0.022^2}{2(0.60)^2 A_a^{\ 2}}+\frac{1.203\times 0.5\times 0.022^2}{2\left(\frac{\pi D_b^{\ 2}}{4}\right)^2}+\left(\frac{0.03\times 5}{2}\right)\frac{1.203\times 0.022^2}{D_b\left(\frac{\pi D_b^{\ 2}}{4}\right)^2}-0.30\ln\left(1-\frac{0.022}{\dot{V}_{o,c}}\right)$$
$$=\overbrace{(0.50)\frac{1.245\times 3.0^2}{2}}^{\Delta P_w}+\overbrace{(1.245-1.203)\times 7.5\times 9.8}^{\Delta P_s}$$

(A.3a)

or

$$\frac{0.0008087}{A_a^{\ 2}}+\frac{0.0002360}{D_b^{\ 4}}+\frac{7.079\times 10^{-5}}{D_b^{\ 5}}-0.30\ln\left(1-\frac{0.022}{\dot{V}_{o,a}}\right)=\overbrace{2.80\ \text{Pa}}^{\Delta P_w}+\overbrace{3.09\ \text{Pa}}^{\Delta P_s}$$

(A.3b)

Central Loop9-10-11-12-13-14-15-16-1

$$\frac{0.001405}{A_d^{\ 2}}+\frac{0.0004100}{D_e^{\ 4}}+\frac{0.0002214}{D_e^{\ 5}}-0.30\ln\left(1-\frac{0.029}{\dot{V}_{o,a}}\right)=\overbrace{2.24\ \text{Pa}}^{\Delta P_w}+\overbrace{4.73\ \text{Pa}}^{\Delta P_s} \quad \text{(A.4)}$$

To achieve the ventilation rate control desired – i.e., 0.022 m^3/s in the local loop and 0.029 m^3/s in the central loop – the nominal sizes of the self-regulating stack dampers must be set slightly larger than the required ventilation rates. Let's set $V_{o,c}$ = 0.025 m^3/s and $V_{o,f}$ = 0.035 m^3/s. With these constraints imposed, Equations A.3 and A.4 become:

Local Loop 1-2-3-4-5-6-7-8-1

$$\frac{0.0008087}{A_a^{\ 2}}+\frac{0.0002360}{D_b^{\ 4}}+\frac{7.079\times 10^{-5}}{D_b^{\ 5}}+0.64\ \text{Pa}=\overbrace{2.80\ \text{Pa}}^{\Delta P_w}+\overbrace{3.09\ \text{Pa}}^{\Delta P_s} \quad \text{(A.5)}$$

Central Loop 9-10-11-12-13-14-15-16-1

$$\frac{0.0001405}{A_d^{\;2}}+\frac{0.0004100}{D_e^{\;4}}+\frac{0.0002214}{D_e^{\;5}}+0.53\,\text{Pa}=\overbrace{2.24\,\text{Pa}}^{\Delta P_w}+\overbrace{4.73\,\text{Pa}}^{\Delta P_s} \tag{A.6}$$

Equations A.5 and A.6 define similar feasible design curves (i.e., as plotted in Figure A.2) that could be used to finalize design decisions.

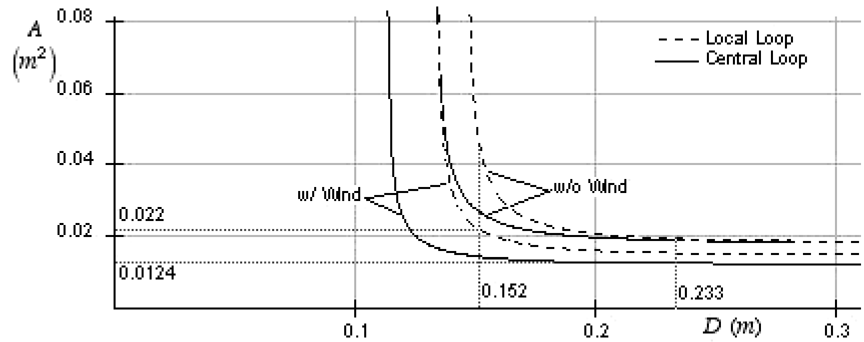

Figure A.2 Feasible design curves (i.e., inlet opening area A vs stack diameter D) for the local and central loops of Design Example 5.

In this two-stack system, however, we would like to size components so that cross-flow between the two ventilation loops is minimized. This could be achieved if the pressures at nodes 4 and 12 are identical – hence, no airflow between these nodes will result:

$$P_4 = P_{12} \tag{A.7}$$

Given the design ventilation rates, the pressures at these two nodes can be related to the inlet vents A_a and A_d for the with-wind case as:

$$P_1 + C_{P2}\frac{\rho U_{ref}^{\;2}}{2}-\frac{\rho V_a^{\;2}}{2C_d^{\;2}A_a^{\;2}} = P_9 + C_{P10}\frac{\rho U_{ref}^{\;2}}{2}-\frac{\rho V_d^{\;2}}{2C_d^{\;2}A_d^{\;2}} \tag{A.8a}$$

with $P_1 = P_9$ and substituting values used above:

$$\textit{With-wind Constraint}\quad A_a = \sqrt{\frac{0.0008087}{0.001405\,A_d^{\;2}+2.165}} \tag{A.8b}$$

For the without-wind case, the constraint becomes:

$$P_1 - \frac{\rho V_a^{\;2}}{2C_d^{\;2}A_a^{\;2}} = P_9 - \frac{\rho V_d^{\;2}}{2C_d^{\;2}A_d^{\;2}} \tag{A.9a}$$

or:

Without-wind Constraint $A_a = 0.7586\, A_d$ (A.9b)

Both cross-flow *design constraints* are plotted below in Figure A.3. The with-wind constraint has an asymptotic limit that places practical limits on component sizing. In addition, the loop equations, Equations A.5 and A.6, are bounded by asymptotic limits. In this case, to inhibit cross-flow the local loop inlet vent A_a must not exceed 0.0193 m^2 yet, due to the asymptotic limit implicit to Equation A.5, A_a must exceed 0.0124 m^2 to be able to provide the desired airflow rate (i.e., for the with-wind case). These limits are also indicated in Figure A.3.

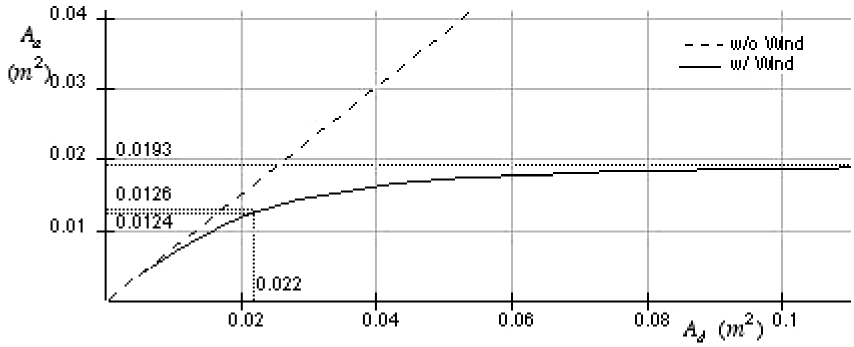

Figure A.3 Plot of the cross-flow constraint condition, Equation A.7.

With the feasible design curves and the cross-flow constraints, Equation A.8 and A.9, in hand a number of approaches may be taken to finalize the design.

A.1.1.1 With-Wind Design

Here, we will seek to satisfy the cross-flow constraint condition for the with-wind case by setting the central loop inlet area to A_d = 0.022 m^2, just within the asymptotic limits shown in Figure A.3. With this decision made, all remaining design variables may be determined as indicated graphically in Figures A.2 and A.3. An accurate determination of these values demands appropriate computational tools – a symbolic and numeric math processor program was used here. The final with-wind design is then:

With-Wind Design
$$\begin{aligned} A_a &= 0.0126\,\mathrm{m}^2\,;\, D_b = 0.233\,\mathrm{m}\,;\, \dot{V}_{o,c} = 0.025\,\mathrm{m}^3/\mathrm{s} \\ A_d &= 0.022\,\mathrm{m}^2\,;\, D_e = 0.152\,\mathrm{m}\,;\, \dot{V}_{o,f} = 0.035\,\mathrm{m}^3/\mathrm{s} \end{aligned} \quad \text{(A.10)}$$

A.1.1.2 Without-Wind Design

To satisfy both the loop equations and the without-wind cross-flow constraint, all four design variables may have to be adjusted. Practically, however, it would be difficult to alter stack duct sizes, although the room outlet to the stack could conceivably be constricted to achieve the design objective.

With this in mind, we will try leaving the central stack duct at the with-wind size (i.e., D_e = 0.152 m) and determine the inlet area required using Equation A.6 (i.e., with ΔP_w set to zero). The result of this step is A_d = 0.045 m^2. From the without-wind cross-flow constraint relation, Equation A.9, we find A_a = 0.034 m^2. To satisfy the without-wind local loop equation using this inlet opening the local stack would have to be reduced to 0.143 m.

Alternatively, we may apply the local loop equation, Equation A.5, to determine the size of the room outlet that would be needed to a) maintain the room inlet at A_a = 0.034 m^2 to satisfy the cross-flow constraint, b) maintain the local stack diameter at 0.233 m for practical reasons, and c) achieve the desired ventilation rate. Recognizing the second term of the local loop equation corresponds to the room outlet, we would thus solve:

$$\frac{0.0008087}{(A_a = 0.034)^2} + \frac{0.0002360}{D_{b,outlet}{}^4} + \frac{7.079 \times 10^{-5}}{(D_b = 0.233)^5} + 0.64 \text{ Pa} = \overbrace{0 \text{ Pa}}^{\Delta P_w} + \overbrace{3.09 \text{ Pa}}^{\Delta P_s} \tag{A.11}$$

to obtain $D_{b,outlet}$ = 0.133 m. In summary the without-wind design is then:

With-Wind Design

$$\begin{aligned} &A_a = 0.034\,\text{m}^2\,;D_{b,outlet} = 0.133\,\text{m}\,;D_{b,stack} = 0.233\,\text{m}\,;\dot{V}_{o,c} = 0.025\,\text{m}^3/\text{s} \\ &A_d = 0.045\,\text{m}^2\,;D_e = 0.152\,\text{m}\,;\dot{V}_{o,f} = 0.035\,\text{m}^3/\text{s} \end{aligned} \tag{A.12}$$

A.1.1.3 Design Example 5 Conclusion

This design example was formulated to demonstrate the design of a ventilation system with multiple loops and to show how an additional design constraint to inhibit cross-flow between these loops might be included in the design process. The process used to find an acceptable solution to the design problem proved to be convoluted and challenging. This resulted from the fact that:

- dual design conditions (i.e., with and without-wind) were considered,
- asymptotic limits of both the loop equations and the cross-flow constraint relation limited component size selection, and
- solution of the resulting nonlinear equations often proved to be difficult.

Furthermore, infiltration was ignored yet it may be expected to significantly impact the design of such a system. Clearly, the design of complex passive ventilation systems will require computationally automation if the value of the proposed methods is to be realized.

While the design results, Equations A.10 and A.12, satisfy the loop equations and cross-flow design constraints it would be practically difficult to manage the adjustment of room inlets and room outlets as wind conditions changed throughout the year. On the other hand, the self-regulating stack terminal devices may provide much of this control. As presently formulated, however, the loop method does not characterize the controllability of the system designed. Thus, the proposed design must be considered a preliminary proposal that will be investigated more thoroughly via annual system simulation.

A.2 Alternative Passive Ventilation Systems

So far, we've considered the application of the loop equation design method to passive ventilation systems with passive stacks. In this section, we will consider applications to two alternative passive ventilation systems – a cross-ventilation system and a balanced-stack system. Methods for the analysis of wind-driven cross ventilation systems are well established and should be familiar. Here, we will extend these methods to account for self-regulation of inlet and outlet airflows and consider the controllability of the system. The design of a balanced-stack system is likely to be unfamiliar yet it will be seen that it is analytically similar to a traditional cross-ventilation system, although it offers the important advantage of being less sensitive to wind direction.

A.2.1 Design Example 6: Sizing Components of a Cross-Ventilation System

Consider the design of the wind-driven cross-ventilation system illustrated in Figure A.4 – a system with self-regulating inlet and outlet vents and transfer grills between rooms to complete the necessary cross-ventilation airflow path. This seemly simple system presents a new analytical challenge – it must be designed for flow reversals that result from changes in wind direction. Consequently, it will be necessary to account for *forward* flow conditions (i.e., as indicated by the arrows in Figure A.4) and *reverse* flow conditions.

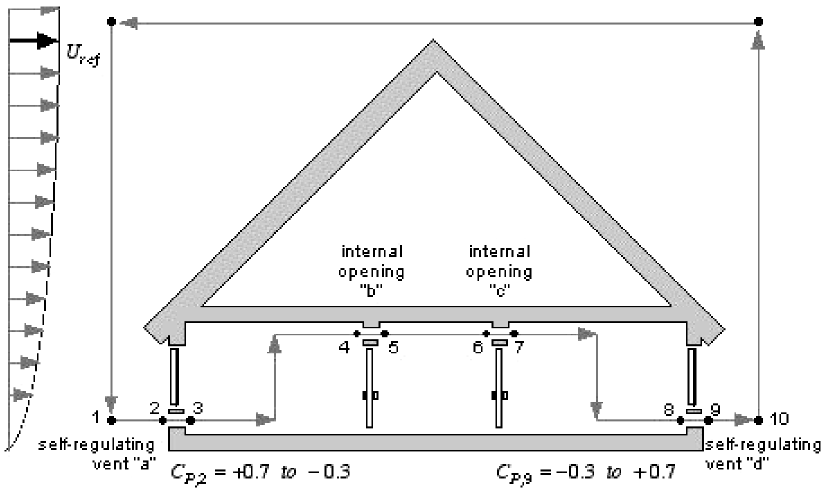

Figure A.4 Wind-driven cross-ventilation system for Design Example 6.

Again, we will assume the system is to be designed to provide a more or less constant ventilation rate of 0.35 ACH for a dwelling with an enclosed volume of 340 m^3. Thus, the cross-ventilation airflow rate will need to be (0.35 h^{-1})(340 m^3)(1 h/3600 s) = 0.033 m^3/s.

A.2.1.1 Wind Pressure Term

As a first step in accounting for flow reversals it is necessary to model wind pressures realistically. Here, the wind pressure coefficients on the two exterior elevations shown will be assumed to vary from +0.7 to –0.3 as described by the Walker/Wilson model illustrated in Figure 5.17. In essence then, the building will be modeled as a unit in a row of attached dwellings with only two exposures. Wind speed and wind direction will be based on the Boston data illustrated in Figures 5.10 and 5.11 and wind speed will be adjusted for a suburban terrain and an unsheltered site as in Design Example 4.

Forward and reverse flow loop equations will be formed subsequently for the pressure loop shown in Figure A.4. With these wind-modeling assumptions made, the wind pressure term ΔP_w of the loop equation can, however, be evaluated directly as:

$$\Delta P_w = +C_{P2}\frac{\rho U_{ref}^{\;2}}{2} - C_{P9}\frac{\rho U_{ref}^{\;2}}{2} = \left(C_{P2}\left(\phi(t)\right) + C_{P9}\left(\phi(t)\right)\right)\frac{\rho(t)\,U_{ref}(t)^2}{2} \qquad \text{(A.13)}$$

where $C_p(\phi(t))$ will be computed using Equation 4.14, the outdoor air density ρ will be taken as a constant of 1.213 kg/m^3 since buoyancy plays no role in this ventilation system, and the reference wind velocity will be computed using the Boston data corrected for a suburban terrain using Equation 5.44. The results of this computational exercise are plotted in Figure A.5.

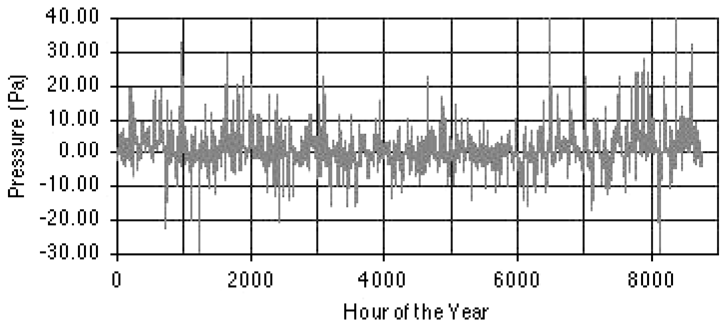

Figure A.5 Time history of the loop equation wind pressure term ΔP_w for the cross-ventilation system of Design Example 6 – Boston data, suburban site.

As might be expected, the driving wind pressure varies wildly from large negative to large positive values – here, within the range of –46 Pa to +32 Pa. A histogram, Figure A.6, of this computed data reveals, however, the driving pressure varies between –6.5 Pa to +10.5 Pa for 90% of the time (i.e., for all but 876 hours of the year) with the mean in the positive range equal to 3.95 Pa and in the negative range –2.71 Pa.

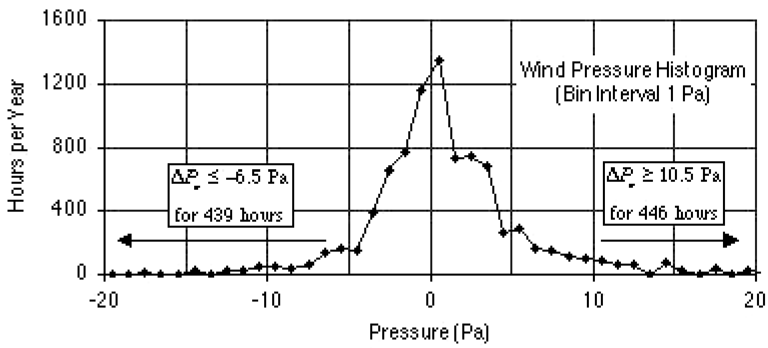

Figure A.6 Histogram of wind pressure data plotted in Figure A.4. (Mean wind pressure difference: positive 3.95 Pa; negative –2.71 Pa.)

A.2.1.2 Self-Regulating Vent Model

Next, strategic decisions regarding the self-regulating vents must be considered. They may, in principle, be placed to limit ventilation rates by throttling either inlet or outlet airflows. To throttle outlet airflows, the left self-regulating vent must be configured to provide relatively little resistance to forward airflow while the right vent must self-regulate to limit flow to the objective 0.033 m^3/s. For reverse flow, the roles of the self-regulating vents must then switch. Existing self-regulating vents have been designed to do just the opposite – they act to block reverse flow and regulate forward flow. Thus to achieve the desired objective we will define the characteristics of a hypothetical ideal vent. While the desired performance could conceivably be achieved using existing technology (e.g., by installing two Dutch devices side-by-side with one positioned in the forward flow and the other in the reverse flow direction), it would be more reasonable to expect the development of a new device for this purpose.

A set of forward and reverse flow models for self-regulating vents was introduced in Chapter 4 that provided value and slope continuity at zero flows. This model may be adapted for our purposes here. With subscript *f* indicating forward flow and *r* reverse flow of the vent itself (i.e., as opposed to the passive ventilation system), a complete self-regulating vent model may be defined as:

Vent Forward Flow $$\Delta P = -\Delta P_{o,f} \ln\left(1 - \frac{\dot{V}}{\dot{V}_{o,f}}\right) \tag{A.14}$$

Vent Reverse Flow $$\Delta P = -\Delta P_{o,r} \ln\left(1 - \frac{\dot{V}}{\dot{V}_{o,r}}\right) \tag{A.15}$$

Slope Continuity Condition $$\frac{\dot{V}_{o,f}}{\Delta P_{o,f}} = \frac{\dot{V}_{o,r}}{\Delta P_{o,r}} \tag{A.16}$$

where, ΔP for both Equation A.14 and A.15 is taken as positive for a pressure drop in the direction of airflow.

A.2.1.3 Loop Equations

To form the loop equations for the cross-ventilation system we must add appropriate models for flow through the two transfer grills at *b* and *c*. These grills are reasonably modeled using the orifice relation with unobstructed areas of A_b and A_c, respectively. With these component models in hand, loop equations for the forward and reverse flow cases may be directly formed as:

Forward Flow

$$-\Delta P_{o,ra}\ln\left(1-\frac{\dot{V}}{\dot{V}_{o,ra}}\right)+\frac{\rho\dot{V}^2}{2C_d^{\,2}A_b^2}+\frac{\rho\dot{V}^2}{2C_d^{\,2}A_c^2}-\Delta P_{o,fd}\ln\left(1-\frac{\dot{V}}{\dot{V}_{o,fd}}\right)=\Delta P_w \tag{A.17}$$

Reverse Flow

$$-\Delta P_{o,fa}\ln\left(1-\frac{\dot{V}}{\dot{V}_{o,fa}}\right)+\frac{\rho\dot{V}^2}{2C_d^{\,2}A_b^2}+\frac{\rho\dot{V}^2}{2C_d^{\,2}A_c^2}-\Delta P_{o,rd}\ln\left(1-\frac{\dot{V}}{\dot{V}_{o,rd}}\right)=\Delta P_w \tag{A.18}$$

where:

- continuity demands that the volumetric flow rates for all components are equal (and will be set to the desired ventilation airflow rate of 0.033 m^3/s):

$$\dot{V}\equiv\dot{V}_a=\dot{V}_b=\dot{V}_c=\dot{V}_d=0.033\,m^3/s \tag{A.19}$$

- the stack term ΔP_s in this case is zero and is not included in the loop equations,
- the wind pressure term ΔP_w in Equation A.17 corresponds to the forward flow wind direction (i.e., as illustrated in Figure A.3) and in Equation A.18 to the reverse flow direction, and
- the self-regulating vent parameters are subject to the slope continuity condition of Equation A.16 as:

$$\frac{\dot{V}_{o,fa}}{\Delta P_{o,fa}}=\frac{\dot{V}_{o,ra}}{\Delta P_{o,ra}}\ \ and\ \ \frac{\dot{V}_{o,fd}}{\Delta P_{o,fd}}=\frac{\dot{V}_{o,rd}}{\Delta P_{o,rd}} \tag{A.20}$$

A.2.1.4 Sizing System Components

The selection of system components using Equations A.17, A.18, A.19, and A.20 alone would be challenging, but practical considerations relating to the use of self-regulating vents greatly simplifies the process. Anticipating reductions in flow due to other resistance along the flow path, we would reasonably select a self-regulating (outlet) vent with a nominal forward flow rate $\dot{V}_{o,f}$ slightly greater than the desired ventilation flow rate of 0.033 m^3/s. Here, we will select $\dot{V}_{o,f2}=\dot{V}_{o,f9}=0.035\,m^3/s$ using the Dutch self-regulating vent technology with $\Delta P_{o,f2}=\Delta P_{o,f9}=0.30\,Pa$. To insure that the outlet vent throttles the system airflows, the nominal reverse flow rate $\dot{V}_{o,r}$ would be set to a larger value to minimize its resistance. Here, we will select $\dot{V}_{o,r2}=\dot{V}_{o,r9}=0.040\,m^3/s$. Finally, to satisfy the slope continuity condition of Equation A.20 the threshold pressure drop for the reverse flow regulation will need to be

$\Delta P_{o,r2} = \Delta P_{o,r9} = 0.34\ Pa$. The pressure flow characteristics of the selected self-regulating vent is plotted in Figure A.7.

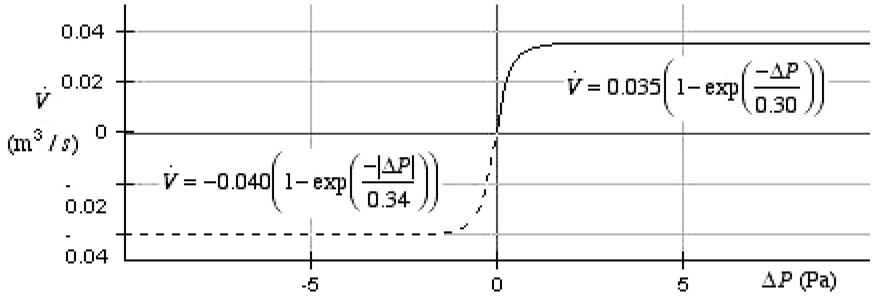

Figure A.7 Pressure flow characteristics of the self-regulating vent selected for the cross-ventilation Design Example 6.

Given the asymmetry of the driving wind pressure histogram, Figure A.5, one could argue that the self-regulating vents for component *a* and *d* could be different. The ability of these vents to self-regulate may be expected, however, to accommodate this asymmetry.

Finally, numerical values may be substituted into the system loop equations leaving only the unobstructed areas of the transfer grills to be determined A_b and A_c. In this case we will leave the design conditions, ΔP_w, as a variable and, again, use a nominal air density of 1.213 kg/m³. For these assumptions the forward- and reverse-flow loop equations are identical, consequently it is reasonable to size the two transfer grills to be of equal size $A_b = A_c \equiv A$:

Forward– and Reverse– Flow

$$-0.34\ln\left(1-\frac{0.033}{0.040}\right)+\frac{(1.213)(0.033)^2}{2(0.6)^2 A^2}+\frac{(1.213)(0.033)^2}{2(0.6)^2 A^2}-0.30\ln\left(1-\frac{0.033}{0.035}\right)=\Delta P_w \tag{A.21}$$

Equation A.21 may be solved for the transfer grill unobstructed area to obtain:

$$A=\sqrt{\frac{0.00367}{\Delta P_w+0.34\ln(0.175)+0.30\ln(0.057)}}\ m^2 \tag{A.22}$$

This relation, plotted in Figure A.8, reveals that the transfer grill area *A* should, ideally, vary with driving wind pressure. As this would be practically difficult to achieve we will compromise and set $A = 0.10\ m^2$ – a value that is practically reasonable and yet appropriate for a driving pressure just larger than the asymptotic limit of Equation A.22 at $\Delta P_w = 1.45\ Pa$. That is to say, transfer grills with this unobstructed area will act to limit ventilation rates only when the driving wind pressures drop below 1.45 Pa.

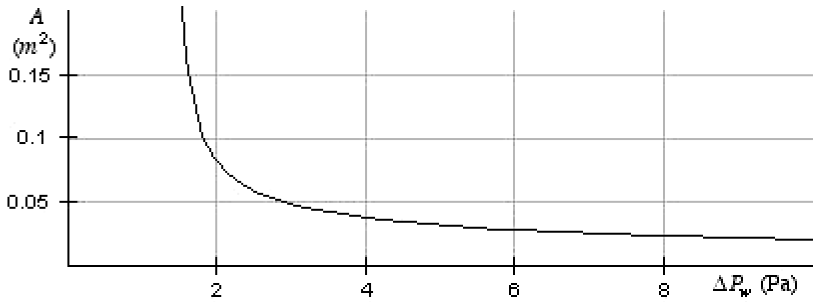

Figure A.8 Required transfer grill areas for the likely range of driving wind pressure differences.

Based on this analysis, then, the cross-ventilation system design is defined in terms of the following component sizes:

$$\dot{V}_{o,f} = 0.035\, m^3/s\,; \dot{V}_{o,r} = 0.040\, m^3/s$$
$$\Delta P_{o,f} = 0.30\, Pa\,; \Delta P_{o,r} = 0.34\, Pa$$
$$A_b = A_c = 0.10\, m^2$$

A.2.1.5 Control Characteristics of the Proposed System

It is quite natural to ask: "How well will the proposed system control ventilation rates to the desired 0.033 m^3/s?" The component sizes may be directly substituted into the loop equation to form an equation that may be used to predict ventilation airflow rates $\dot{V}$ from the computed wind driven pressures plotted in Figure A.5 as:

$$-0.34\ln\left(1-\frac{\dot{V}}{0.040}\right)+\frac{(1.213)\dot{V}^2}{2(0.6)^2(0.10)^2}+\frac{(1.213)\dot{V}^2}{2(0.6)^2(0.10)^2}-0.30\ln\left(1-\frac{\dot{V}}{0.035}\right)=\Delta P_w \tag{A.23}$$

This exercise was completed using numerically approximate solutions to Equation A.23. The results are plotted in Figure A.9.

This passive ventilation system is seen to be able to regulate ventilation airflow rates to within 10% of the objective rate of 0.033 m^3/s for over 6,200 hours or 71% of the year. Furthermore, while under-ventilation can not be entirely avoided, over-ventilation and the energy penalty associated with it, can – i.e., the self-regulating vents, as designed, limit ventilation rates to 0.035 m^3/s.

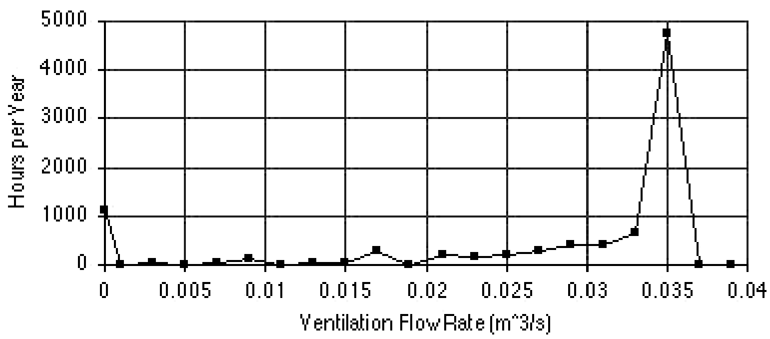

Figure A.9 Histogram of ventilation flow rates for the self-regulating cross-ventilation system of Design Example 6 – Boston Data, suburban site.

A.2.1.6 Conventional System Comparison

To place these results in perspective it is useful to consider the performance of a conventional cross-ventilation system employing fixed area trickle vents and, again, transfer grills. For such a system we may reasonably model the trickle vents using the orifice component model with unobstructed areas of A_a and A_d for the vents at locations *a* and *d* respectively and A_b and A_c for the transfer grills. In this case, the forward and reverse flow loop equations are identical and take the following form:

$$\frac{\rho \dot{V}^2}{2C_d^{\ 2} A_a^2} + \frac{\rho \dot{V}^2}{2C_d^{\ 2} A_b^2} + \frac{\rho \dot{V}^2}{2C_d^{\ 2} A_c^2} + \frac{\rho \dot{V}^2}{2C_d^{\ 2} A_d^2} = \left|\Delta P_w\right| \tag{A.24}$$

where the absolute value of the pressure term is used to generalize the equation for both reverse and forward flow directions.

If we choose to configure the system symmetrically (i.e., with identical trickle vents $A_v \equiv A_a = A_d$ and transfer grills $A_g \equiv A_b = A_c$) and, again employ an air density of 1.213 kg/m^3, a discharge coefficient of C_d = 0.60, and set the ventilation airflow rate to the desired value of 0.033 m^3/s the system loop equation simplifies to:

$$\frac{0.00367}{A_v^2} + \frac{0.00367}{A_g^2} = \left|\Delta P_w\right| \tag{A.25}$$

Sizing the system components for the wildly varying driving wind pressures to be expected is problematic for this conventional system. Here we will size system components for the annual mean wind pressure condition which, for the Boston data shown in Figure A.5, is 3.41 Pa. Substituting this value into the right hand side of the system loop equation, Equation A.25, we obtain a design relation that establishes feasible combinations of trickle vent and transfer grill areas that will achieve the 0.033 m^3/s ventilation flow rate objective for the design wind pressure difference of 3.41 Pa. This relation is plotted below in Figure A.10.

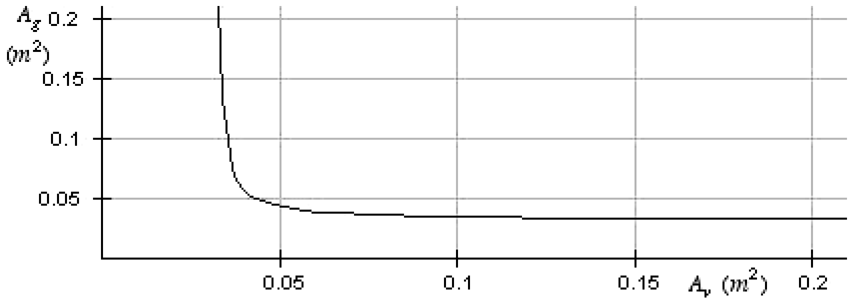

Figure A.10 Feasible design curve for the conventional cross-ventilation system of Design Example 6.

Again, the feasible design relation is limited by asymptotes (here, $A_g = 0.0328$ m^2 and $A_v = 0.0328$ m^2) that define lower limits on the size of system components. With these limits in mind, we can proceed to specify system components sizes. Following the example of the self-regulating system above, we will set the unobstructed areas of the transfer grills to 0.10 m^2 and substitute this value into Equation A.25 to establish sizes for the trickle vents. If this is done, we will find we need trickle vents with unobstructed areas of 0.0347 m^2.

To evaluate the performance of this proposed design we may simply substitute these component sizes into the system loop equation, Equation A.24, to obtain:

$$3131.3\dot{V}^2 = |\Delta P_w| \ \ or \ \ \dot{V} = \sqrt{\frac{|\Delta P_w|}{3131.3}} \tag{A.26}$$

Using this relation and the wind pressure data plotted in Figure A.5, cross-ventilation flow rates may be determined for each hour of the annual record. A histogram of the computed results is compared to that obtained for the self-regulating system in Figure A.11.

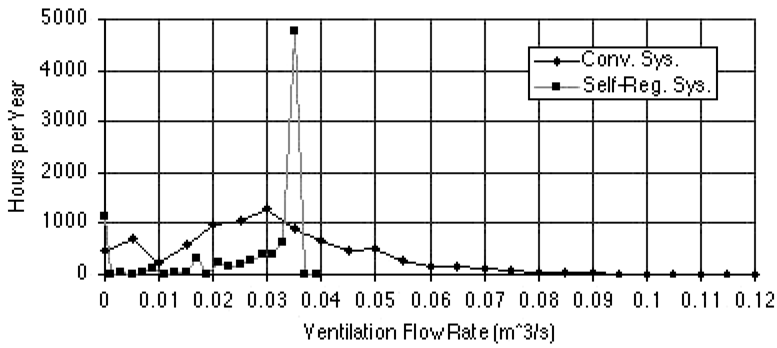

Figure A.11 Comparison of ventilation flow rate histograms for the conventional and self-regulating cross-ventilation systems of Design Example 6 – Boston Data, suburban site.

The advantages of the self-regulating system are evident from this comparison. While both systems provide nearly equal annual mean ventilation rates – 0.0270 m^3/s for the self-regulated system and 0.0276 m^3/s for the conventional system – the conventional system suffers far more hours of both under- and over-ventilation. Furthermore, the conventional system maintains ventilation flow rates within 10% of the intended rate of 0.033 m^3/s for only 33% of the year while, as noted above, the self-regulated system does so for 71% of the year.

A general principle may be inferred from these computed results:

> *Passive Design Strategy*
>
> *Self-regulating vents act to limit over-ventilation by simply throttling airflows when driving pressures exceed the threshold pressure for which they are designed. They also act to limit the number of hours during which under-ventilation occurs if the threshold pressure is sufficiently low in comparison to the expected driving pressures the system will experience. For the examples considered, a self-regulating vent threshold pressure less than 0.50 Pa may be expected to achieve both objectives.*

A.2.2 Design Example 7: Sizing Components of a Balanced-Stack System

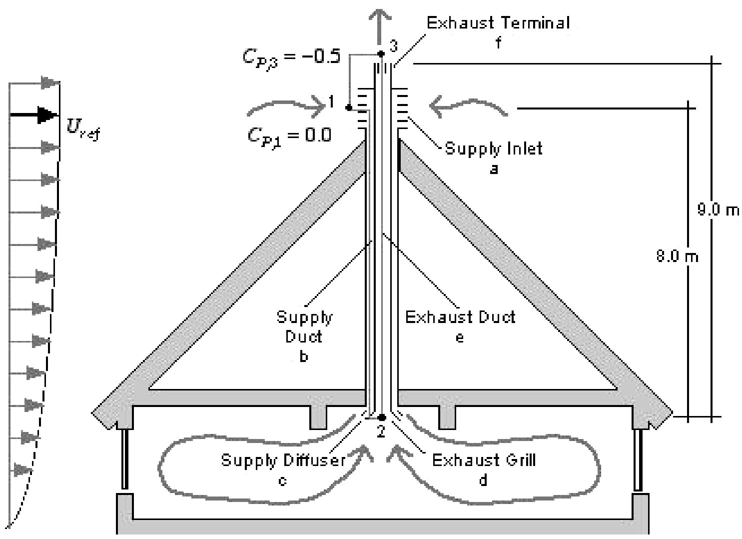

Figure A.12 Wind-driven balanced stack ventilation system for Design Example 7.

Balanced-stack systems offer an alternative to cross-ventilation systems that use wind-driven pressures to provide ventilation for air quality control. They offer the distinct advantage that they can be more readily configured to be insensitive to wind direction. In this design example, we will consider the design of the balanced-stack system illustrated in Figure A.12 that is configured with co-axial supply and exhaust ducts – a system that is commercially available in northern Europe. Other configurations with separate supply and exhaust duct locations are also possible.

In this co-axial system, supply air flows radially into the supply duct through a perimeter inlet, moves downward in the annular space provided, and is injected into the building by an annular diffuser. Room air is exhausted via the central duct vertically through the exhaust terminal device to complete the airflow path. Wind airflows act to create a negative pressure at the exhaust terminal device – here illustrated as a square-cut duct with a grill to inhibit animal entry – and, due to the annular form of the supply inlet, a neutral pressure at the supply inlet. Under ideal circumstances, the supply airflow rate will equal the exhaust airflow rate – thus this system is identified as a *balanced* system.

A.2.2.1 Loop Equations

Following the flow loop illustrated in Figure A.12, the ventilation air encounters the flow resistances of the supply inlet, supply duct, supply diffuser, exhaust grill, exhaust duct, and the exhaust terminal device. Loop equations may then be formed using a) the duct fitting component relation for the supply inlet, ceiling diffuser, and ceiling exhaust grill and b) the duct component for the supply and exhaust ducts. The exhaust terminal device may be modeled using either the duct fitting component relation for an unregulated system, or the self-regulating vent component relation for a regulated system. Under ideal circumstances the supply airflow rate will equal the exhaust rate thus the airflow rate through all components will be equal – here identified simply as $\dot{V}$.

Unregulated System

$$\overbrace{\frac{\rho C_a \dot{V}^2}{2A_a^2}}^{\text{supply inlet}} + \overbrace{\frac{fL_b}{2}\frac{\rho \dot{V}^2}{D_{hb}A_b^2}}^{\text{supply duct}} + \overbrace{\frac{\rho C_c \dot{V}^2}{2A_c^2}}^{\text{supply diffuser}} + \overbrace{\frac{\rho C_d \dot{V}^2}{2A_d^2}}^{\text{exhaust grill}} + \overbrace{\frac{fL_e}{2}\frac{\rho \dot{V}^2}{D_{he}A_e^2}}^{\text{exhaust duct}} + \overbrace{\frac{\rho C_f \dot{V}^2}{2A_f^2}}^{\text{exhaust terminal}} = \Delta P_s + \Delta P_w \quad \text{(A.27)}$$

Regulated System

$$\overbrace{\frac{\rho C_a \dot{V}^2}{2A_a^2}}^{\text{supply inlet}} + \overbrace{\frac{fL_b}{2}\frac{\rho \dot{V}^2}{D_{hb}A_b^2}}^{\text{supply duct}} + \overbrace{\frac{\rho C_c \dot{V}^2}{2A_c^2}}^{\text{supply diffuser}} + \overbrace{\frac{\rho C_d \dot{V}^2}{2A_d^2}}^{\text{exhaust grill}} + \overbrace{\frac{fL_e}{2}\frac{\rho \dot{V}^2}{D_{he}A_e^2}}^{\text{exhaust duct}} \overbrace{- \Delta P_o \ln\left(1 - \frac{\dot{V}}{\dot{V}_o}\right)}^{\text{exhaust terminal}} = \Delta P_s + \Delta P_w$$

(A.28)

where a nominal value of air density ρ is used for all components.

For the purposes of preliminary calculations the following values for component parameters will be assumed:

- air density: ρ = 1.213 kg/m^3
- supply inlet loss coefficient: C_a = 0.25
- supply duct: friction factor f = 0.03; length L_b = 8.0 m
- supply diffuser loss coefficient: C_c = 0.30
- exhaust grill loss coefficient: C_d = 0.50
- exhaust duct: friction factor f = 0.03; length L_b = 9.0 m
- exhaust terminal:
- unregulated: loss coefficient: C_d = 0.50
- regulated: threshold pressure difference: ΔP_o = 0.30 Pa

The loss coefficients used are based on those published in the ASHRAE Handbook of Fundamentals (ASHRAE 1997). As the design becomes more specific, these values could be revised to reflect as-built conditions. The threshold pressure used for the self-regulated terminal device is based on that achieved in the Dutch vent discussed earlier.

Again, we will assume the system is to be designed to provide a more or less constant ventilation rate of 0.35 ACH for a dwelling with an enclosed volume of 340 m^3. Thus, the ventilation airflow rate will need to be $\dot{V}$ = (0.35 h^{-1})(340 m^3)(1 h/3600 s) = 0.033 m^3/s. With this ventilation objective and the component model parameters set, the formal loop equations presented above, Equations A.27 and A.28, may be numerically evaluated:

Unregulated System

$$\overbrace{\frac{0.000165}{2A_a^2}}^{\text{supply inlet}}+\overbrace{\frac{0.00159}{D_{hb}A_b^2}}^{\text{supply duct}}+\overbrace{\frac{0.000198}{2A_c^2}}^{\text{supply diffuser}}+\overbrace{\frac{0.000330}{2A_d^2}}^{\text{exhaust grill}}+\overbrace{\frac{0.00178}{D_{he}A_e^2}}^{\text{exhaust duct}}+\overbrace{\frac{0.000330}{2A_f^2}}^{\text{exhaust terminal}}=\Delta P_s+\Delta P_w \tag{A.29}$$

Regulated System

$$\overbrace{\frac{0.000165}{2A_a^2}}^{\text{supply inlet}}+\overbrace{\frac{0.00159}{D_{hb}A_b^2}}^{\text{supply duct}}+\overbrace{\frac{0.000198}{2A_c^2}}^{\text{supply diffuser}}+\overbrace{\frac{0.000330}{2A_d^2}}^{\text{exhaust grill}}+\overbrace{\frac{0.00178}{D_{he}A_e^2}}^{\text{exhaust duct}}\overbrace{-0.30\ln\left(1-\frac{0.033}{\dot{V}_o}\right)}^{\text{exhaust terminal}}=\Delta P_s+\Delta P_w \tag{A.30}$$

A.2.2.2 Isothermal Wind-Driven Flow

To gain insight into the behavior of this balanced stack system we will first consider sizing system components for hypothetical isothermal conditions using the Boston wind speed record adjusted for a suburban site from Design Example 4.

For the given stack configuration, winds will act to create a range of pressures from negative to positive around the annular supply inlet. The free flow of air from position to position immediately within the supply inlet grill will tend to equalize these pressure differences so that, on average, the wind pressure acting at the top of the supply duct will be essentially neutral and, importantly, insensitive to wind direction. For these reasons we will assume an effective wind pressure coefficient at the supply inlet of C_{P1} = 0.0. The vertically aligned, square-cut exhaust duct – if extended above nearby structures – will also be insensitive to wind direction. The effective wind pressure coefficient for this type of stack terminal has been determined to be approximately C_{P3} = -0.50 (Welsh 1994; Welsh 1995; Welsh 1995; Welsh 1995). Consequently, the wind pressure term of the loop equations may be evaluated as:

$$\Delta P_w=\left(C_{P1}-C_{P3}\right)\frac{\rho U_{ref}^2}{2}=(0.50)\frac{\rho U_{ref}^2}{2} \tag{A.31}$$

In some (unusual) circumstances, the prevailing wind direction may be relatively constant and thus the designer may seek to develop a stack designed specifically for the dominant wind direction. In such a circumstance, the designer may conceivably realize much larger driving pressure differences for a given wind speed – i.e., achieve a larger ΔC_P – that would allow system components to be sized somewhat smaller for the same ventilation rate objective. The design configuration being considered here accepts, however, lower driving wind pressures – i.e., smaller ΔC_P's – for improved insensitivity to wind direction and thus greater control of ventilation airflow rates. For most sites, it is likely that this strategy will result in the best system performance.

Passive Design Strategy

Directionally insensitive stack designs that necessarily compromise driving wind pressure differences may be expected to offer greater control over ventilation flow rates and thus improved system performance.

To proceed, the reference wind velocity may be computed using the Boston data from Design Example 4 (corrected for a suburban terrain) and the driving wind pressure time history directly evaluated using Equation A.31. The results of this exercise are plotted below in Figure A.13.

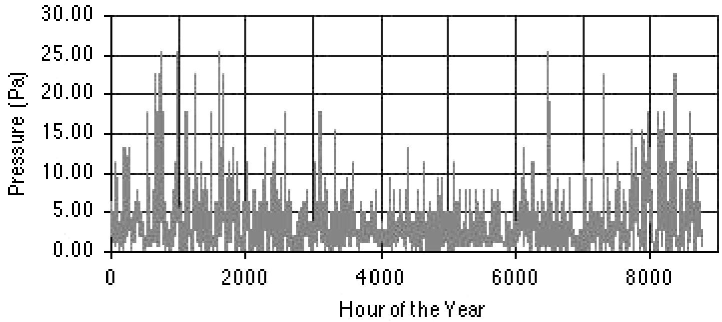

Figure A.13 Time history of the wind pressure term ΔP_w for the balanced stack ventilation system of Design Example 7 – Boston data, suburban site.

Again, the driving wind pressure varies wildly. However, a histogram, Figure A.14, of this computed data reveals that the driving pressure varies between approximately 0.5 Pa to 7.0 Pa for 90% of the time (i.e., for all but 876 hours of the year) with the mean equal to 3.31 Pa.

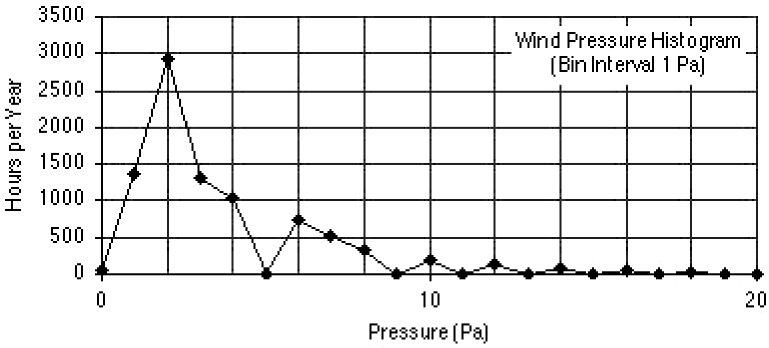

Figure A.14 Histogram of wind pressure data plotted in Figure A.13. (Mean wind pressure difference 3.31 Pa.)

It may be most reasonable to size components of the unregulated balanced stack system based on the annual mean driving wind pressure ΔP_w = 3.31 Pa. The regulated system could be designed for a lower design pressure, since it will limit flows for higher wind pressures, or if some under ventilation can be tolerated this system could also be designed for the annual mean condition. The former choice will result in slightly larger system components than the latter. For both strategies, it would be reasonable to set the nominal flow rate of the self-regulating exhaust terminal device to a value slightly greater than the design ventilation flow rate (i.e., 0.033 m^3/s). Here, we will set this value to $\dot{V}_o$ = 0.035 m^3/s. Hence we obtain numerically evaluated loop equations for the two systems being considered:

Unregulated System

$$\overbrace{\frac{0.000165}{2A_a^2}}^{\text{supply inlet}} + \overbrace{\frac{0.00159}{D_{hb}A_b^2}}^{\text{supply duct}} + \overbrace{\frac{0.000198}{2A_c^2}}^{\text{supply diffuser}} + \overbrace{\frac{0.000330}{2A_d^2}}^{\text{exhaust grill}} + \overbrace{\frac{0.00178}{D_{he}A_e^2}}^{\text{exhaust duct}} + \overbrace{\frac{0.000330}{2A_f^2}}^{\text{exhaust terminal}} = 3.31 \text{ Pa} \quad (A.32)$$

Regulated System

$$\overbrace{\frac{0.000165}{2A_a^2}}^{\text{supply inlet}} + \overbrace{\frac{0.00159}{D_{hb}A_b^2}}^{\text{supply duct}} + \overbrace{\frac{0.000198}{2A_c^2}}^{\text{supply diffuser}} + \overbrace{\frac{0.000330}{2A_d^2}}^{\text{exhaust grill}} + \overbrace{\frac{0.00178}{D_{he}A_e^2}}^{\text{exhaust duct}} \overbrace{- 0.30\ln\left(1-\frac{0.033}{0.035}\right)}^{\text{exhaust terminal}} = 3.31 \text{ Pa}$$

(A.33)

Following the procedures that are now familiar, the limiting values of each of the design parameters for each of the two system configurations may be directly evaluated to obtain:

Table A.1 Minimum values of system design parameters to achieve a ventilation airflow rate of 0.033 m^3/s for a 3.31 Pa driving wind pressure.

	Supply Inlet A_a (m^2)	Supply Duct $D_{hb}A_b^2$ (m^5)	Supply Diffuser A_c (m^2)	Exhaust Grill A_d (m^2)	Exhaust Duct $D_{he}A_e^2$ (m^5)	Exhaust Terminal A_e (m^2)
Unregulated System	0.00706	0.000479	0.00774	0.00999	0.000539	0.00999
Regulated System	0.00821	0.000646	0.00899	0.01161	0.000727	N.A.

At this point a designer would reasonably impose practical constraints on the selection of system components. For the co-axial stack configuration, here it would be convenient to use standard duct sizes. The geometric properties of a co-axial circular stack with inner radius R_i and outer radius R_o are:

$$A_b = \pi(R_o^2 - R_i^2)\,;\, D_{hb} = \frac{4\pi(R_o^2 - R_i^2)}{2\pi(R_o + R_i)}\,;\, A_e = \pi R_i^2\,;\, D_{he} = 2R_i$$

Using these geometric relations, a table of section properties for candidate configuration may be formed. Such a table is presented in Table A.2 for 0.05 m (2 inch) size increments.

Comparing the section properties of these candidate duct configurations to the minimum requirements of Table A.1 it is clear that the compound duct parameter D_hA^2 will control design. The third configuration with R_I = 0.15 m (6 in) and R_o = 0.25 m (10 in) exceeds these minimum requirements for both the regulated and unregulated systems. Thus, this configuration is feasible. (It is interesting to note that one commercially available balanced stack system is available in three sizes with R_o = 0.20 m or 0.25 m or 0.30 m – Configuration 3 corresponds to the mid size alternative.)

Table A.2 Section properties of candidate balanced-stack configurations. (R_I is the inner duct radius and R_o is the outer duct radius.)

	Supply Duct A_b (m^2)	Supply Duct $D_{hb}A_b^2$ (m^5)	Exhaust Duct A_e (m^2)	Exhaust Duct $D_{he}A_e^2$ (m^5)
Configuration 1 R_I = 0.10 m R_o = 0.15 m	0.0393	0.000154	0.0314	0.000197
Configuration 2 R_I = 0.15 m R_o = 0.20 m	0.0550	0.000302	0.0707	0.00150
Configuration 3 R_I = 0.15 m R_o = 0.25 m	0.126	0.00315	0.0707	0.00150

Accepting Configuration 3, the values for the compound duct parameter D_hA^2 may be substituted into the loop equations to more narrowly define limiting values for the other design parameters as:

Unregulated System

$$\overbrace{\frac{0.000165}{2A_a^2}}^{\text{supply inlet}}+\overbrace{\frac{0.00159}{2(0.00315)}}^{\text{supply duct}}+\overbrace{\frac{0.000198}{2A_c^2}}^{\text{supply diffuser}}+\overbrace{\frac{0.000330}{2A_d^2}}^{\text{exhaust grill}}+\overbrace{\frac{0.00178}{2(0.00150)}}^{\text{exhaust duct}}+\overbrace{\frac{0.000330}{2A_f^2}}^{\text{exhaust terminal}}=3.31\text{ Pa} \quad \text{(A.34)}$$

Regulated System

$$\overbrace{\frac{0.000165}{2A_a^2}}^{\text{supply inlet}}+\overbrace{\frac{0.00159}{2(0.00315)}}^{\text{supply duct}}+\overbrace{\frac{0.000198}{2A_c^2}}^{\text{supply diffuser}}+\overbrace{\frac{0.000330}{2A_d^2}}^{\text{exhaust grill}}+\overbrace{\frac{0.00178}{2(0.00150)}}^{\text{exhaust duct}}\overbrace{-0.30\ln\left(1-\frac{0.033}{0.035}\right)}^{\text{exhaust terminal}}=3.31\text{ Pa}$$

(A.35)

These loop equations may be employed to refine the estimates of minimum sizes for the remaining component design parameters by systematically evaluating the limiting cases again. The results of this exercise are tabulated below in Table A.3.

Table A.3 Minimum values of system design parameters to achieve a ventilation airflow rate of 0.033 m^3/s for a 3.31 Pa driving wind pressure for a coaxial stack of R_I = 0.15 m (6 in) and R_o = 0.25 m (10 in). Values in parenthesis equal fraction of available duct cross section.

	Supply Inlet A_a (m^2)	Supply Diffuser A_c (m^2)	Exhaust Grill A_d (m^2)	Exhaust Terminal A_e (m^2)
Unregulated System	0.0101 (8%)	0.0111 (9%)	0.0143 (20%)	0.0143 (20%)
Regulated System	0.0147 (12%)	0.0162 (13%)	0.0209 (30%)	N.A.

With these minimum values as a guide and the loop equations in hand, final sizing may be then determined. For example, if we set the supply inlet and supply diffuser unobstructed areas to 0.025 m^2 or 20% of the available cross-sectional area of the supply duct and, for the regulated system, set the exhaust grill and terminal areas equal, then we can use the loop equations to solve for the exhaust grill and terminal areas A.

Unregulated System

$$\overbrace{\frac{0.000165}{2(0.025)^2}}^{\text{supply inlet}}+\overbrace{\frac{0.00159}{0.00315}}^{\text{supply duct}}+\overbrace{\frac{0.000198}{2(0.025)^2}}^{\text{supply diffuser}}+\overbrace{\frac{0.000330}{2A^2}}^{\text{exhaust grill}}+\overbrace{\frac{0.00178}{0.00150}}^{\text{exhaust duct}}+\overbrace{\frac{0.000330}{2A^2}}^{\text{exhaust terminal}}=3.31\text{ Pa} \quad \text{(A.36)}$$

Regulated System

$$\overbrace{\frac{0.000165}{2(0.025)^2}}^{\text{supply inlet}}+\overbrace{\frac{0.00159}{0.00315}}^{\text{supply duct}}+\overbrace{\frac{0.000198}{2(0.025)^2}}^{\text{supply diffuser}}+\overbrace{\frac{0.000330}{2A^2}}^{\text{exhaust grill}}+\overbrace{\frac{0.00178}{0.00150}}^{\text{exhaust duct}}\overbrace{-0.30\ln\left(1-\frac{0.033}{0.035}\right)}^{\text{exhaust terminal}}=3.31\text{ Pa} \quad \text{(A.37)}$$

Solving each of these equations for the single unknown yields a feasible design. The final design specification is tabulated in Table A.4.

Table A.4 Final system design parameters to achieve a ventilation airflow rate of 0.033 m^3/s for a 3.31 Pa driving wind pressure. Values in parenthesis equal fraction of available duct cross section.

	Outer Duct Radius R_o (m)	Inner Duct Radius R_i (m)	Supply Inlet A_a (m^2)	Supply Diffuser A_c (m^2)	Exhaust Grill A_d (m^2)	Exhaust Terminal
Unregulated System	0.25	0.15	0.025 (20%)	0.025 (20%)	0.0252 (36%)	A_e= 0.0305 (m^2) (43%)
Regulated System	0.25	0.15	0.025 (20%)	0.025 (20%)	0.0431 (61%)	$\dot{V}_o$=0.035 (m^3/s)

It must be emphasized that this final design specification is, in principle, just one of many possibilities. The number of practical possibilities – i.e., here limited to duct radii of 0.05 m increments – is however far more limited. With this preliminary design in hand, the supply inlet, supply diffuser, exhaust grill, and exhaust terminal device may be detailed and fabrication drawings prepared.

A.2.2.3 Performance of the Proposed Systems

The regulated and unregulated systems proposed above were designed to provide a ventilation airflow rate of 0.033 m^3/s (0.35 ACH) for the annual mean driving wind pressure difference of 3.31 Pa. These design conditions were selected with the hope that the systems would, minimally, provide an annual average ventilation flow rate close to the objective of 0.033 m^3/s and, ideally, maintain relatively constant ventilation rates close to this value. The actual detailed hour-by-hour performance of the proposed systems may be evaluated, as before, using the loop equations analytically to evaluate the success of the designs.

To proceed, the design parameters of Table A.4 and the component parameters enumerated earlier are substituted into the loop equations, Equations A.27 and A.28, to yield:

Unregulated System

$$\overbrace{242.6\dot{V}^2}^{\text{supply inlet}} + \overbrace{46.2\dot{V}^2}^{\text{supply duct}} + \overbrace{485.2\dot{V}^2}^{\text{supply diffuser}} + \overbrace{477.5\dot{V}^2}^{\text{exhaust grill}} + \overbrace{109.2\dot{V}^2}^{\text{exhaust duct}} + \overbrace{326.0\dot{V}^2}^{\text{exhaust terminal}} = \Delta P_w(t) \qquad \text{(A.38a)}$$

or,

$$1687.7\dot{V}^2 = \Delta P_w(t) \qquad \text{(A.38b)}$$

Regulated System

$$\underbrace{242.6\dot{V}^2}_{\text{supply inlet}} + \underbrace{46.2\dot{V}^2}_{\text{supply duct}} + \underbrace{485.2\dot{V}^2}_{\text{supply diffuser}} + \underbrace{326.0\dot{V}^2}_{\text{exhaust grill}} + \underbrace{109.2\dot{V}^2}_{\text{exhaust duct}} \underbrace{-0.30\ln\left(1-\frac{\dot{V}}{0.035}\right)}_{\text{exhaust terminal}} = \Delta P_w(t)$$

(A.39a)

or,

$$1209.2\dot{V}^2 - 0.30\ln\left(1-\frac{\dot{V}}{0.035}\right) = \Delta P_w(t) \tag{A.39b}$$

Using these relations and the wind pressure data plotted in Figure A.13, balanced-stack ventilation flow rates may be determined for each hour of the annual record. Histograms of the computed results are compared in Figure A.15.

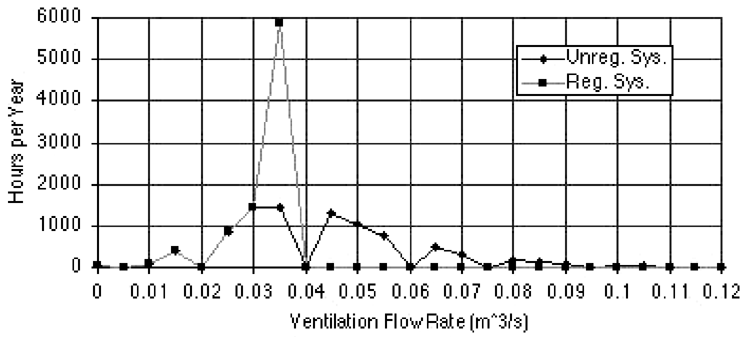

Figure A.15 Comparison of ventilation flow rate histograms for the unregulated and regulated balanced-stack ventilation systems of Design Example 7 – Boston Data, suburban site.

An analysis of the computed results reveals that the annual mean ventilation rate provided by unregulated system is 0.040 m^3/s while that of the regulated system is 0.030 m^3/s – the former over-ventilating by 21% while the latter under-ventilating by 9%, on average. The comparison of histograms shows that the regulated system performs better overall than the unregulated system, effectively inhibiting over-ventilation throughout the year. Nevertheless, both systems suffer under-ventilation although the number of hours of significant under-ventilation is relatively small.

Based on the performance results, fine-tuning of the system design appears to be warranted. The mean performance of the regulated system could be improved if a self-regulating terminal device could be developed with a lower threshold pressure ΔP_o. Alternatively, but less desirably, the nominal volumetric flow rate of the self-regulating outlet device could be increased slightly to edge up the mean ventilation rate and other component sizes adjusted accordingly. To gain more precision, component parameters (e.g., fitting loss coefficients and the duct friction factors) can be revised to reflect likely as-built and flow intensity conditions and the loop equations reformed.

Clearly, the effort to effect this fine-tuning would be significant, yet buoyancy effects and infiltration have not been accounted for. Consequently, this effort would be misdirected. Infiltration could be accounted for following the iterative procedure presented in Design Example 3. Buoyancy effects will be considered briefly in the next section. While computations for the added complexity could conceivably be done manually, the full design task would benefit from the digital implementation of methods presented.

A.2.2.4 Buoyancy Effects

The stack-driven pressure term ΔP_s was assumed zero in the isothermal case considered above yet in reality buoyancy effects can play a role in the balanced-stack ventilation system. Three hypothetical limiting cases can be considered:

- a well-insulated stack with the supply air at the outdoor air temperature and the exhaust air at the (well-mixed) indoor air temperature,
- an ideal heat recovery stack with the supply air temperature varying from the outdoor air temperature at the supply inlet to the indoor air temperature at the supply diffuser and the exhaust air temperature varying from the indoor air temperature at the exhaust grill to the outdoor air temperature at the exhaust terminal, and
- an uninsulated stack placed within the conditioned space so that the supply air temperature rapidly approaches indoor air temperature.

The stack-driven pressure term ΔP_s may be formed for each of these three limiting cases by summing the hydrostatic contributions $\rho g \Delta z$ as the pressure loop is circuited. If this is done, one will discover that for Case 2 ΔP_s is identically zero, for Case 3 ΔP_s will be negligibly small, and only for Case 1 will ΔP_s be significant. The possibility of heat recovery is an important energy-conserving advantage offered by the balanced-stack system. Consequently, the designer should strive to design the balanced stack system to maximize its heat recovery potential. If this goal is achieved then the isothermal design and analysis presented above should suffice.

Nevertheless, it is useful to consider Case 1 to gain some sense of the quantitative importance of the stack-driven pressure term ΔP_s. For the geometry illustrated in Figure A.12, ΔP_s for Case 1 is simply:

$$\Delta P_s = \rho_o g \Delta z_{3.2} - \rho_i g \Delta z_{2,3} = (9.8\,\mathrm{m/s^2})(9\,\mathrm{m})\left(\rho_o - \rho_i\right)$$

Outdoor air temperatures for Boston vary from a low of – 13 °C to a high of 36 °C. If indoor air temperatures are maintained at 20 °C and 25 °C during the heating and cooling seasons respectively, the stack-driven pressure term will be – 3.7 Pa during the peak summer condition and + 13.5 Pa during the minimum winter condition. That is to say that buoyancy effects can not only be significant but can act, in the summer season, in opposition to the wind-driven pressure difference (i.e., if indoor air in the dwelling is cooled mechanically). Again, it must be emphasized that the heat-recovery potential of the balanced stack system should be used, in which case the buoyancy effects would become negligible.

A.3 Mechanically Assisted Passive Ventilation

Through proper sizing and, especially, the use of self-regulating vents passive ventilation systems may be designed to effectively eliminate the possibility of over-ventilation. When the driving stack and wind pressures drop to low values, however, these system can no longer provide the ventilation intended. While this problem may occur infrequently, it is a shortcoming intrinsic to passive ventilation systems. Mechanical assistance offers a direct solution to this problem that may be expected to carry a small energy penalty since its use will be infrequent if the mechanically assisted passive ventilation system is properly designed.

In this section, we will consider adding an axial fan to the balanced-stack system of Design Example 7. High performance, variable speed axial fans are available that may be readily installed in passive ventilation stacks and, thereby, provide the mechanically assistance needed when natural driving forces become small.

A.3.1 Design Example 8: Sizing an Axial Fan for a Balanced-Stack System

Consider the addition of an axial fan to the balanced-stack systems of Design Example 7. If this fan is placed in-line with the exhaust stack, then it may be modeled simply by adding the relation for a fan component, Equation 5.13, to the loop equations formed above, Equations A.27 and A.28, as:

Unregulated System

$$\overbrace{\frac{\rho C_a \dot{V}^2}{2A_a^2}}^{\text{supply inlet}} + \overbrace{\frac{fL_b}{2}\frac{\rho \dot{V}^2}{D_{hb}A_b^2}}^{\text{supply duct}} + \overbrace{\frac{\rho C_c \dot{V}^2}{2A_c^2}}^{\text{supply diffuser}} + \overbrace{\frac{\rho C_d \dot{V}^2}{2A_d^2}}^{\text{exhaust grill}} + \overbrace{\frac{fL_e}{2}\frac{\rho \dot{V}^2}{D_{he}A_e^2}}^{\text{exhaust duct}} + \overbrace{a_o\left(\frac{D}{D_o}\right)^2\left(\frac{N}{N_o}\right)^2 + a_1\frac{\dot{V}}{\left(\frac{D}{D_o}\right)}\left(\frac{N}{N_o}\right) + a_2\frac{\dot{V}^2}{\left(\frac{D}{D_o}\right)^4}}^{\text{axial fan}} + \overbrace{\frac{\rho C_f \dot{V}^2}{2A_f^2}}^{\text{exhaust terminal}} = \Delta P_s + \Delta P_w \tag{A.40}$$

Regulated System

$$\overbrace{\frac{\rho C_a \dot{V}^2}{2A_a^2}}^{\text{supply inlet}} + \overbrace{\frac{fL_b}{2}\frac{\rho \dot{V}^2}{D_{hb}A_b^2}}^{\text{supply duct}} + \overbrace{\frac{\rho C_c \dot{V}^2}{2A_c^2}}^{\text{supply diffuser}} + \overbrace{\frac{\rho C_d \dot{V}^2}{2A_d^2}}^{\text{exhaust grill}} + \overbrace{\frac{fL_e}{2}\frac{\rho \dot{V}^2}{D_{he}A_e^2}}^{\text{exhaust duct}} + \overbrace{a_o\left(\frac{D}{D_o}\right)^2\left(\frac{N}{N_o}\right)^2 + a_1\frac{\dot{V}}{\left(\frac{D}{D_o}\right)}\left(\frac{N}{N_o}\right) + a_2\frac{\dot{V}^2}{\left(\frac{D}{D_o}\right)^4}}^{\text{axial fan}} \overbrace{- \Delta P_o \ln\left(1 - \frac{\dot{V}}{\dot{V}_o}\right)}^{\text{exhaust terminal}} = \Delta P_s + \Delta P_w \tag{A.41}$$

where, it will be recalled, a_o, a_1, and a_2 are a base fan performance curve coefficients; D is the fan size with D_o the size of the base fan; and N is the fan speed with N_o the speed of the base fan.

These loop equations would be valid when the fan is operating. When it is not operating, its resistance to airflow should be modeled. Thus the original equations, Equations A.27 and A.28, should also be modified by the addition of a model for the resistance of the (nonoperating) fan. Unfortunately, the nonoperating flow characteristics of fans are generally not available, although they could readily be measured. It may be sufficient to account for this resistance by using a fitting loss relation and coefficient for a simple duct obstruction such as a damper or screen that provides a similar obstruction to flow as the still fan would.

In a more general application, these loop equations could be used to size all system components. Here we will simply select the size and speed of a fan, given base fan performance characteristics and the system component sizes selected in Design Example 7. To do so, however, we must establish the design conditions (i.e., ΔP_s and ΔP_w) that should be used to size the fan. Returning to the histogram of wind pressure data for this particular system, Figure A.14, it is seen that driving wind pressures fall below 0.5 Pa for only 5% of the year. So, this value will be used as a reasonable design condition.

To proceed, all previous component parameters, sizes and volumetric flow rates must be substituted into Equations A.40 and A.41. The volumetric flow rates for all components including the fan are equal, by continuity, to the objective design ventilation flow rate of $\dot{V} = 0.033$ m^3/s. Values for the previous component parameters are enumerated in Design Example 7 and final design sizes are tabulated in Table A.4. For the fan, we will use the base fan performance data illustrated in Figure 5.2 (i.e., $a_o = 120$, $a_1 = -40$, and $a_2 = -600$).

Substituting all values into the governing loop equations we obtain the same fan design equation for both the regulated and unregulated systems. This is because both systems were designed for the same driving pressure (i.e., 3.31 Pa), hence upon substitution the sum of the pressure drops of all components except the fan component will equal this same design pressure. The final result is:

$$\overbrace{120\left(\frac{D}{D_o}\right)^2\left(\frac{N}{N_o}\right)^2 - 40\frac{0.033}{\left(\frac{D}{D_o}\right)}\left(\frac{N}{N_o}\right) - 600\frac{0.033^2}{\left(\frac{D}{D_o}\right)^4}}^{\text{axial fan}} + \overbrace{3.31\text{ Pa}}^{\text{all other components}} = 0.5\text{ Pa} \qquad \text{(A.42)}$$

This fan design relation is plotted in Figure A.16.

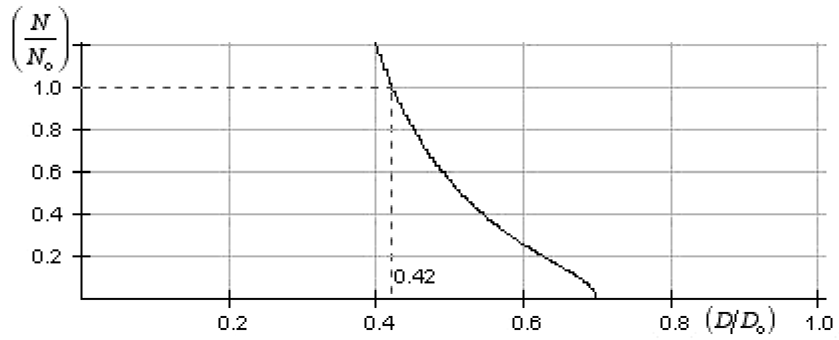

Figure A.16 Feasible combinations of fan size ratio (D/D_o) and speed ratio (N/N_o) for the fan-assisted balanced stack system of Design Example 7 for a design pressure difference of 0.5 Pa.

This relation establishes feasible combinations of the fan size ratio (D/D_o) and speed ratio (N/N_o) that will achieve the design ventilation flow rate of 0.033 m^3/s when the total stack and wind driving pressure difference drops to 0.5 Pa. We may use this relation to make a final design specification. For example, if we select a fan to operate at full speed for these conditions (i.e., $N/N_o = 1.0$) we would need a fan 42% (i.e., $D/D_o = 0.42$) of the base fan size. At larger driving pressures, the fan could then operate at a fraction of its full operating speed.

A.3.1.1 Conclusion to Design Example 8

As this design example illustrates, fan components can be directly included in the formation of pressure loop equations and, therefore, sized using these loop equations. When included, however, asymptotic limits to the loop equation are no longer simply defined as the mathematical form of the fan component relation is not hyperbolic in terms of the fan component design variables or powers of these variables. Consequently, the design of fan-assisted systems must follow a modified procedure. One possible procedure may be generalized from this design example:

Step 1: Apply the loop equation design methodology to size all other passive system components system, accounting (approximately) for the passive resistance of the fan, for probable or mean design conditions.

Step 2: Reformulate the system loop equation(s) by replacing the passive resistance of the fan with the fan component relation, substitute all component variables determined in Step 1, and use the resulting equation to size the fan for probable minimum design conditions.

The determination of the probable minimum design condition must be approached carefully. As stressed in previous examples completely still wind may be unlikely at many sites. Consequently, it will be most reasonable to design for minimum probable wind pressures. Depending on the system global geometry and topology, stack pressures may be negative. In these instances, one should investigate the probable minimum combination of concurrent wind and stack pressures. It is entirely possible that this minimum will be negative in certain cases.

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Order Code for this publication: TN54