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Optimum Ventilation and Air Flow Control in Buildings

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Volume 2

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-one IEA Participating Countries to increase energy security through energy conservation, development of alternative energy sources and energy research development and demonstration (RD&D). This is achieved in part through a programme of collaborative RD&D consisting of forty-two Implementing Agreements, containing a total of over eighty separate energy RD&D projects. This publication forms one element of this programme.

Energy Conservation in Buildings and Community Systems

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. Seventeen countries have elected to participate in this area and have designated contracting parties to the Implementing Agreement covering collaborative research in this area. The designation by governments of a number of private organisations, as well as universities and government laboratories, as contracting parties, has provided a broader range of expertise to tackle the projects in the different technology areas than would have been the case if participation was restricted to governments. The importance of associating industry with government sponsored energy research and development is recognized in the IEA, and every effort is made to encourage this trend.

The Executive Committee

Overall control of the programme is maintained by an Executive Committee, which not only monitors existing projects but identifies new areas where collaborative effort may be beneficial. The Executive Committee ensures that all projects fit into a pre-determined strategy, without unnecessary overlap or duplication but with effective liaison and communication. The Executive Committee has initiated the following projects to date (completed projects are identified by *):

Annex 1 Load Energy Determination of Buildings*
Annex 2 Ekistics and Advanced Community Energy Systems*
Annex 3 Energy Conservation in Residential Buildings*
Annex 4 Glasgow Commercial Building Monitoring*
Annex 5 Air Infiltration and Ventilation Centre
Annex 6 Energy Systems and Design of Communities*
Annex 7 Local Government Energy Planning*

Annex 8 Inhabitant Behaviour with Regard to Ventilation*
Annex 9 Minimum Ventilation Rates*
Annex 10 Building HVAC Systems Simulation*
Annex 11 Energy Auditing*
Annex 12 Windows and Fenestration*
Annex 13 Energy Management in Hospitals*
Annex 14 Condensation*
Annex 15 Energy Efficiency in Schools*
Annex 16 BEMS - 1: Energy Management Procedures*
Annex 17 BEMS - 2: Evaluation and Emulation Techniques*
Annex 18 Demand Controlled Ventilating Systems*
Annex 19 Low Slope Roof Systems*
Annex 20 Air Flow Patterns within Buildings*
Annex 21 Thermal Modelling*
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Annex 30 Bringing Simulation to Application
Annex 31 Energy Related Environmental Impact of Buildings
Annex 32 Integral Building Envelope Performance Assessment
Annex 33 Advanced Local Energy Planning

Annex V Air Infiltration and Ventilation Centre

The IEA Executive Committee (Building and Community Systems) has highlighted areas where the level of knowledge is unsatisfactory and there was unanimous agreement that infiltration was the area about which least was known. An infiltration group was formed drawing experts from most progressive countries, their long term aim to encourage joint international research and increase the world pool of knowledge on infiltration and ventilation. Much valuable but sporadic and uncoordinated research was already taking place and after some initial groundwork the experts group recommended to their executive the formation of an Air Infiltration and Ventilation Centre. This recommendation was accepted and proposals for its establishment were invited internationally.

The aims of the Centre are the standardisation of techniques, the validation of models, the catalogue and transfer of information, and the encouragement of research. It is intended to be a review body for current world research, to ensure full dissemination of this research and based on a knowledge of work already done to give direction and firm basis for future research in the Participating Countries.

The Participants in this task are Belgium, Canada, Denmark, Germany, Finland, France, Netherlands, New Zealand, Norway, Sweden, United Kingdom and the United States of America.

17TH AIVC CONFERENCE
"Optimum Ventilation and Air Flow Control in Buildings"

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OPTIMUM VENTILATION AND AIR FLOW
CONTROL IN BUILDINGS

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CALCULATION METHODS FOR THE DETERMINATION OF AIR FLOW RATES IN DWELLINGS

Andrew Cripps

Building Research Establishment
Garston, UK

Calculation Methods for the Determination of Air Flow Rates in Dwellings

by Andrew Cripps, BRE

The members of CEN TC156, WG2, Ad Hoc Group 4, on whose work this paper is based are

V Dorer, Switzerland
P Barles, France
A Blomsterberg and T Carlsson, Sweden
A Cripps, UK
W de Gids, The Netherlands
E Kukkonen, Finland
J-G Villenave, France

Abstract

Ad Hoc Group 4 of Working Group 2 of CEN TC156 (Ventilation) was set up to put forward standardised techniques for estimating ventilation rates in dwellings. The purpose of the standard is to ensure that different people carrying out calculations with the same input data will obtain the same result. This will allow the use of these results in energy, heating load, IAQ or other calculations.

The methods proposed use two different techniques, an explicit and an implicit one. The explicit one involves more approximations, but can be carried out with a hand calculator. The implicit one requires the use of a computer.

This paper explains the methods used, justifies the approximations made and gives examples of the use of the explicit method. These show that the explicit method gives results in good, but not perfect, agreement with experiment, and also that the method is simple to use.

Introduction

The Ad Hoc Group 4 of CEN TC156, Working Group 2 was set up to develop a simplified method for ventilation calculations. The purpose is to ensure that people working with the same data in different places will achieve the same result. This is not likely if they use different models, or the assumptions used by each are not made clear.

Ventilation calculations are needed within three other tasks:

- Energy use predictions
- Air quality estimates
- Heat load calculations

Because of the differences in the applications each type of calculation has a different requirement for weather data, although the basic calculation is the same. For average energy calculations something close to average weather conditions must be used. For air quality, weather data with a low wind speed and a low temperature difference would be used, whilst for heat load calculations a 'worst case' weather situation will be used.

It is expected that other CEN standards will refer to this proposed standard when they need to include the ventilation rate as a component of their calculation. This will apply particularly to calculations of energy use in buildings, which currently use a very simple algorithm for ventilation rate, and to a proposed CEN standard on the design of residential ventilation systems.

In order to make a simplified model many assumptions have to be made. These are made when the input data are considered, and in the design of the methods. The input data are discussed in the next section. The following section describes how the explicit method is used, including the assumptions made within it. The implicit method is presented next, with the minimum of detail as it is based on other models. Finally some examples of how the explicit method has been used are given, and the implications for the future are discussed in the conclusions.

Input data required

To calculate the ventilation rate using either method the following data are required. If they are not available then they need to be estimated, and this must be made clear in the presentation of the results.

Building and dwelling:
- the type of building,
- the building height,
- the degree of shielding from the wind,
- the number of facades of the dwelling which are exposed to the wind,
- the air leakage rate of the dwelling (or the n_{50} value and the volume),
- the distribution of the air leakage over the envelope.

Ventilation system:
- the type of system (natural, mechanical extract or mechanical balanced system),
- the capacity of the ventilation system,
-     - natural ventilation openings and duct terminal heights,
-     - mechanical flows,
- the time these provisions are used.

Finally the climatic data have to be known:
- wind speed,
- external temperature
- internal temperature (measured or assumed).

Within the draft standard there is a lot of guidance on values of these for default cases. None of this is discussed further here.

The Explicit method

The idea of the explicit method is that it can be carried out easily using a pocket calculator or in a very simple spreadsheet. In order to achieve this there are a number of assumptions which need to be made. These are discussed below.

Both of the explicit and implicit methods are based on widely used air flow equations for a single zone model. These include equations for leakage flows, the flow rates through openings and the pressure differences generated from the wind and stack effects. These are not discussed further here. However the explicit method has to take some extra assumptions to avoid the need to use an iterative solution method.

1. Because the leeward side has a greater area than the windward side it is assumed that the air flow is dominated by the leakage of the windward side.
2. There is an effective pressure difference across the windward side, which is a summation of windward and leeward pressures. The internal pressure is assumed to be close to the leeward side pressure.
3. Default values for wind pressure coefficients, valid for a wind sector of approximately $\pm 60^{\circ}$ to the facade axis, are given in the standard. The wind direction is not considered more specifically.
4. The impact of the distribution of surface leakage is accounted for by correction factors calculated by comparison with a single zone model.
5. Flow rates due to the wind and stack components are added in quadrature.
6. Airing is treated as a single sided, single room ventilation effect through open window and doors. Cross-ventilation effects are not considered.

Using these assumptions the calculation becomes a series of straightforward steps, the outline for which is shown in figure 1. This shows the order in which the flows are calculated and then combined to give the totals. Although there are a number of stages to this calculation, in many cases several of them may not be present. In the Examples given later the BRE case only required the infiltration flow, qv-inf, as all the other flows were zero. For details of the equations and the details of the input data needed see the draft standard [1], available from Viktor Dorer at EMPA in Switzerland.

Internal flows in an apartment (i.e. flow from the staircase area of an apartment block into the apartment), combustion induced flows and room to room flows within a dwelling have also been considered, but are not addressed in this paper.

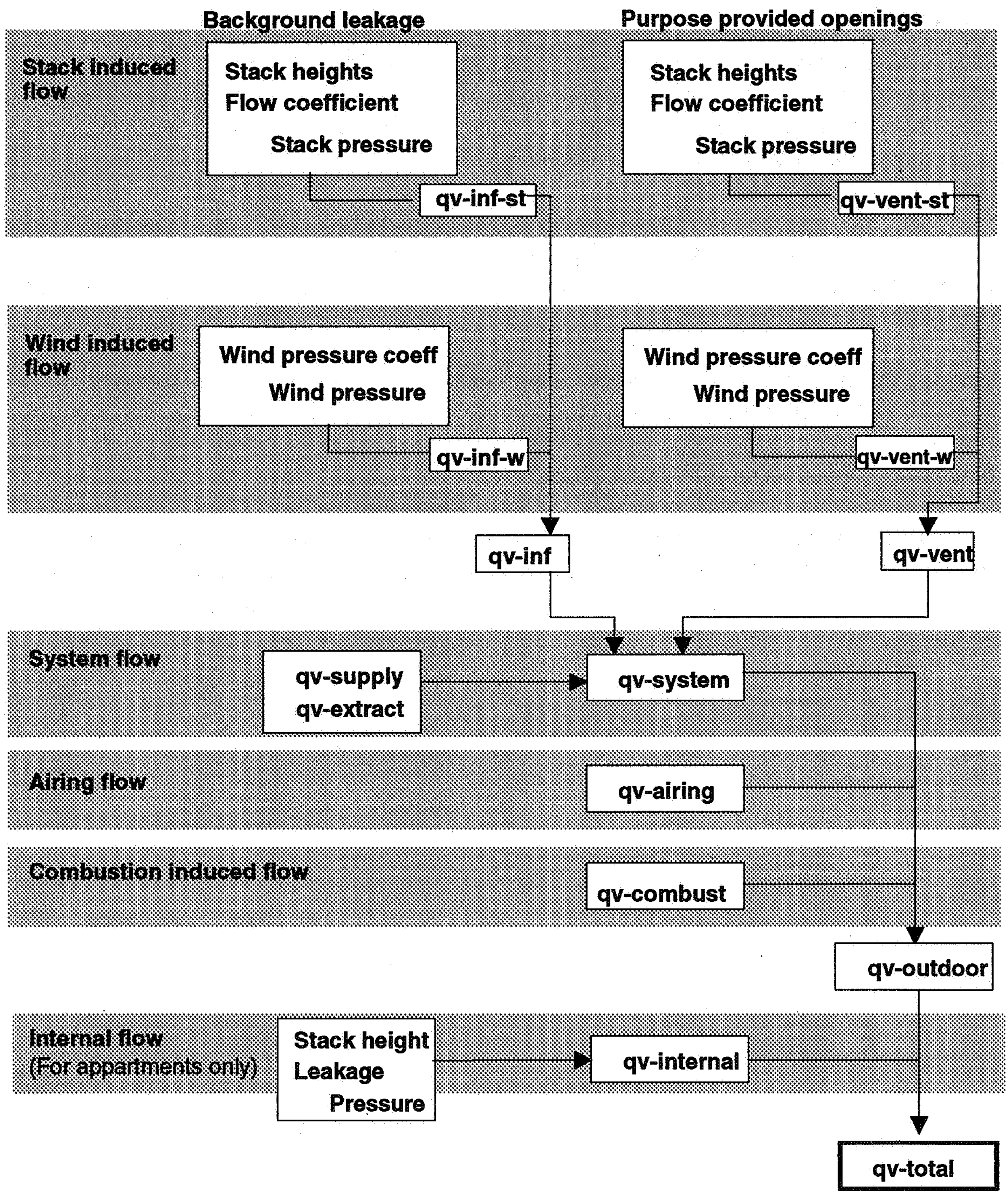

Figure 1: Calculation flow diagram for the explicit method

The implicit method

This method is based on the widely used single zone model [2,3,4,5]. This means that the dwelling is represented by one zone with one temperature and one pressure value. This zone pressure value has to be determined by an iterative process based on a flow balance equation of all the flows entering and leaving this zone.

Given the wide availability of computers the implicit method might be widely used as it will run easily on a very small PC or similar computer. However for some applications the explicit method is easier to use; for example it can be built into a spreadsheet for carrying out energy calculations, so it is not clear at this stage which method will be most useful.

Because the implicit method is widely understood, and not discussed in the examples below it is not taken further in this paper, but more information can be found in the draft standard [1].

Example 1: Swedish single family house

Thomas Carlsson and colleagues at SP in Sweden (the Swedish National Testing and Research Institute) have experimental measurements for a single family house in Sweden. They have also used the explicit method to estimate the ventilation rate from corresponding weather data. The house is described in figure 2.

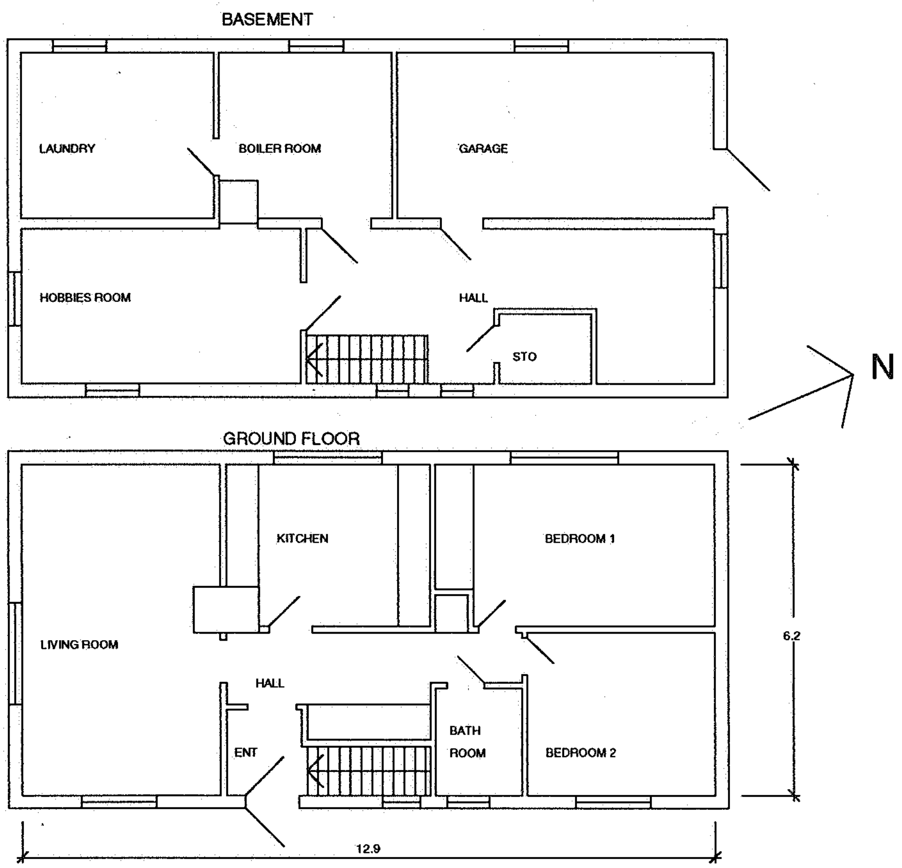

Figure 2: The floor plan of the Swedish test house

The data to be used for the calculation are as follows:

Building type	Single Family House
Roof slope	22°, i.e. in range 10°-30°
Ventilation system	Natural ventilation with passive stack, no vents
Overall leakage	7.8 ach, exponent n = 0.79
Building height	5 m
Volume	277 m^3
Temperatures	18.8°C inside, 0.3 °C outside
Shielding	normal
Wind velocity at site	0.9 m/s

Using these data, and the equations from the draft standard the flows are found to be:

qv-inf-st = 12 l/s,
qv-inf-w = 4 l/s.

These stack and wind components combine to give a total predicted flow rate of

qv-total = 13 l/s.

The measurements taken by SP give a flow rate for these conditions

q-measured = 16 l/s ± 10%.

Therefore in this case the simplified model is giving a good agreement with the measured data, considering the small number of input data used.

Example 2: Swedish multifamily building

The method has also been used on 2 apartments within a multifamily building in Sweden. A diagram of the apartment within the building is shown as Figure 3. The building is slightly complex in form, with a bend in the centre of around 70°.

The basic data are the same for both apartments:

Building type	Two room apartment in 4 storey building, with vertically open staircase
Roof slope	> 30°
Ventilation system	Natural ventilation with passive stack and vents
Building height	14 m
Shielding	Open

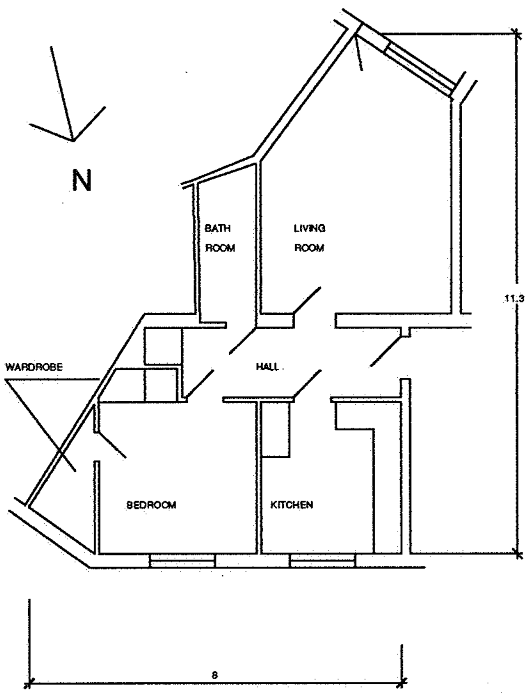

Figure 3: Floor plan of apartment within multifamily building

There are measurement results available for a flat on the second floor, and another on the fourth, and these are presented separately, along with the case dependent data.

Second floor apartment

Overall leakage	2.2 ach, exponent n = 0.86
Volume	139 m^3
$H_{inf} = H_{vent}$	10.5 m (the effective stack height, which depends on the passive stack)

	Cold set	Mild set
Internal temperature, °C	19.3	24
External temperature, °C	-0.9	14
Wind speed, m/s	0.9	1.3
Predicted flow, l/s	25	18
Measured flow, l/s	35 ± 10%	32 ± 10%

Table 1: Data and results for second floor apartment

Fourth floor apartment

Overall leakage	1.8 ach, exponent n = 0.71
Volume	134 m^3
$H_{inf} = H_{vent}$	5.1 m (this is the effective stack height, and depends on the passive stack)

	Cold set	Mild set
Internal temperature, °C	19.2	22
External temperature, °C	3.1	22
Wind speed, m/s	1.4	1.4
Predicted flow, l/s	17	8
Measured flow, l/s	24 ± 10%	19 ± 10%

Table 2: Data and results for fourth floor apartment

These results are not as good as the one for the single family case. In particular the model under-predicts significantly when the stack effect is small compared to the wind, as is occurring in each of the 'mild' cases. This suggests that there is a problem with the assignment of the pressure coefficients in the simplified model. It is not clear if this is really an error in the model or a detail caused by the geometry of the building. The building in which the apartment is located is not a simple shape, it includes a 'bend' of close to 70 ° in the middle, and this cannot easily be accounted for in such a simple model.

Example 3: BRE test house

The third example comes from one of the BRE low energy test houses which are discussed more in a paper by Hartless presented at this conference [6]. These data come from one of a matched pair of typical UK houses for which continuous monitoring of ventilation rates and weather variables have been taken over several years. Two extracts from this monitoring are considered here.

Building type	Single Family House
Roof slope	>30°
Ventilation system	Natural ventilation no passive stack, vents closed
Overall leakage	13.8 ach, exponent n = 0.62
Building height	5 m
Volume	207 m^3
Shielding	normal

This particular house has a relatively leaky floor above a ventilated void, which is quite a common construction type in the UK. Because of this an extra compensation factor was needed since the model was developed assuming low flow rates through the floor. Because the weather data are collected at half hour intervals it is possible to plot the measured ventilation rates against the predictions from the model. The first period is from July 1995, the second from March 1995.

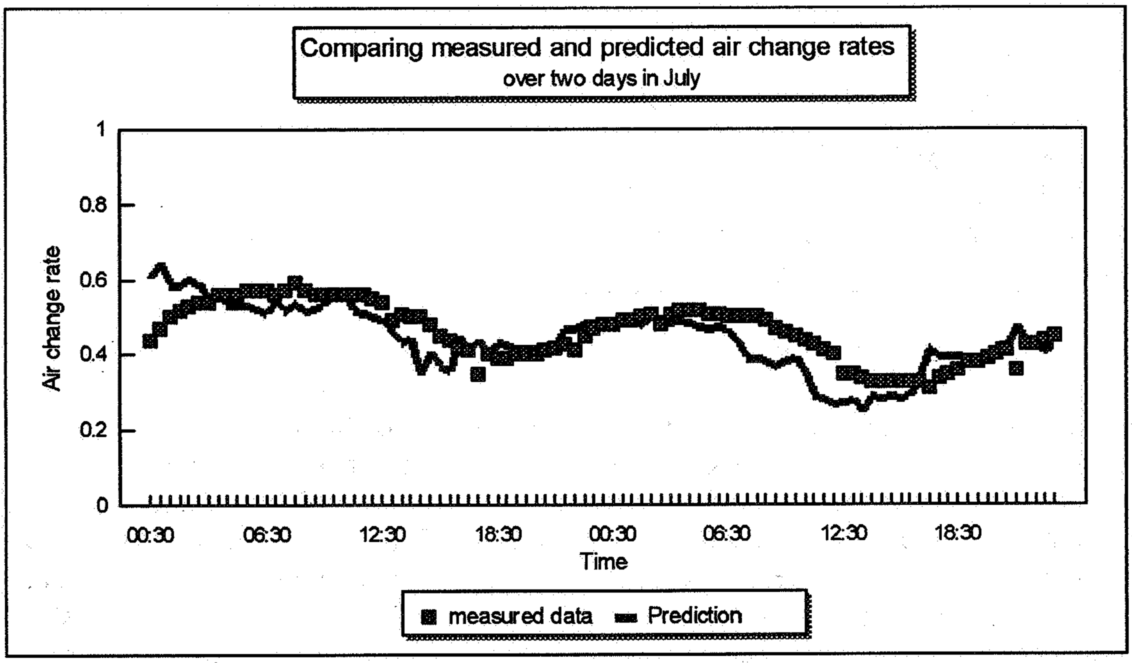

Figure 4: Graph of predicted and air change rates against time: data from July

These results are particularly good with close correlation between the measured and predicted air change rates. This period had relatively low wind speeds, the average measured on site being 2.5 m/s.

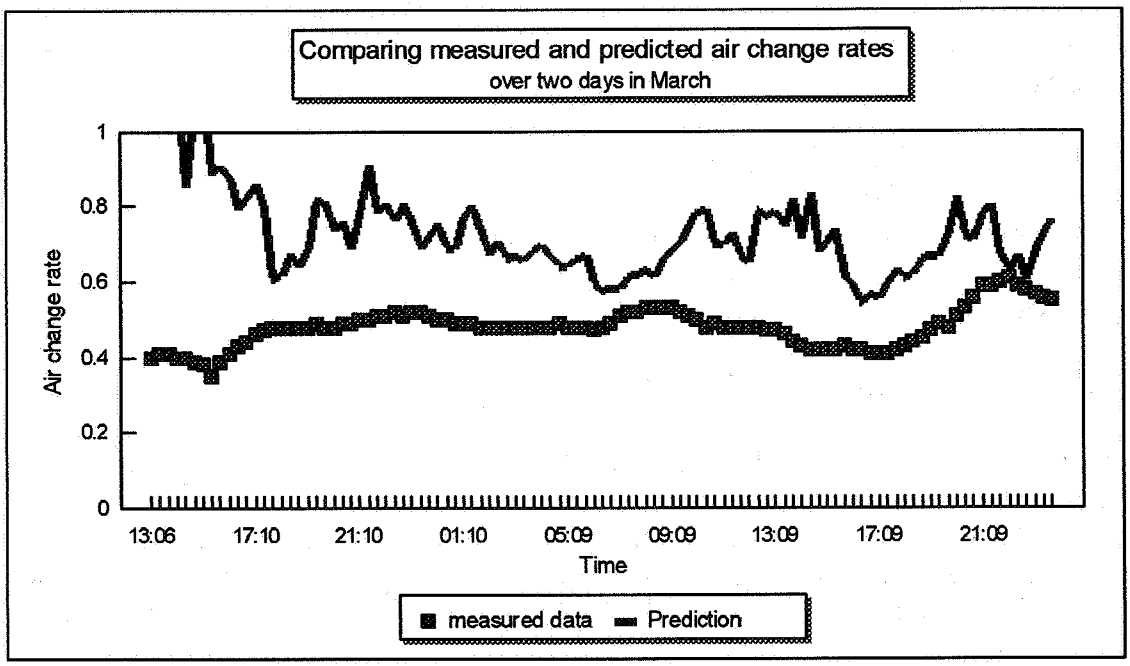

Figure 5: Graph of predicted and air change rates against time: data from March

In the second set presented as Figure 5 the wind speeds were much higher, with an average of 5m/s. The correlation here is much worse and this suggests, as with the earlier Swedish results, that there is more work to be done on pressure coefficients. However it should be noted that Hartless in his work [6] used a computer based single zone model and had the same problems modelling the March data.

Conclusions

In this paper the Draft CEN standard simplified method for ventilation calculations has been described and three examples of its use have been presented. These show that the model is easy and quick to use. It is the requirements of a standard that make the documentation needed rather long.

The examples show that the model gives results which have reasonable agreement to measured data where the stack effect dominates over the effect of the wind. As with nearly all models it has more trouble with the case when wind speed dominates, and more work is therefore needed on the pressure coefficients to be used.

The next step for the standard is for the draft to be considered again by Working Group 2 of the Technical Committee TC156. It is not too late to comment on the draft, but don't wait too long, as we are trying to complete the standard in the next year. Contact Viktor Dorer at EMPA for a copy of the current draft standard.

References

1 Draft standard CEN TC 156/ WG2/N203 *Calculation Methods for the Determination of Air Flow Rates in Dwellings*, 16. Oct. 1995, available from Viktor Dorer at EMPA, Switzerland.

2 Warren P R and Webb B C *The relationship between tracer gas and pressurisation techniques,* Proceedings of the first AIC conference, Windsor, UK, 1980, published by AIVC, UK.

3 Cripps A J and Hartless R P, *'BREVENT' A ventilation model,* BRE report AP66, December 1992, Published by BRE, UK.

4 Liddament M, *AIDA - an air infiltration development algorithm.* Air Infiltration Review, vol 11, no 1, Dec 1989.

5 Sherman M and Grimsrud D T, *Infiltration-pressurisation correlation: simplified physical modelling.* ASHRAE semi-annual meeting, Denver, USA, June 1980.

6 Hartless R P, *Subfloor and house ventilation rates: comparing measured and predicted values,* To be published in the proceedings of the 17th AIVC conference, Gothenburg, Sept 1996, by AIVC, UK.

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

17th AIVC Conference, Gothenburg, Sweden,
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(Title) **PROBABILISTIC ANALYSIS OF AIR INFILTRATION IN A SINGLE FAMILY HOUSE**

(Authors) Krystyna Pietrzyk and Kamal Handa

(Affiliation) Building Aerodynamics, Chalmers University of Technology

S-412 96 Gothenburg, Sweden.

SYNOPSIS

A Probabilistic model of air change rate in a single family house based on full-scale measurements has been developed. The probability of air change rate exceeding certain prescribed limits (risk of improper ventilation or excessive heat flow) is evaluated by utilising the distribution function based on calculated air flow rate. In this way the results are expressed in terms of the R-S model generally used in the safety analysis of structures. In particular, the probability of excessive and/or insufficient air infiltration can be expressed in terms of safety index β to describe the reliability of the building with respect to natural ventilation. The probability density functions for the air change rate have been established for different wind directions

Two methods of reducing the risk of unhygienic conditions have been studied. The first one is based on introducing extra small openings uniformly distributed over the building envelope and providing a fully naturally ventilated system. The second method consists in introducing mechanical exhaust ventilation system coupled with natural ventilation. Probability distributions of air change rates have been analysed for these two cases.

1. INTRODUCTION

Probabilistic model of air infiltration has been developed from full-scale measurements carried out on a single family timber framed detached house located on the outskirts of Gothenburg Figure 1 shows the plan of the house seen from above togather with eight wind direction sectors. Details of the test house and the model presented herein are based on the work in reference 1.

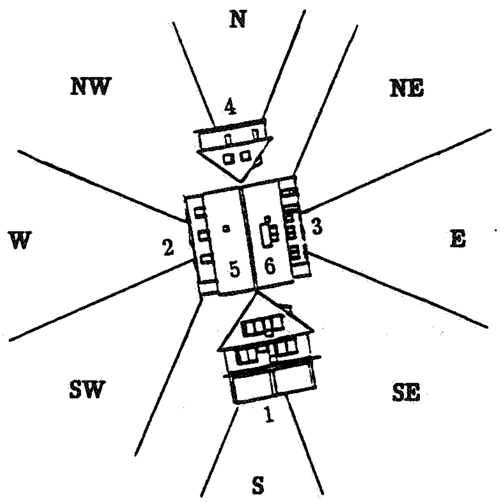

Figure 1. *Plan of the test house and sectors for wind direction*

Wind velocity and direction togather with pressure differences across the building envelope and outside and inside temperatures were recorded continuously for a period of eight months. Measurements of the airtightness of the building by *blower door test* method and the measurements of the air change rate by *tracer gas decay* technique have also been performed.

2. STATISTICAL PROPERTIES OF THE AIR CHANGE RATE

The air change rate per hour ACH is calculated from the air flow rate Q and the volume V of the enclosure and is defined as:

$$ACH = Q / V$$

The air flow rate is a function of the pressure difference and the leakage properties and is given by:

$$Q = K A \sqrt{\Delta p}$$
$$K = f(Re)\ \alpha \sqrt{2/\rho}$$

Where K is a leakage function of pressure difference ($\sqrt{m^3 / kg}$), $f(Re)$ is a coefficient of frictional resistance, α is the relative leakage area (A_1 /A) and ρ is the air density. Δp is the pressure difference. The air flow rate Q depends on the

- Climatic parameters (wind velocity, wind direction and temperature difference)
- Structural parameters (leakage characteristics of the building)
- Serviceability parameters (internal temperature, intentional openings etc).

All of these parameters should be treated as random variables with their own statistical properties. Wind velocity variations in time can be described by two parameter ***Weibull*** distribution. The full-scale measurements of air temperatures have shown that the temperature difference between outside and inside of the house follows *Normal* distribution. Leakage properties of the house can be different depending on the wind direction. For the openings, where a boundary flow occurs, leakage properties can vary significantly with the magnitude of pressure difference across the building envelope. It implies that the leakage properties become function of randomly distributed pressure differences.

Model of air change rate ACH based on the full-scale pressure measurements has been developed by assuming that:

- the mean rate of air flow through the leaks is determined by 10-minute mean pressure differences across the building envelope,
- the amount of air entering and leaving the house is in balance during the whole measurement period,
- the leakage area is uniformly distributed over the building envelope except one large opening on the south side of the building,

Air change rates have been evaluated as 10-minute mean values for each hour of the measurement period based on separate estimation of the amount of air entering and leaving the house. Histograms of air change rate have been calculated for eight wind direction sectors.

Air change rate for wind blowing from the south and south-east directions is approximately *log-Normally* distributed. Air change rates for other wind directions seem to follow *Normal* distribution . Because of different types of distributions of air change rates for certain wind directions, further analysis has been done within the groups of homogeneous data.
The probability density functions have been fitted to air change rates for two cases:

Case 1. Large opening is situated on the windward side and wind direction $\theta_1 = \{S, SE\}$

Case 2. Large opening is not situated on windward side and wind direction $\theta_2 = \{N, NE, E, SW, W, NW\}$

For *Case 1*, three-parameter *log-Normal* distribution with threshold of 0.05 has been fitted to the data of *ACH* as shown in figure 2. For *Case 2*, *Normal* distribution truncated at 0.05 has been fitted as plotted in figure 3. These two distributions have been checked by the Kolmogorov test at significance level of 0.05.

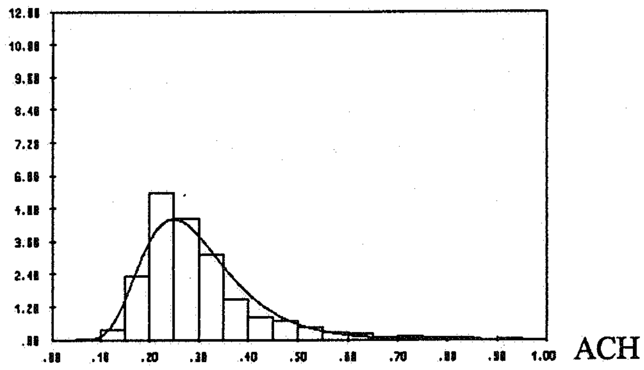

Figure 2. *Histogram and the probability density function fitted to ACH for Case 1*

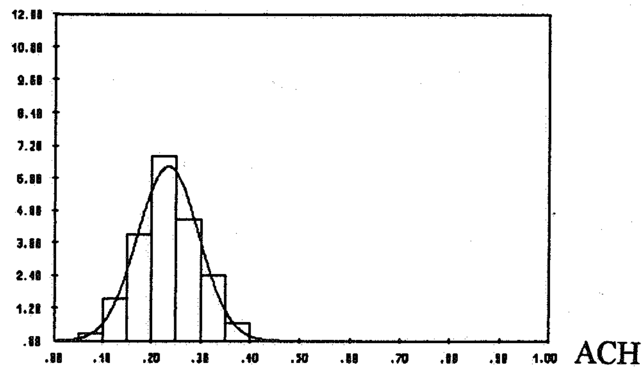

Figure 3. *Histogram and the probability density function fitted to ACH for Case 2*

3. RISK ANALYSIS OF AIR INFILTRATION IN A BUILDING

Malfunctioning of ventilation in the house occurs when air change rate is less than allowable minimum value R_1 leading to unhygienic conditions or is higher than the maximum value R_2 resulting in unnecessary heat losses. On the basis of estimated distribution of air change rate probability of *ACH* exceeding certain limits R_1 and R_2 can be evaluated. The problem of air flow through the building envelope is considered in terms of load-resistance model, where resistance R describes the serviceability limit states R_1 and R_2 and the load effect S expresses the air change rate. Serviceability conditions are not fulfilled when $S<R_1$ or $S>R_2$. Figure 4 shows a hypothetical model for the serviceability limit state for *ACH*.

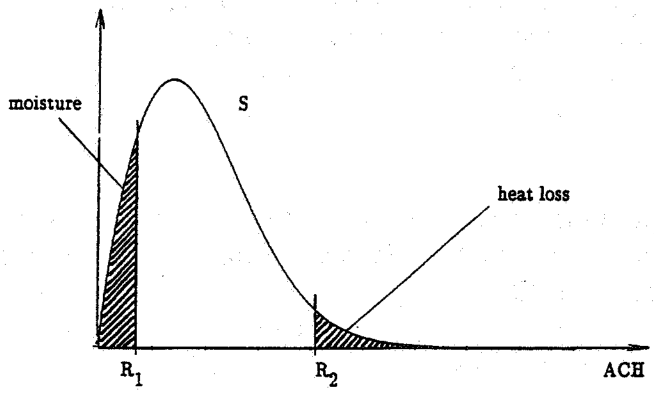

Figure 4. *Modelling of serviceability limit states for ACH.*

For assumed allowable limits R_1 and R_2, safety indices β_1 and β_2 can be calculated from the standardised probability density function of *ACH*. Since there are two different probability density functions which have been estimated for different wind direction sectors,the method of calculation of safety indices is presented for *Normal* and *log-Normal* distributions in Table 1.

Distribution function	***Safety Index*** β_1	***Safety Index*** β_2
Normal	$(\mu_x - R_1) / \sigma_x$	$(R_2 - \mu_x) / \sigma_x$
Log Normal	$[\mu_y - \ell n(R_1 - y_0)] / \sigma_y$	$[\ell n(R_2 - y_0) - \mu_y] / \sigma_y$

Table 1. Safety Indices for *Normal* and *log-Normal* distributions

The probability density functions can be read from the standard normal distribution tables.

The assessment of the allowable values of R_1 and R_2 are governed by the Code of Practice as well as a subjective estimation based on economy. According to the Swedish Code of Practice the minimum value of air flow is 0.35(l / s) per 1m^2 of the floor area which means that 0.52 changes per hour is the minimum requirement for the test house. Risk of air change rate less than prescribed limit R_1 is obtained by calculating the conditional probabilities for different wind direction.

$$P(ACH<R_1) = P(\theta=\theta_1) \times P(ACH<R_1 \mid \theta=\theta_1) + P(\theta=\theta_2) \times P(ACH<R_1 \mid \theta=\theta_2)$$

where $P(ACH<R_1 \mid \theta=\theta_1)$ is the conditional probability for *log- Normal*
and $P(ACH<R_1 \mid \theta=\theta_2)$ is the conditional probability for *Normal* distribution.

For R_1=0.52 , the probability of unhygienic conditions for the test house is equal to 0.99 if the natural infiltration is the only source of supply.

4. THE INFLUENCE OF IMPROVED VENTILATION ON THE AIR CHANGE RATE DISTRIBUTION

Two methods for reducing the risk of unhygienic conditions have been studied . The first one is based on introducing small openings uniformly distributed over the building envelope and providing a fully naturally ventilated system. The second method consists of introducing extra exhaust mechanical ventilation. A brief description is presented below.

First method

The air change rate is approximated as a linear function of the area of the openings in the buildings envelope. Figure 5 shows that the increase in the mean air change rate results in the decrease of risk of inadequate ventilation. Figure 5 also shows the influence of type of probability density function on the *ACH*. It can be noted that for p_l below 10^{-2} the air flow rate is larger for *Normal* compared to *log-Normal* distribution.

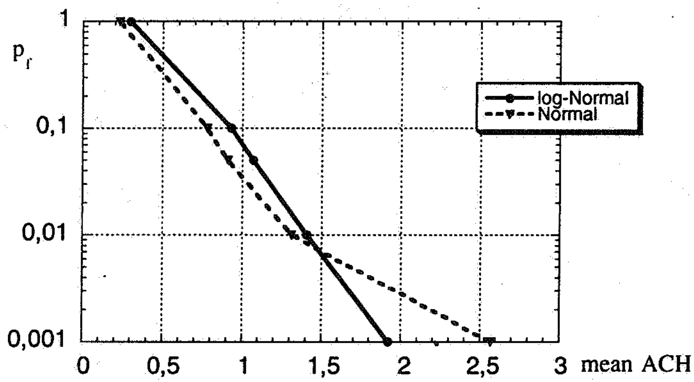

Figure 5. *Risk of unsatisfactory ventilation P(ACH<0.52) of ACH for naturally ventilated house*

The mean value of air change rate and proper size of openings required to reduce the risk of inadequate ventilation to p_l has been calculated from the conditional model and it is shown in Table 2.

Second method

In order to reduce the risk of unhygienic conditions in the house one can provide some extra air supply , e.g. by means of mechanical exhaust ventilation. The total value of air change rate ACH_{tot} is obtained by adding the contributions from both natural and exhaust ventilation.

$$ACH_{tot} = \sqrt{ACH^2_{nat} + ACH^2_{exh}}$$

We assume that the air change rate caused by exhaust ventilation is a deterministic quantity defined as $c=ACH_{exh}$ and that natural ventilation is a random quantity defined as $x=ACH_{nat}$. The probability density function for total air change rate $z=ACH_{tot}$ can be written as:

$$f_z(z) = [z/\sqrt{(z^2 - c^2)}]\, f_x(\sqrt{z^2 - c^2}) \qquad \text{for } x>c$$
$$z = \sqrt{x^2 + c^2}$$

The first order estimates of the expected value μ_z and the variance σ_z of the variable z can be obtained by expanding the function in Taylor series about the mean μ_x and retaining the first-order term of such expansion which results in:

$$\mu_z = \sqrt{\mu_x^2 + c^2} \quad \text{and} \quad \sigma_z^2 = \sigma_x^2 \, \mu_x^2 / (\mu_x^2 + c^2)$$

For assumed risk of failure, the necessary value of the efficiency of the exhaust ventilation and the actual mean and standard deviation of ACH_{tot} have been calculated in Table 2 . a is the size constant of the intentional openings provided for increasing the rate of air infiltration in order to achieve certain level of safety.

p_1	β_1	ACH_{nat}		a	ACH_{tot}		ACH_{exh}
		μ_x	σ_x		μ_z	σ_z	
0.001	3.08	2.42	0.80	9.7	0.64	0.034	0.59
0.01	2.32	1.34	0.44	5.4	0.61	0.035	0.59
0.1	1.27	0.82	0.27	3.3	0.57	0.038	0.51
0.989	-4.68	0.25	0.08	1.0	0.25	0.086	

Table 2 *Calculation of ACH for assumed probability and calculated from experimental data (shaded area)*

Figure 6 shows the influence of two different types of ventilation in the test house.

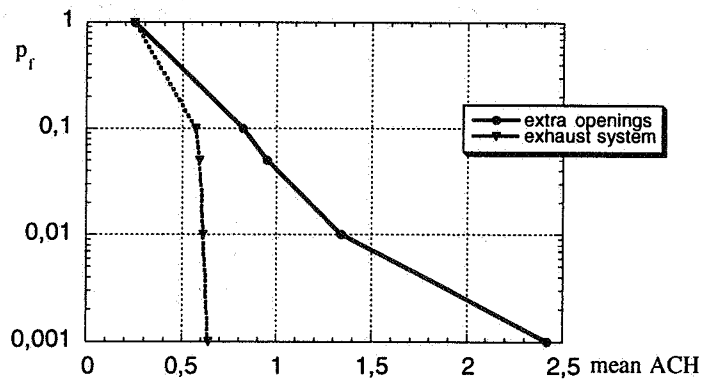

Figure 6. *The risk of unhygienic conditions $P(ACH_{tot}<0.52)$ for the mean value of ACH for the naturally ventilated house and for the house with exhaust ventilation*

The *first method* consists of increasing the natural ventilation, while the *second method* involves the use of mechanical exhaust system. For given risk of unhygienic conditions (e.g. 10^{-2}) one needs significantly higher mean air change rate for purely ventilated house (1.3/h) than for a house with exhaust system (0.6/h). The random character of climatic conditions has significant influence on the natural ventilation . The coefficient of variation of air change rate for the naturally ventilated test house has been calculated as 0.34. Coefficient of variation for air change rate for total ventilation (exhaust and natural) I_Z is obtained from:

$$I_z = I_x / [1 + (c^2 / \mu^2_x)]$$

A plot of ratio of air change rates caused by mechanical exhaust ventilation and natural infiltration is shown in Fig.7. Introduction of exhaust system causes significant decrease in the coefficient of variation of air change rate compared to the naturally ventilated house.

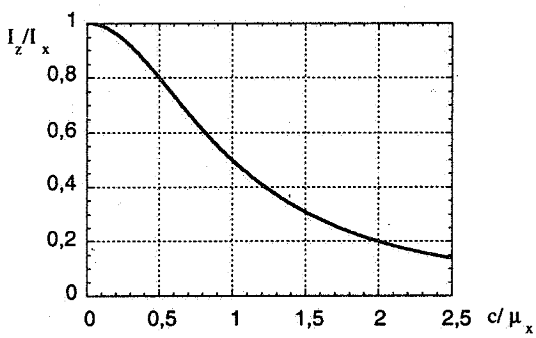

Figure 7. *Relationship between ratio I_z/I_x and ratio a / μ_x*

Figure 8 shows the probability density functions plotted from the data given in Table 2 concerning the assumed risk of inadequate ventilation p_l of 10^{-1} and the experimental data shown shaded. The results show that for the house under consideration , one needs to introduce additional openings in the envelope of the building with an equivalent area of about 3.3 times than original one or to switch on the mechanical exhaust system with efficiency of 0.51/hour for adequate air supply. This is true even if the contribution of natural driving forces is included in the model. The mean value of air change rate calculated for the two methods are different. For pure natural ventilation the mean value is 0.82/hour and for the combined effect of natural and exhaust ventilation it is equal to 0.57/hour. It means that for certain specified risk of unhygienic conditions the naturally ventilated house should be better ventilated than in the case of exhaust system

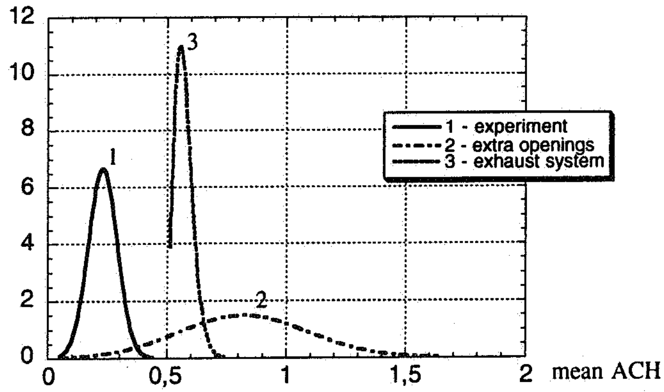

Figure 8. *Probability density functions of ACH: for data calculated from the experiment (1), based on risk level of p_l = 0.01 for improved natural ventilation (2) and based on risk level of p_l = 0.01 for combined effect of natural and exhaust ventilation (3) as shown in Table 2.*

5. CONCLUSIONS

The main conclusions of the work are:

1. Probability density functions of *ACH* are estimated for different wind directions and the results show the influence of wind on theirs shape when the leakages are non uniformly distributed over the building envelope. For cases where no large openings are situated on the windward side, the distribution of air change rate is found to be *Normal*. When a large opening is present on the windward side, the air change rate follows *log-Normal* distribution.

2. Risk analysis of malfunctioning of ventilation is carried out on the basis of conditional model of ACH and it considers two aspects dealing with insufficient and excessive ventilation. The assessment of the allowable values R_1 and R_2 *(R_1- limit for risk against unhygienic conditions, R_2 - limit for risk of unnecessary heat loss)* is a matter of regulations in the Code of Practice (R_1) as well as subjective estimations based on economical costs (R_2).

3. The risk of improper ventilation has been presented in the form of safety index β. Risk of uhygenic conditions in the test house of order 10^{-3} ($\beta \approx 3.$) depends on the type of distribution of *ACH Normal* and *log-Normal* which are used in presenting the probabilistic model.

4. The coefficient of variation of *ACH* for the house with exhaust ventilation is significantly smaller than for naturally ventilated house. For a certain level of risk of unhygienic conditions the naturally ventilated house should be better ventilated than in the case of using exhaust system.

6. ACKNOWLEDGEMENTS

The authors thank the Swedish Council for Building Research (BFR) for financial support .

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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System Safety Analysis on the Performance of Mechanical Ventilation Systems

Johnny Kronvall

J&W Consulting Engineers
Slagthuset
S-211 20 Malmö
Sweden

Synopsis

System safety of the performance of mechanical ventilation systems can of course be analysed by means of general methods for system safety analysis. Such methods are used a lot in industrial practice, especially in manufacturing industry. However applications on ventilation systems are more or less non-existing today. This paper summarises today's methods for system safety analysis and shows possible future ways of applying the methods on performance analyses of mechanical ventilation systems..

1 INTRODUCTION

Reliability, in the context of ventilation performance can be defined as:

the probability that the ventilation system provides certain required air flow rates in each occupied part of a building during the time between scheduled maintenance occasions.

The required air flow rates may be, for example, a specified fraction of the nominal air flow rates or certain fixed values.

As the probability of failure (complete failure or malfunction) is a key issue, the result of an evaluation procedure should be expressed in such terms.

The impact of human behaviour on ventilation reliability can be extensive. For example, the user can hazard the ventilation performance of his dwelling by obstructing the supply air terminal devices in order to avoid draughts. The draught will disappear, but the intended air flow patterns in the dwelling are changed. The performance will also deteriorate if maintenance is performed badly or neglected.

In this paper methods for evaluating the system safety of mechanical ventilation systems for dwellings are outlined. The paper forms part of the Swedish contribution to the work of IEA-Annex 27 "Evaluation and Demonstration of Domestic Ventilation Systems".

2 SYSTEM SAFETY ANALYSIS IN GENERAL

Some different kinds of routines for system safety analysis are shown in figure 1.

The ***qualitative methods*** help us to understand the logical structure of different failure modes of the product, and how they interact. The ***quantitative methods*** use available data on the failure tendency of the components, estimations of times for repairing and human faults. These data can eventually be used for the calculation of the probability of a certain type of break-down of the system. The selection of a specific method depends on the complexity of the system, the amount of available statistical data and the degree of influencing human factors.

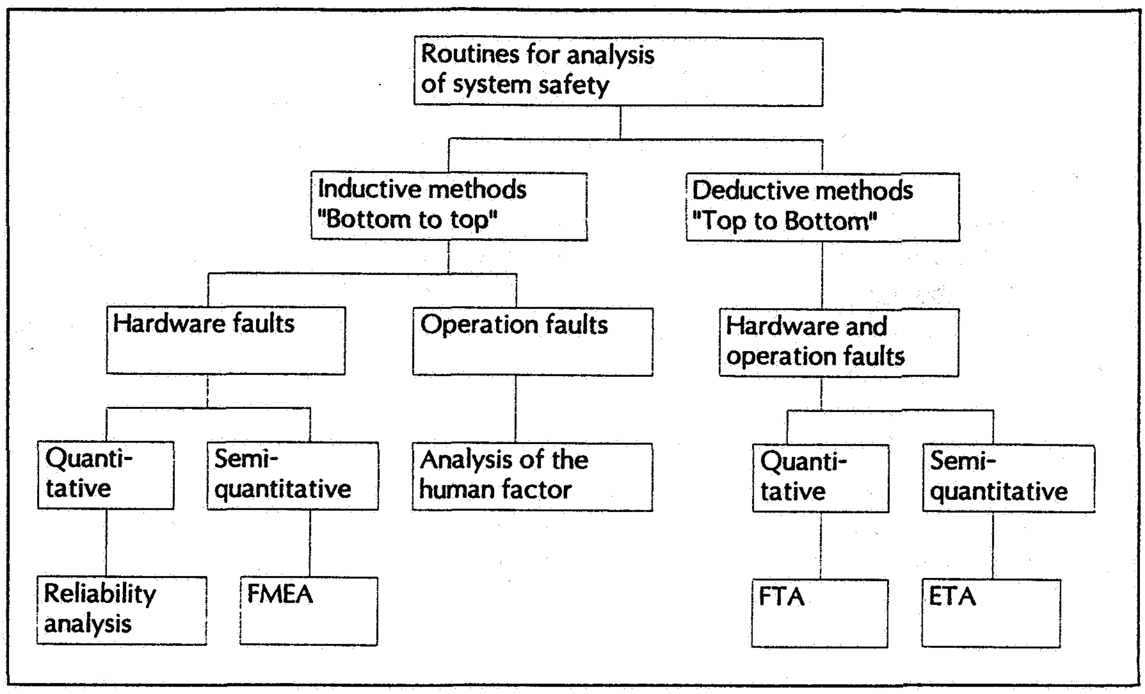


Figure 1 Different kinds of routines for system safety analysis. FMEA = Failure Modes and Effects Analysis, FTA = Fault Tree Analysis, ETA = Event Tree Analysis. After Rao (1992).

With the ***inductive methods***, the analysis work is started at components´ level by finding the failure modes of them. After that, one tries to find out what consequences there are on the system as a whole caused by a break-down on components´ level. Thus, the inductive analysis works gradually upwards from low component level up to part-system level and finally the system level. This is the way ***Failure Modes and Effects Analysis (FMEA)*** works. The military US standard US MIL-STD 1629 describes in detail how a FMEA-analysis should be worked out.

The ***deductive methods*** have a top-event as a starting point. Gradually one works downwards in the system and tries to find out the causes of the top-event. Thus, the way of working is opposite to the technique used in the inductive techniques. FTA (Fault Tree Analysis) and ETA (Event Tree Analysis) are examples of deductive methods.

FMEA is certainly the most commonly used technique for system analysis as far as product design is concerned.

ETA (Event Tree Analysis) is a graphical description of all possible events in a system. The method is based on binary logic, as events are seen in the perspective of if the have happened or not. A component is regarded as either working or non-working. Thus, it is not possible to take into account a "partially defect" state of a component. If the probabilities of each of all possible events in the tree is known, it possible to calculate the probability of (different) chains of events. The outcome of an ETA is a number of chains of events and their consequences for the system. The probability of each chain is also shown. ETA is a good technique for comparing different system configurations with each other from a perspective of operational safety. The method was initially developed for evaluating the safety of nuclear power plants.

FTA (Fault Tree Analysis) is frequently used for the analysis of complex systems. The method is extensively used within the nuclear and the aerospace industry. It is a deductive tool and as the method is highly standardised it has been used a lot. The user easily decides the degree of complexity of the system studied, as the method allows for studying separate parts of the system (so called sub-trees) one at a time. Evaluations by means of fault-tree analysis was originally developed by H.A. Watson at the Bell Telephone Laboratories in 1962. The purpose was to analyse the safety concept in connection with the launching equipment for the Minuteman-missiles. After that Boeing used and developed the method further.

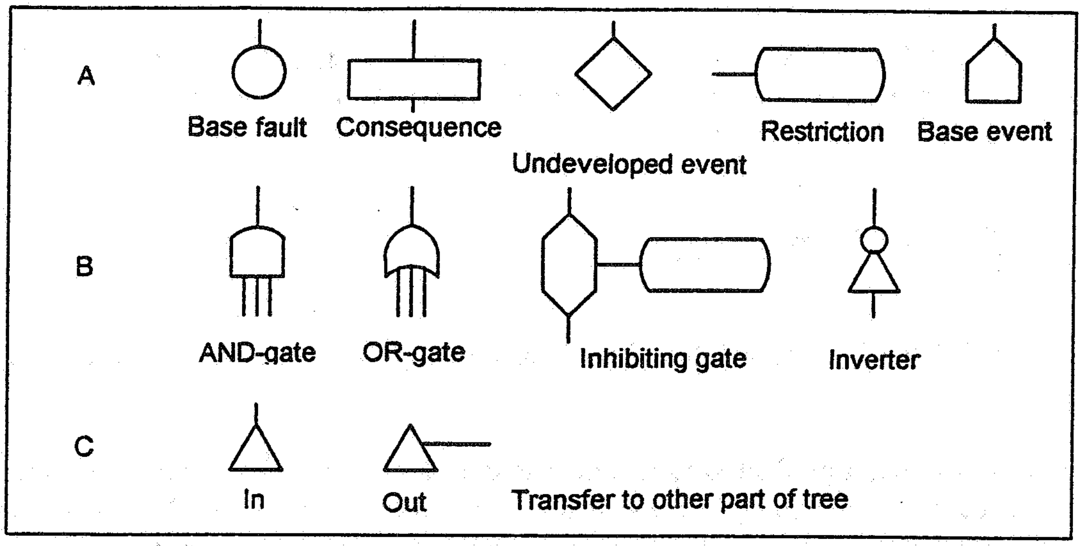
The purpose of FTA is to find the logical structure behind a fault event. Usually a FMEA is performed first, in which the system design, the operation of the system and the environment is analysed in order to find the causality of a fault. Thus FMEA is an important step towards the understanding of the system. Without such an understanding, it is not possible to perform a fault-tree analysis.

The performance of the product is described in a flow chart in which the flows of information, signals and other relevant aspects are specified. Then the flow chart is used for identifying the different functional sequences from inside to outside. Finally a logical chart is designed, in which the functional connections has been translated to logical relations.

In the logical diagram different logical symbols, see figure 2, are used.

Salem et al. (1976) has summarised the working methodology for fault-tree construction in the following six points:

1. The first step in the fault-tree analysis is to define a suitable TOP event that constitutes a serious system failure.
2. Usually several different, but equivalent, fault-trees can be constructed for a given system. Also different TOP events lead to different fault-trees.
3. For any specified TOP event, each possible event is examined to see whether it can, either alone or in conjunction with some other event(s), cause the TOP event.
4. The primary events that lead to the TOP event and the secondary events that cause each of the primary events are determined. The procedure is continued until all basic failures are identified.
5. The set of events that are all required to produce an event of interest are connected to AND gates.
6. The set of events that can individually produce an event of interest are connected to OR gates.



- Base fault: Faults caused by a component or a part-system, for which a probability can be assigned (from known empirical data).
- Consequence: A fault or event caused by a combination of other events via a logical gate.
- Undeveloped event: An singular fault, i.e. a fault that cannot be split up (developed) due to lack of information or lack of meaning.
- Restriction: Condition that must be fulfilled and directing an associated gate.
- Base event: An event normally occurring when the system is working.
- AND-gate: All in-signals must be true for opening the gate.
- OR-gate: One or more in-signals must be true for opening the gate.
- Inhibiting gate: No out-signal if the associated condition is fulfilled.
- Inverter: Changes one into zero (true turns false) or vice versa.
- Transfer symbols: Indicates that the branch continues into another tree (In) or indicates the top-event in a sub-tree (Out).

Figure 2 Standardised symbols for fault-tree analysis. Rau (1992).

3 MODELS FOR RELIABILITY

We use the symbol "*R*" for reliability. As *R*, by definition, is a probability it can be stated that

$$0 \leq R \leq 1$$

The reliability is often time-depending so

$$r = R(t) \qquad -\infty < t < \infty$$

When working with probabilities there are certain stochastic variables (SV) that are of primary interest. One of these determines the ability of the system to maintain the decided performance in an adequate way. This SV is named ***Time To Failure (TTF)*** or alternatively ***Time Between Failure (TBF)*** and is denoted T. The first one is used when non-repairable systems are considered while the other one is used to describe repairable systems. The SVs TTF and TBF must have some kind of statistical distribution. The notation F(t) denotes the probability that TTF and TBF will not be greater than *t*, shortly noted $P(T \leq t)$. Thus we can write

$$F(t) = P(T \leq t) = \int_{o}^{t} f(\tau)d\tau \qquad t \geq 0$$

which is called the fault probability. The corresponding density functions are denoted $f(t)$.

The probability that a product is functioning within the given time interval is denoted $R(t)$ and is called the reliability of the system, or the survival function of the system. As survival is the opposite of fault the following equation describes the reliability.

$$R(t) = P(T > t) = \int_{t}^{\infty} f(\tau)d\tau \qquad t \geq 0$$

We also realises that:

$$F(t) + R(t) = P(T \leq t) + P(T > t) = 1$$
$$\frac{dF(t)}{dt} = f(t) = -\frac{dR(t)}{dt}$$

These models make it possible to quantify the operation safety by means of the theories of probability and statistics. By using the expressions above, some more definitions are possible. The expected life cycle or the ***Mean Time To Failure (MTTF)*** and the ***Mean Time Between Failure (MTBF)*** could thus be written (E(T) denotes the expectation value.).

$$E(T) = \int_{0}^{\infty} \tau f(\tau)d\tau$$

or

$$E(T) = \int_{0}^{\infty} R(\tau)d\tau$$

E(T) is regarded as either MTTF or MTBF.

4 OPERATION SAFETY ON SYSTEM LEVEL

The functional structure of a system is often represented by a block diagram with different structures; series or parallel structures or combinations.

Series structures are used when it is demanded that all components work. The following symbols are used:

E_i = the event that the component i works at the time $t = t_0$.

$r_i = P(E_i)$ = the reliability of the component i at the time $t = t_0$.

R_s = the system reliability at the time $t = t_0$.

Thus, we have:

$$R_s = P(E_1 \cap E_2 \cap \ldots \cap E_n) = \prod_{i=1}^{n} r_i$$

If the performance of the system only demands that at least one of the components works, we have a ***parallel structure***. We have:

$$R_s = P(E_1 \cup E_2 \cup \ldots \cup E_n) = 1 - \prod_{i=1}^{n}(1 - r_i)$$

The expressions above are special cases of the so called " k of n model ". This general model expresses the behaviour of a system that demands that at least k out of n components work.

$$R_s = \sum_{j=k}^{n} \binom{n}{j} r^j (1-r)^{(n-j)}$$

The general expression above is valid only if all r_i are the same.

General structures normally includes both series and parallel semi-structures. In many cases the problem can be broken down to a pure series case or a parallel case.

5 APPLICATION TO MECHANICAL VENTILATION SYSTEMS

Mechanical ventilation systems are built up by a number of mechanical and electrical components, such as fan(s), electrical motor(s), damper(s), silencer(s), air terminal devices, system(s) for automatic control etc. The way that these components influence the performance of the system can of course be described in a fault-tree analysis. An attempt to work out a fault-tree for a simple mechanical exhaust ventilation system of a building is shown in figure 3.

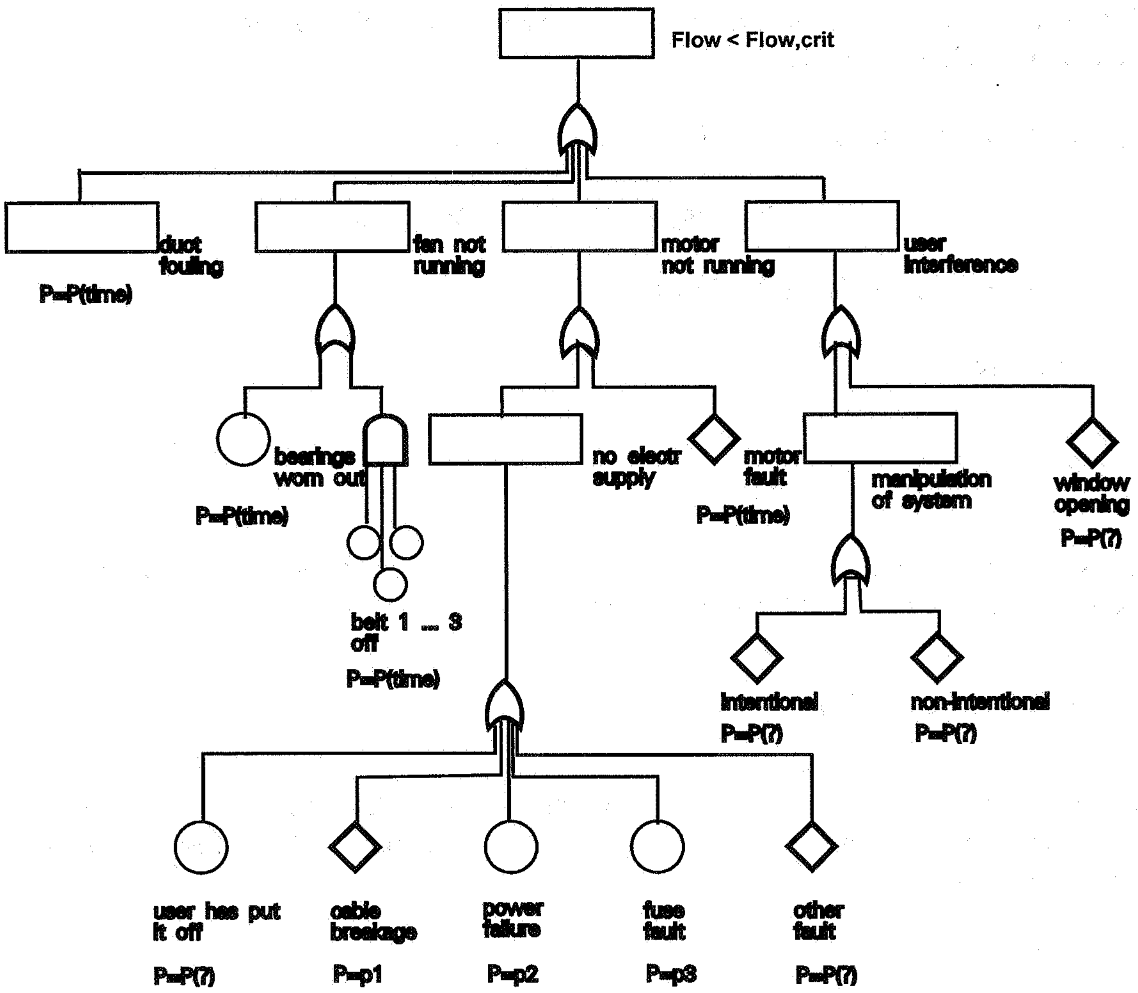


Figure 3 Fault-tree for a mechanical exhaust system in a building.

As TOP event is chosen a performance state of the system characterised by that the flow rate is lower than a critical value. At a level below the top event four different events that can cause malfunction are identified; *duct fouling* (which over time decreases the flow rates of the system), *fan not running*, *motor not running* and *user interference*. For each of these, other basic events at lower level(s) are identified. So far, as we work qualitatively, no major problems arises. What we do is essentially to find the logical structure for how different sources for malfunction influence the performance.

However, if we want to quantify the risk for malfunction or the probability of proper function, i.e. the reliability, we quickly run into a number of problems, most of them originating from the fact that we do not know the probability of failure for individual events. There are principally three different kinds of probabilities to estimate for individual events.

Fixed probabilities (marked as p1, p2 and p3 in figure 3). These are probabilities which are, in principle, not depending on time. For example power failure can be estimated if you can acquire data on how many hours per year you can expect power failure from the electricity company.

Time-depending probabilities. These are depending on in which state, i.e. at what time you analyse the problem. The failure intensity for mechanical components, for example, is not the same as long as the component is fairly new compared to when it grows older. Another example is duct-fouling with its consequences being gradually lower flow rates. The fact that failures appear independent of each other over time and that the failure intensity of individual components are depending on time, implies that a qualitative fault-tree analysis can not be performed as a single one, but be repeated with certain time intervals regarding the time of the use of the system.

More or less unknown probabilities. In the context of ventilation performance, typical examples are events based on user influence. These probabilities are very little known, not only because people are different, but also that the design of the ventilation system influences the behaviour.

The authors impression is that the current severe problems connected with the correct estimation of failure probabilities makes it very difficult and also very doubtful to perform accurate qualitative studies on system level for mechanical ventilation systems today. However, performing qualitative analyses based on construction of fault-trees can be very profitable in order to analyse how different failures are interconnected. Even quite rough guesses of probabilities of individual failures can be made, thus giving rough indications on performance on system level.

6 ACKNOWLEDGEMENTS

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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(Title)

Does the power law rule for low pressure building envelope leakage?

(Authors)

Iain S. Walker*, David J. Wilson and Max H. Sherman***

(Affiliation)

* Energy Performance of Buildings Group
Energy and Environment Division
Lawrence Berkeley National Laboratory
Berkeley, CA, USA

** Department of Mechanical Engineering
University of Alberta
Edmonton, AB, Canada

Does the power law rule for low pressure building envelope leakage?

Iain S. Walker, David J. Wilson and Max H. Sherman

1 Synopsis

Although the power law has been broadly accepted in measurement and air infiltration standards, and in many air infiltration calculation methods, the assumption that the power law is true over the range of pressures that a building envelope experiences has not been well documented. In this paper, we examine the validity of the power law through theoretical analysis, laboratory measurements of crack flow and detailed field tests of building envelopes. The results of the theoretical considerations, and field and laboratory measurements indicate that the power law is valid for low pressure building envelope leakage.

2 Introduction

The functional form of the pressure flow relationship for building envelopes has been a topic of debate. Historically, some practitioners supported a power law equation (Liddament (1987)) and others a quadratic form (Etheridge (1987)). The power law formulation has gained almost universal acceptance for building envelope leakage in:

- measurement standards for building envelopes, e.g., ASTM (1982), CGSB (1986) and ISO (1995),
- ventilation standards, e.g., ASHRAE 119 (1988) and 136 (1993), and
- many infiltration models.

Many of these standards and calculation procedures use the power law function to extrapolate from data measured at higher pressure differences down to the pressures experienced by the building envelope for natural infiltration. This extrapolation is only valid if the power law can be applied over a wide range of pressures and flows, and errors in extrapolation will lead to biases in predicted infiltration rates. Using a power law formulation assumes that the pressure-flow relationship is independent of flow rate. This assumption can be questioned because the traditional engineering concepts for pressure-flow relationships are based on Reynolds number dependence, where the relationship between flow and pressure drop depends on the flowrate. A justification for neglecting Reynolds number dependence is that building leaks tend to be short in the direction of flow, and therefore the pressure-flow relationship is dominated by entry, developing flow and exit losses. We will examine laboratory measurements on systems dominated by these factors to see if the result can be adequately represented by a power law. Theoretical considerations will also be presented to show why the flow in building leaks can be represented by a power law.

In addition to crack flow measurements performed by other authors, the particular experiments performed for this study examined flow through building envelopes with special attention paid to measurements made at low pressure differences and flowrates. These low pressure difference tests are needed because routine pressurisation tests on houses cover a range of 15 to 50 Pa for CGSB (1986) tests and 12.5 to 75 Pa for ASTM (1982) tests. The lower limits of these ranges are due to the resolution limits of commonly used pressure transducers and the effects of fluctuating wind pressures that can cause a great deal of scatter as shown by Modera and Wilson (1989). Data taken during these tests must be extrapolated below 10 Pascals to estimate the leakage function in the 1 to 10 Pascal range typical of residential building infiltration conditions. This paper will examine how well the power law and quadratic functions can be extrapolated successfully to lower pressures and how flow through individual leaks combine when determining whole building envelope flows. Test results will be presented for whole house pressurisation at low temperature differences and windspeeds required to reveal the low pressure leakage function.

3 Pressure - flow relationships for crack flow

3.1 Quadratic form

The pressure-flow relationships for fully developed laminar flow and turbulent (orifice) flow give the limiting cases for crack flow where:
For laminar flow and for turbulent flow

$$Q = K_1 \Delta P \,, Q = K_2 \Delta P^{\frac{1}{2}} \qquad \textbf{1, 2}$$

where Q = Flowrate [m^3/s], ΔP = Pressure difference across crack [Pa], K_1 [m^3/sPa] and K_2 [$m^3/sPa^{0.5}$] are flow coefficients. Equation 2, for fully turbulent orifice flow, has been used often in ventilation modelling, beginning with Shaw (1907) and still in use today (ASHRAE Chapter 23 (1989)). The laminar and turbulent equations can be combined into a quadratic form (as used by Etheridge (1977)) such that,

$$\Delta P = AQ + BQ^2 \qquad \textbf{3}$$

where A [(Pa s)/m^3] is the flow coefficient for fully developed laminar friction losses and B [(Pa s^2)/m^6] is the coefficient for entry, exit and turbulent flow losses. Unfortunately Equation 3 gives the pressure drop for a known flowrate. For ventilation studies a correlation is needed to give flowrate as a function of the applied pressure difference due to wind, stack and mechanical ventilation effects. Equation 3 can be expressed in a more useful form as

$$Q = \frac{-A + \sqrt{A^2 + 4B\Delta P}}{2B} \qquad \textbf{4}$$

In Equation 4 only the positive root is required because all real flows are positive.
Baker et al. (1987) used standard fluid mechanics principles for flow between parallel plates to determine A and B, such that

$$A = \frac{12\mu z}{Ld^3}, \quad B = \frac{\rho Y}{2d^2L^2} \qquad \textbf{5, 6}$$

where m [Kg/ms] is dynamic viscosity, L [m] is the width of the crack, d [m] is the gap thickness, z [m] is the distance in flow direction (crack length), ρ [Kg/m^3] is the fluid density and Y is a factor that depends on the crack geometry. For example, Baker et al., used empirically determined values of Y = 1.5 for a straight crack, 2.5 for an L-shaped crack and 3.5 for a double bend crack. The predictions for A and B were compared to measured data by Baker et. al. for various crack geometries with errors typically less than 20%. Baker et. al. found that values of A and B fitted by least squares to Equation 4 gave a better fit than the theoretical values to their measured crack flow data for some simple crack geometries.

For flow in pipes, Sherman (1992) has summarised the work of previous authors (Boussinesq (1891), Schiller and Agnew (1922) and Langhaar (1942)) on linearized Navier-Stokes equations to estimate A and B as:

$$A = \frac{128\mu z}{\pi D^4}, \quad B = \frac{8m\rho}{\pi^2 D^4} \qquad \mathbf{7, 8}$$

where D is the pipe diameter, and m is a factor to account for the linearization of the Navier-Stokes equations.

The quadratic equation allows the flow to vary from laminar to turbulent over a range of flowrates. However, this equation is based on combining fully developed laminar and turbulent flows and entry and exit losses. This can be physically unrealistic for the convoluted crack geometries typical of building leaks in which the flow is rarely fully developed because the flow has to begin its development after each sharp change of direction.

3.2 Power law form

The power law relationship has the form

$$Q = C\Delta P^n \qquad \mathbf{9}$$

where C [m^3/sPa^n] is the flow coefficient and n is the flow exponent. The flow exponent has the limiting values of 0.5 and 1 for fully developed turbulent and laminar flows respectively. Sherman (1992) has developed a non-dimensionalized pressure that relates the ratio of total pressure drop to the critical pressure drop that occurs when the pressure drop due to fully developed laminar flow is equal to the pressure drop form combined entry, exit and flow acceleration effects. This parameter, S, has been related to the power law exponent, n , which allows the power law exponent to be related to the crack geometry, such that

$$S = \frac{1}{8}\frac{(1-n)n}{\left(n-\frac{1}{2}\right)^2} = \frac{\Delta P}{\frac{512\pi^2\mu z^2}{m\rho A^2}} \qquad \mathbf{10}$$

where A is the cross sectional area of the crack. The flow can then be expressed as a function of S:

$$Q = \frac{16\pi\nu z}{m}\phi S^n \qquad \mathbf{11}$$

where ν is the kinematic viscosity and ϕ is a power law factor depending on the exponent, n.

4 Developing flow for a single crack

Given typical building crack geometries and flowrates the flow in building leaks is likely to be developing flow. Some researchers suggest that the flow exponent, n, is constant over a wide range of flowrates and pressure differences for cracks similar in geometry to building leaks. Shapiro, Siegel and Kline (1954) studied laminar flow in the entrance region of smooth circular tubes. W. Jones of Ontario Hydro proposed in a letter in 1987 that the results of Shapiro, Siegel and Kline imply an exponent of n = 2/3 for this entrance region developing flow regime. This is also a typical value for n found from pressurisation testing of houses. Although tempting, this does not prove that flow in cracks in building envelopes is undeveloped laminar flow because the developing flow regime studied by Shapiro et. al. is only dominant over an entry length of less than one diameter. It remains an intriguing coincidence, however, and requires further research. Honma's (1975) experiments on parallel flat plates showed that n is constant over a very wide range of flowrates and pressures for a given crack geometry. Honma performed tests from 1 to 50 Pa, encompassing the typical values experienced by a building envelope.

Other work has found that the power law exponent, n, may vary with flowrate. Kreith and Eisenstadt (1957) tested circular capillary tubes with length to diameter (aspect) ratios ranging from 0.45 to 17.25, and found that n depends on aspect ratio for laminar flow where $Re_D < 2000$ (Re_D is Reynolds number based in tube diameter, D). Most building leakage sites fall into this category. For example, a 1 mm diameter crack with orifice type flow will have a $Re_D \approx 85$ for 1 Pa pressure drop and $Re_D \approx 400$ for 10 Pa pressure difference. Kreith and Eisenstadt's measurements showed that at high aspect ratios the flow became more laminar and n approached 1, while at low aspect ratios the entrance effects were more dominant and n approached 1/2.

5 Flow through arrays of cracks

Previous work by Baker et. al. (1987), Honma (1975) and several other researchers has concentrated on flow through an individual crack or cracks in series. In a real building the total leakage is the sum of many individual cracks of differing flow characteristics in **series and parallel** with each other that are distributed over the building envelope.

5.1 Parallel Cracks

The flow may be modelled as an array of cracks in parallel. For laminar flows

$$\Delta P_L = R_L Q_L \qquad \textbf{12}$$

where ΔP_L is the pressure drop across the laminar flow crack, R_L is the flow resistance and Q_L is the flowrate. Similarly, for orifice like cracks

$$\Delta P_O = R_O^2 Q_O^2 \qquad \textbf{13}$$

where ΔP_O is the pressure drop across the orifice flow crack, R_O is the flow resistance and Q_O is the flowrate. For cracks in parallel an electrical analogy is to have the flow resistances in parallel such that

$$Q_{total} = Q_L + Q_O \text{ and } \Delta P_{total} = \Delta P_L = \Delta P_O \qquad \textbf{14, 15}$$

Substituting Equations 12 and 13 in Equation 14 and using Equation 15 gives

$$Q_{total} = \frac{\Delta P_{total}}{R_L} + \frac{\Delta P_{total}^{1/2}}{R_O} \quad \mathbf{16}$$

Equation 16 expresses the relationship between total flow and total pressure drop in terms of combined laminar and orifice type leaks in parallel.

5.2 Series cracks

This flow is equivalent to inlet and exit turbulent flow losses in series with fully developed laminar flow. This is the same as the quadratic flow discussed earlier and advocated by Etheridge (1977) and others. The laminar and orifice type flows are described by Equations 12 and 13. In this case the flows are the same and the pressures add so that

$$Q_{total} = Q_L + Q_O \quad \text{and} \quad \Delta P_{total} = \Delta P_L + \Delta P_O \quad \mathbf{17, 18}$$

and the pressure drop can be written in terms of the two types of flow

$$\Delta P_{total} = R_L Q_L + R_O^2 Q_O^2 \quad \mathbf{19}$$

Equation 19 can realistically only be applied to a single crack whereas Equation 16 can be applied to an array of cracks.

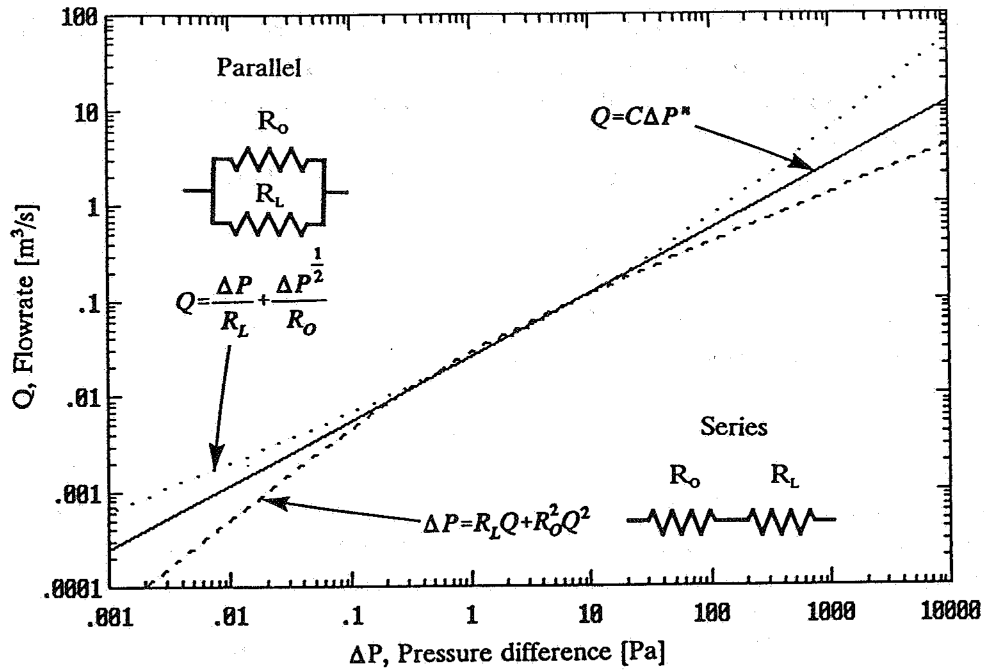

Figure 1 Illustration of series, parallel and power law crack flow model behaviour.

The different behaviour of power law, series resistance and parallel resistance crack flow equations is shown in Figure 1. The logarithm of pressure and flowrate are plotted in Figure 1 to better distinguish between the different equations. The power law equation plotted in Figure

1 appears as a straight line with a constant slope due to its constant exponent (in this example the exponent value was chosen to be n = 2/3). R_O and R_L for the resistance crack flow equations were found by fitting to the power law relationship at 1 Pa and 10 Pa as this is the typical pressure range experience by building envelopes due to natural wind and stack effects. For the parallel cracks $R_O = 51.0$ and $R_L = 184.9$ and for the series crack resistances $R_O = 24.65$ and $R_L = 19.45$. Figure 1 shows how the series cracks become more like laminar flow (slope = 1 on log-log plot) at low flow rates and orifice flow (slope = 0.5) at higher flowrates. For parallel cracks the reverse is true with orifice flow dominating at low flow rates and laminar flow at higher flowrates. Over the range of interest for ventilation (1 Pa to 10 Pa) there is very little difference between the three methods. This is partly because all three methods were chosen to be equal at 1 Pa and 10 Pa. If the methods had been equated over a different range larger differences over the range of interest would be observed.

The relationships illustrated in Figure 1 show that a combination of series and parallel leaks in an experiment may result in a pressure-flow relationship that fits a power law type equation even though the dominant flow regimes in each individual leak may change over the range of experimental pressures and flow rates.

6 Fan pressurisation testing

In a real building there are cracks of many geometries that include both series and parallel leaks. To determine which crack flow method is the best for describing real building, leakage experiments have been performed on full size buildings using the method of fan pressurisation testing. The buildings were tested with the large holes (e.g., furnaces flues) sealed to observe pressurisation test results for arrays of parallel an series cracks. The tests were repeated with flues open to look at combining the small cracks in the building envelope with large holes.

Standard methods for fan pressurisation exist in Canada (CGSB (1986)) and the U.S.A. (ASTM (1982)). Both standards have recommended values for the pressure differences at which to take measurements. These pressure differences cover a range of 15 to 50 Pa for CGSB tests and 12.5 to 75 Pa for ASTM tests. Most of the time the actual pressures caused by wind and temperature difference (stack) effects on a building will be less than 10 Pa. It is a fair question to ask if test results from high pressures may be extrapolated to the lower pressures that a building envelope usually experiences, because at lower flowrates the flow characteristics of the leaks may be different. This would imply that a different flow coefficient, C, and flow exponent, n, apply at the low pressures a building experiences due to natural conditions than at the elevated pressures of a fan pressurisation test. The reason for not testing at lower pressures is that the pressures due to wind and stack effects are of the same magnitude as those that the blower system is attempting to superpose on the house, and fluctuations in the wind pressures on the house create scatter in the test data. In addition, the pressure test assumes that there is the same pressure difference across each leak during the test and with significant wind or stack pressures this is no longer true.

For this study fan pressurisation tests were conducted at the Alberta Home Heating Research Facility (AHHRF) located south of Edmonton, Alberta, Canada, on open agricultural terrain. The facility consists of six test houses, each constructed in a different way in order to examine different heating and ventilating strategies. The houses were unoccupied and the fan pressurisation test system was automated, which allowed over 5,000 fan pressurisation tests to be performed between 1983 and 1988. Windspeed, wind direction, and ambient temperature data were taken from meteorological towers at the test site.

The flowrates were measured using a laminar element flowmeter, and were corrected for pressure and temperature variation, and the effects of fan loading on the laminar element flowmeter at high fan speeds and low flowrates (which occur in a "tight" house). Pressure and flowrate measurements were taken over 15 seconds and averaged for each data point.

The indoor-outdoor pressure difference was measured using a pressure averaging manifold that had a pressure tap on each wall of the building. This manifold was used to reduce the wind pressure effects. Offset pressures due to stack and wind effects with the fan not in operation were measured at every data point. A damper was closed over the fan opening for each offset reading because the fan opening can change the pressure distribution of the building significantly.

Figure 2 shows the results of a typical test in a house with very little envelope leakage with and without an open 15 cm diameter furnace flue with a 7.5 cm diameter orifice at the bottom (House #1 at AHHRF). Because the leakage of this house is small without the flue, the value of $C = 0.0028$ is lower and $n = 0.73$ is higher than for a typical house (C is less than for a furnace flue). The open furnace flue increases the total leakage so that $C = 0.085$. Note that one cannot simply add the values of leakage coefficient for a house with the flue closed and for a flue to obtain the leakage coefficient for a house with the flue open unless their flow exponents are the same. The value of flow exponent ($n = 0.56$) with the flue open is lower than with the flue closed because the flue flow exponent is about 1/2 (as shown by the laboratory measurements), and performing a test with the flue open will bring the value of the flow exponent for the whole building closer to 1/2.

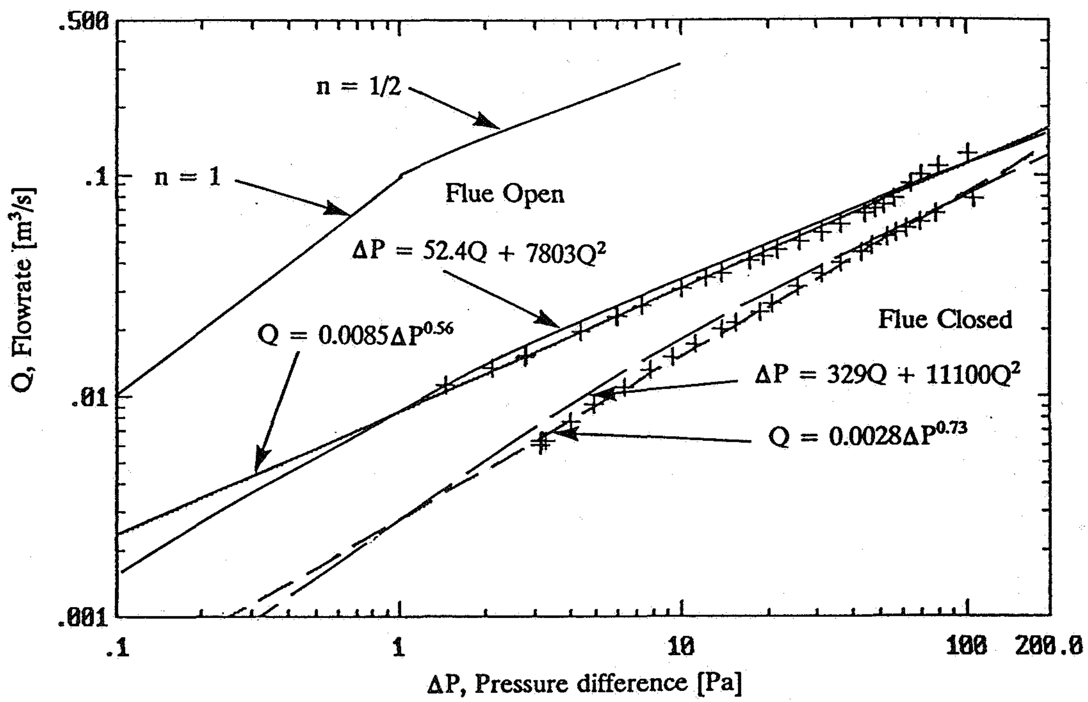

Figure 2 Blower test results for a brick walled house, with an open 15 cm I.D. flue with a 7.5 cm diameter orifice at the bottom, and with a blocked flue. Windspeed < 1.0 m/s and $\Delta T < 10°C$.

Figure 3 shows the results of a test performed in one of the test houses with a wood frame and single stud walls (House #2 at AHHRF with an open 15 cm diameter furnace flue with a 7.5 cm diameter orifice at the bottom). The value of C = 0.012 is smaller than for an average home because the test houses are not very large (50 m^2 floor plan bungalows), but the value of n = 0.66 is typical of tests performed elsewhere. A curve showing the quadratic leakage function is also shown in Figures 2 and 3. The quadratic was fitted at 1 and 100 Pa to determine A and B for Equation 3.

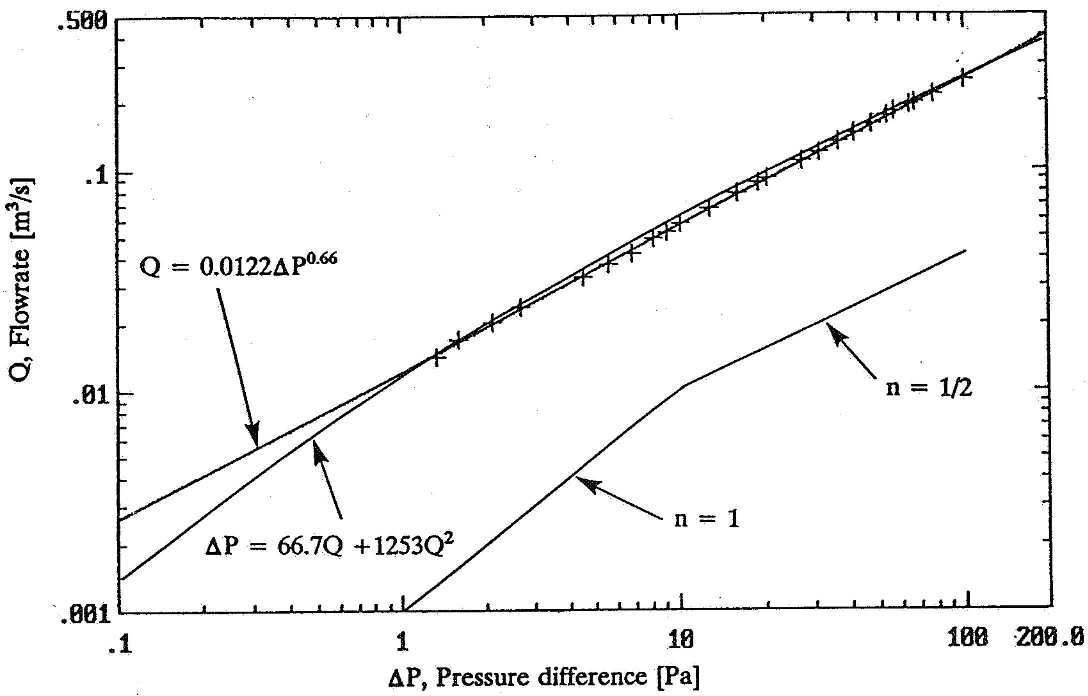

Figure 3 Blower test results for a house with single stud wood framed walls, with an open 15 cm I.D. flue with a 7.5 cm diameter orifice at the bottom. Windspeed = 0.84 m/s and ΔT = 10°C.

The results shown in Figures 2 and 3 show that the power law formulation works well for houses with an array of small cracks as well as in houses with additional large holes (n this case a furnace flue. The type of leaks depend on building construction, so for houses of similar construction a typical value of n may be specified. Examining the results of tests performed at AHHRF and other published values, n is typically 2/3.

A significant observation to be made from the results of these tests, as shown in Figures 2 and 3 is that the relationship between flowrate and pressure difference does not change over the range of values tested which include typical pressure differences driving ventilation flows. There is no observable trend towards more laminar flow at low flowrates and pressures (i.e. n approaches 1) or more turbulent flow at higher values (i.e. n approaches 0.5) or visa versa. This shows that the C and n derived from blower test results in the 10 to 50 Pa range are true constants describing the building leakage for the purposes of ventilation calculations. The power law fits the data because the leaks are relatively short and convoluted for the building envelope resulting in flows that are never fully developed, also a building envelope is a combination of parallel leaks and series leaks that when combined can result in power law behaviour. The quadratic function attempts to make the leakage function more laminar at low flow rates and more turbulent at high flow rates and this trend is not observed in the data. These results also imply that tests at higher pressures of the CGSB and ASTM standards can extrapolated to determine the leakage characteristics of a building for the pressure range that a building actually experiences.

The results of fan pressurisation testing at AHHRF are repeatable only at lower windspeeds as the amount of scatter in the data increases with windspeed. At windspeeds greater than 1.5 m/s the scatter in the data at low pressures is too great to apply a least squares procedure to find the correct C and n. At windspeeds above 1.5 m/s pressure data below 10 Pa should not be used. The data taken at higher indoor-outdoor pressure differences, as for the ASTM and CGSB standards, can be used to find C and n up to windspeeds of about 4 m/s. With less sophisticated test procedures the low windspeed limit should be lowered to yield valid results.

7 Summary

In this paper we have shown that a power law pressure-flow relationship can be developed from a theoretical background. This has been applied to measured flows and pressures in buildings with small cracks only, and to combinations of the small building envelope cracks and large holes (a furnace flue).

The following are some key points developed in this paper:

- Fundamental fluid flow theory can be used to develop power law formulations for crack flow.
- The classical view of laminar flow ($Q \propto \Delta P$) at low flows and turbulent flow ($Q \propto \Delta P^2$) at high flows is not valid for combinations of series and parallel leaks (as found in real building envelopes).
- House pressurisation tests have shown that the power law is valid over the range of pressures typically experienced by a naturally ventilated house.

These results imply that the assumption of apower law relationship used by many standards and measurement procedures is valid. In addition, extrapolation of results from tests at high pressures to those typically experienced by a building envelope does not introduce a bias in infiltration predictions.

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Improve Train Tunnels. A dynamical ventilation model.

Hans J.C. Phaff and Willem F. de Gids

TNO Building and Construction Research
P.O. Box 29
NL 2600 AA Delft
The Netherlands
Fax +31 15 608 432
Tel +31 15 608 608
J.Phaff@bouw.tno.nl

SYNOPSIS

Train tunnels and subways are an interesting field of ventilation.
Trains move air through tunnels at rates of 600 m^3/s (over 2 x 10^6 m3 per hour) which is much more than flow rates in buildings.
Air pressures can vary up to some 3000 Pa leading to air velocities in the range of 10 to 50 m/s. This can lead to unsafe situations and thermal discomfort.
The development of high speed trains causes more concern for better tunnel design.
Modern stations often house small shops and restaurants, that require lower air velocities for thermal comfort.
A dynamical ventilation model has been made to study effects of improvements.
An array of controlled fans seems to be a very effective draught remover.

The dynamical model is programmed to serve as a demo to give insight in the matter and can be shown in just a couple of minutes from a PC. A copy can be requested via e-mail to J.Phaff@bouw.tno.nl Subject: Train Demo. This demo version is just intended for demonstration purposes and must not to be used for the design of real tunnels.

1. INTRODUCTION

Train tunnels have always been the subject of calculations and optimizations. From the viewpoint of ventilation modelling, train tunnels are very interesting. Optimization of a train tunnel through a mountain is the tantalizing task for the engineer. A long tunnel low in the mountain, or first a long climb and shorter higher positioned tunnel. Long tunnels have a higher resistance, are longer and cost much more energy to drive through. The shorter high tunnel costs more energy to drive up the mountain. A wide tunnel will have less friction losses because much air can move around the train, but wider tunnels are more expensive to build.
High velocity trains in tunnels need an enormous extra amount of power to overcome the extra air resistance in a tunnel.
Unfortunately in Holland the mountain aspect is hard to find. Except for the South East appendix of the country, ground level is rarely elevated to more than a few times ten meter above sea level due to deposits in the ice ages. About half of the country is at or a few meters below the average sea level and as flat as a carefully baked pancake. Water levels are kept a few decimeter below the ground level by many ditches and a grid of canals that lead to locks that sluice the water out at low tide. This all is to prevent Holland to get back to it's medieval marsh land. In deeper polders electrical pumps remove the excess water.
In this muddy land there is also a need for tunnels and subways. One has measured that these tunnels move up and down a bit with the tide.

The subject we are looking at here is comfort in a train tunnel with a large station hall built at ground level over the subway platforms. As the tunnel is not far below ground level there are many ventilation openings to bypass the train's pressure wave. From the ventilation point of view these are very moderate tunnels. In the station hall are shops and restaurants. People sit there and expect a comfortable indoor climate. This is in contrast to stations a few years ago where every body stood, ran and waited, well dressed for weather, for their train. In winter the tunnel is below 10 °C and air comes at a moderate maximum of 5 m/s over the stair cases and person walkalators into the station hall, over the restaurant guests.

This is far from comfortable. A single entering train generates a few meter per second in the stair cases that lasts for a couple of minutes. Long enough to move out there from your restaurant seat, or start serious complaints. Even without Fanger in hand, every body will admit that this will not work long.

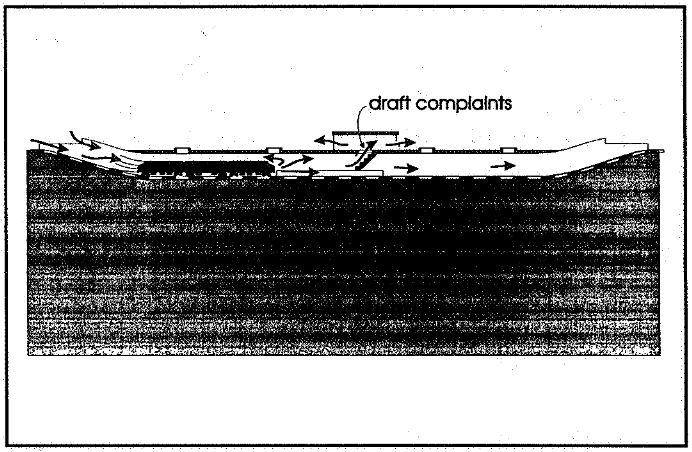

Figure 1 Tunnel with draft complaints

TNO has had a long time experience with train tunnels and models that predict the effect of design changes on energy, pressures and air velocities and start conditions for smoke movement in case of an emergency. This work has been done by dr. ir. H.B. Bouwman and was not published. This knowledge has been picked up again and converted in a set of PC programs.

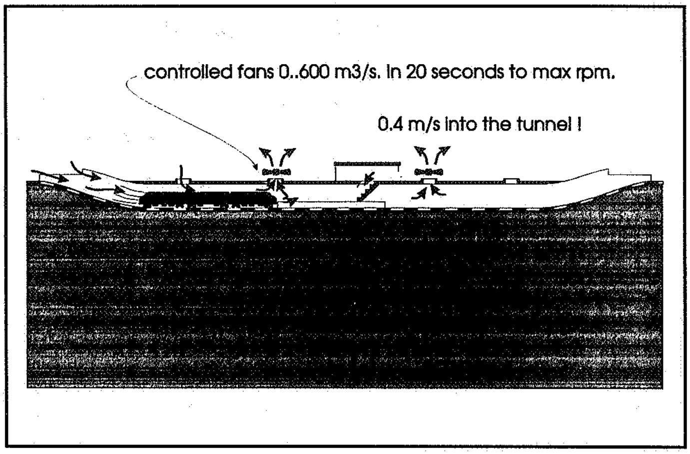

Figure 2 Tunnel with fans.

2. METHOD

The model is basically a dynamical bernoulli equation and the normal technical transport equations through ducts with side branches and air flow around bodies, mainly by a correct use of hydraulical diameters. The tunnel is subdivided in sections. Every section may have a ventilation duct to the outside. The train is modelled with its friction loss at the front, side and rear. A special concern is the air velocity along the train. The network of tunnel segments and side branches is numerically solved but the pressure losses per section are mostly analytical solutions. A very complex task for the program is

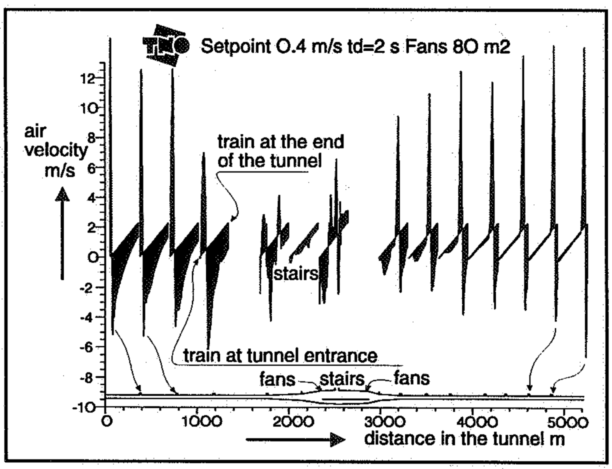

Figure 3 Simulated air velocities in the ventilation openings.

the administration of the movement of the train and subsequently the changing lengths occupied by the train while passing the sections and the many ventilation openings.
The model allows for any train, any shape of tunnel and ventilation openings. The train can be operated with a given speed profile in time or given brake characteristics and acceleration power.
The tunnel can also be operated with wind pressures at the openings, or a train can start just after or during the passage of another train.

3. DEMO

The DEMO is a full version of the actual program. It's input is limited to only a few variations and changes of parameters. The dimensions of train, tunnel and ventilation openings are fixed. What can be changed is:

1 a multiplication factor for the velocity profile of the train
2 the train can stop or drive through at constant speed
3 you can select 3 runs:
  - run 1 original situation
  - run 2 extra ventilation openings before and after the platforms
  - run 3 the TNO solution. Mount 20 to 40 axial low pressure fans of some 1.5 to 2 m diameter max 10 m/s air velocity in the fans or 20 m/s at smaller fans.

At the end of one run you must ESC the program and run again for a next setup.
The program explains which keys are active, but any key will move you to the next graph.
Pressing ESC will give you the possibility to quit the program, even in the middle of a graph.
At the end of each run a small text file is created with some explanation. All graphs are written to HPGL files on disc. The program will thus run faster on hard disk in a separate directory.

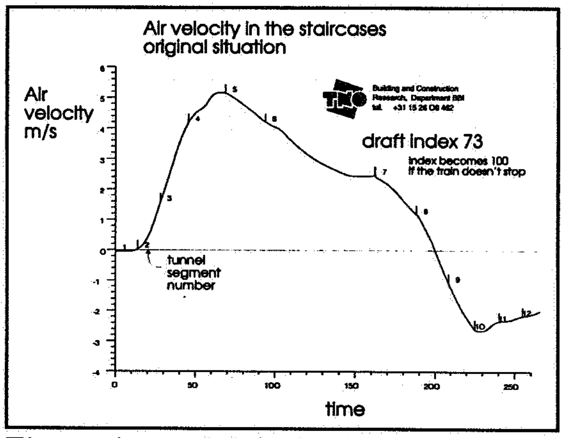

Figure 4 Original situation at the stairs. Run 1.

Run 2 gives a reduction of the draft, but does not eliminate it at high costs of the large ventilation shafts. Run 3 can fully eliminate any draft. It can even make sure there is a small air velocity from the station into the tunnel. Fans can be payed from saved costs for the smaller shafts.
The operation costs are despite the large flowrates, very low as the pressure differences across the fans is very low.

As the fouling with railway dust can be absolutely prevented with the fans, this saves much more than the electricity costs to run the fans.
The controller for the fans is very simple and stable. Time from low to full fan speed is

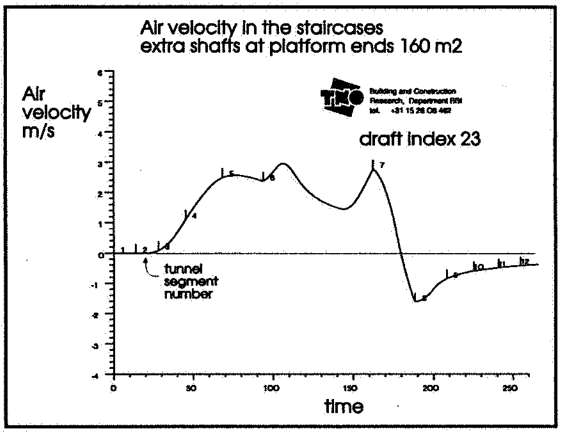

Figure 5 Extra shafts. Still draft complaints.

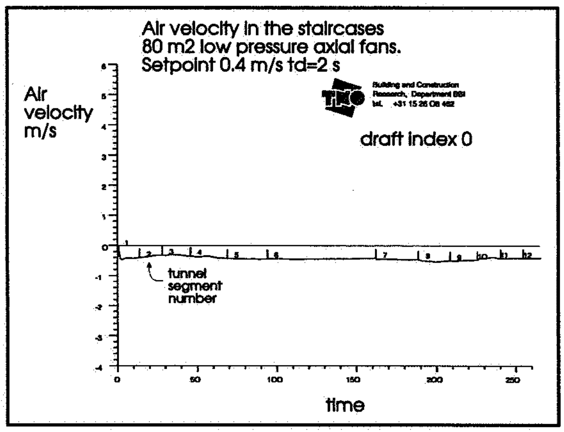

Figure 6 TNO solution, controlled fans. No draft.

some 30 seconds and for these fan sizes this is not any problem. There is no need for adjustable pitch of the blades of the fans.

Besides these luxury things like draft complaints the model also calculates the extra energy for the train to drive through the tunnel and the air pressures and pressure changes in the tunnel and around the train.

4. CONCLUSION

This tunnel program can give an insight in air velocities in tunnels and the influence of variations in the design. It is intended to find the optimum ventilation and airflow control in tunnels. With the aid of the program solutions can be tested and ideas for new approaches will be found.

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

17th AIVC Conference, Göteborg, Sweden, 17-20 September, 1996

Multi-Zone Calculations and Measurements of Air Flows in Dwellings

Åke Blomsterberg
Thomas Carlsson
Charlotte Svensson

Swedish National Testing and Research Institute
Box 857
S-50115 Borås
Sweden

Swedish National Testing and Research Institute
Box 857
S-50115 Borås
Sweden

J&W Consulting Engineers
Slagthuset
S-21120 Malmö
Sweden

Synopsis

A study of the reliability of systems by considering the ability of different systems to maintain a required air flow rate over time is included in a subtask of IEA Annex 27 "Evaluation and Demonstration of Domestic Ventilation Systems". Measurements and calculations were performed to determine the variation in ventilation rates due to variation in climate and variation in performance of the ventilation system. Dwellings with passive stack, mechanical exhaust and mechanical exhaust-supply ventilation, representative of the Swedish housing stock, were studied.

Diagnostic tests were carried out, to discover if the installed ventilation system was functioning as designed and to determine certain values e. g. characteristics of inlets and outlets. The airtightness was tested. The air flows in mechanical ventilation were measured. The continuous monitoring included measurements in dwellings of overall and local (individual rooms) ventilation rates, and measurements of boundary conditions, during three different periods. The ventilation rates were monitored using tracer gas; passive techniques for monthly averaging and constant concentration for hourly averaging. The measured ventilation rates were simulated using COMIS, a multi-zone air flow network model. The simulated and measured average total outdoor ventilation rates agree reasonably well, while there can be disagreement for hourly rates and individual rooms. This paper presents and discusses the measurements and the calculations and compares the two.

1. INTRODUCTION

The overall scope of IEA Annex 27 "Evaluation and Demonstration of Domestic Ventilation Systems" is to establish a general evaluation tool, which makes it possible to pre-evaluate the overall performance of different ventilation systems for domestic buildings in different climates. A number of performance criteria are dealt with within the annex. They include e. g. air quality, thermal comfort, energy, noise, life-cycle costs, moisture and reliability. The Swedish part of the research in the annex covers the reliability aspect of domestic ventilation, i. e. the ability of different systems to maintain a required flow rate over time. The work is divided into:

1 numerical simulation of ventilation in typical dwellings
2 measurements in representative dwellings
3 numerical simulation of measured dwellings and comparison with the measurements
4 development of a design tool for determining the reliabilty of a ventilation system
5 application of the developed design tool on typical dwellings.

This paper presents results from phase 2 and 3, which were carried out during 1995.

2. THE DWELLINGS TESTED

The dwellings which were examined in this project represent typical Swedish buildings. They are representative as to building technology, size of the building and ventilation system. Important criteria, when choosing the dwellings, were type of ventilation, year of construction, and number of storeys (see table 2.1).

Table 2.1 The tested buildings.

Ventilation system	Year of construction	Storeys	Floor area, m²	Remark
Apartment building				
Balanced	1988	4	120, 114	Ground, top floor
Exhaust	1990	3	58, 50	Ground, top floor
Passive stack	1955	3	56, 55	Ground, top floor
One-family house				
Balanced	1991	1½	128	Crawl-space
Exhaust	1976	1½	160	Slab on grade
Passive stack	1958	1	114	Basement

The dwellings with passive stack ventilation have exhaust air terminal devices in bathrooms, kitchens and laundry-rooms and sometimes outdoor air supply to the other rooms through outdoor air vents near windows. Air is exhausted to the outside through vertical shafts.

The dwellings with exhaust fan ventilation have exhaust air terminal devices in bathrooms, kitchens and laundry-rooms and outdoor air supply to the other rooms through outdoor air vents near windows. The tested one-family house has no outdoor air vents.

The dwellings with balanced ventilation have exhaust and supply ventilation. Air is exhausted from rooms such as bathrooms, kitchens and laundry-rooms and air is mainly blown into bedrooms and living-rooms.

3. METHODS

3.1 Measurements

The measurements were started with diagnostic tests to discover if the installed systems were functioning as designed and to determine certain values:
- pressurization in order to determine the irtightness of the building envelope (Blomsterberg 1990) and the leakage area of the outdoor air vents and the passive stacks. The estimated inaccuracy in the measured airtightnes is ± 10 %.
- IR-scan in order to determine the location of the leakage paths
- measurements in order to determine air flows in ducts of mechanical ventilation systems. These air flows were measured at the air terminal devices. The estimated inaccuracy for the measured exhaust air flows is ± 5 % + 0.5 l/s and for the measured supply air flows ± 7 % + 0.5 l/s.

The actual monitoring phase (during a winter, spring/fall and summer period) included:
- short-term monitoring using constant concentration tracer gas during 1 - 6 days to determine hourly variations in ventilation rates in unoccupied dwellings. The supply of outdoor air to several individual rooms simultaneously is obtained directly (Blomsterberg 1990). The estimated inaccuracy in the measured outdoor air ventilation rate is ± 8 % + 0.5 l/s.
- long-term monitoring using passive tracer gas in order to determine monthly averages of ventilation rates in occupied dwellings. Two different tracer gases were employed in order to be able to determine the ventilation rates for the entire dwelling and at least one bedroom. A

homogeneous emission technique was used (Stymne 1994). The estimated inaccuracy in the measured outdoor air ventilation rate is between ± 5 % and ± 15 %.
- measurements of the outdoor temperature at the site and the indoor temperature in at least two rooms. The temperatures were recorded using thermistors connected to one-channel dataloggers. Hourly values were stored. The estimated inaccuaracy is ± 0.5 K.
- measurements of the wind speed and direction at a nearby weather station. The wind data was stored as hourly averages.

3.2 Simulations - input

All simulations of air flows during the monitoring phases were made hour-by-hour using COMIS (Feustel 1995), a multi-zone network model.

Average pressure coefficients were determined using values from windtunnel measurements (Orme 1994). Each facade was given an average value, as it is very difficult to determine pressure coefficients for individual openings. The shielding conditions in different directions were considered when chosing the pressure coefficients. The ceiling was given the pressure coefficient 0.0, as being well shielded by the roof. For the passive stack ventilated dwellings, the pressure coefficient for the top of the shaft was assumed to be -0.3. Measured wind direction and velocity were used. Local wind conditions were calculated by COMIS, taking building height and surrounding buildings into account.

The dwellings were divided into different zones where each room is one zone. Measured indoor temperatures were used. Most doors were closed. The leakage through an open door was calulated using the standard equation for a non-ideal sharp-edged orifice. For closed doors the geometry of the crackage round the doors was measured. Using the basic equations for fluid flow (Kronvall 1980) through ducts and obstructions a power function was determined describing the air flow through the closed door. The interior walls and intermediate floors were assumed to be airtight.

The air leakage of the building envelope, based on the results from a fan pressurization/depressurization test of the entire dwelling, was described with a power function. The average value of the pressurization and depressurization test is used, as the leakage paths are likely to experience positive and negative pressure differences.

The result of the IR-scan and the knowledge of the used construction technique was used to determine the location of the leakage paths. Firstly the overall leakage was distributed to the facades, the ceiling and the floor according to each components area. For some dwellings the overall leakage of an individual room was increased based on the result of the IR-scan. Secondly on each facade 50 % of the air leakage was assumed to be located 0.6 m above the floor (representing cracks between the floor and the wall, and windows), and 50 % 1.8 m above the floor (representing windows and cracks between the ceiling and the floor). All leakage paths were given the same air flow exponent i.e. the one determined by the airtightness test.

The mechanical ventilation systems themselves weren't simulated, as the main interest was to study the interaction between air flows and the building envelope. For each air terminal device a fixed flow was given based on measurements. For the dwellings with passive stack ventilation, the stacks were simulated as zones with openings in the bottom and at the top. The flow resistance of the duct part of the passive stacks was included as a single resistance. The characteristics of the exhaust air terminal devices in the passive stack systems, were taken from data supplied by the manufacturer and from laboratory measurements. Airing was calculated by COMIS.

4. RESULTS

4.1 Airtightness

The average airtightness of the tested apartments is 2.3 ach, compared with 6.0 for the tested one-family houses. Most of the tested buildings are fairly representative for their year of construction, with the exception for the apartments with passive stack and exhaust ventilation. The tested apartments with passive stack ventilation are much tighter and the ones with exhaust ventilation leakier (Blomsterberg 1995).

4.2 Ventilation rates - short-term

4.2.1 Passive stack ventilation

The average total outdoor air ventilation rates for the measuring periods show a reasonable agreement between simulations and measurements (see figure 4.1). The simulation is very close to the measurement for the one-family house. The underprediction is 1 %, while for the apartments the differences are larger.

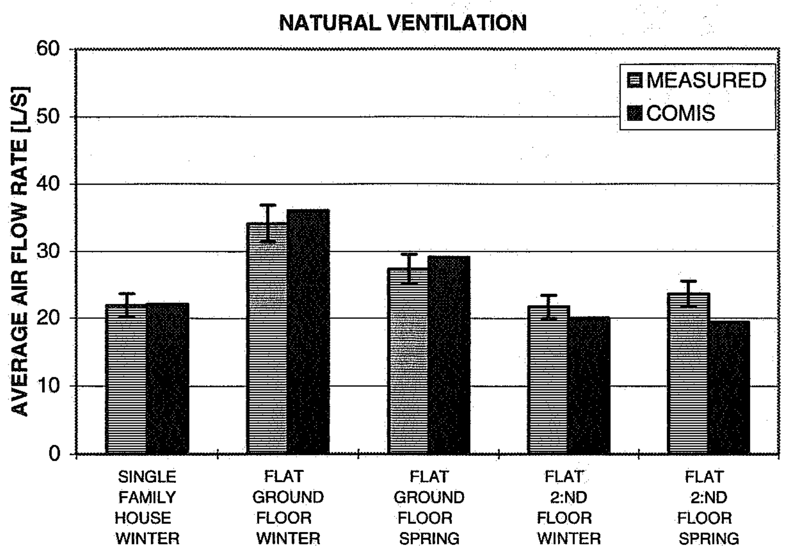

Figure 4.1 Comparison between measured and predicted average total outdoor air ventilation rates in the passive stack ventilated dwellings.

The overall ventilation rate varies over time. This is especially true for the winter measurements in one of the apartments, where the total outdoor air ventilation varies between 27 l/s (0.6 ach) and 52 l/s (1.4 ach) (see figure 4.2). The simulations indicate a temperature dominated ventilation, while the measurements indicate a wind dominated ventilation. Simulations and measurements of the dwellings show hourly variations in the total ventilation rate of the same magnitude, but not always at the same time.

The outdoor air ventilation rates are very different for different rooms (see figure 4.2 - 4.4)

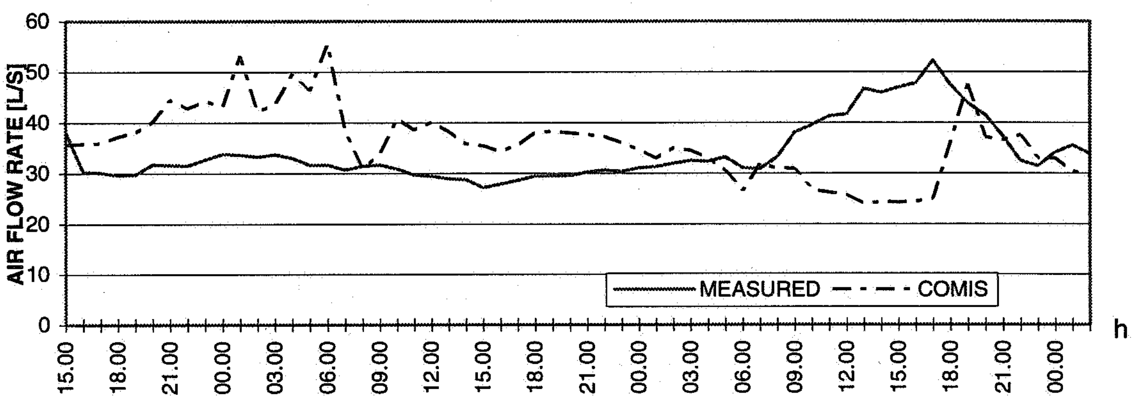

Figure 4.2a Measured and predicted outdoor air ventilation rates (total) for the passive stack ventilated apartment on the ground floor.

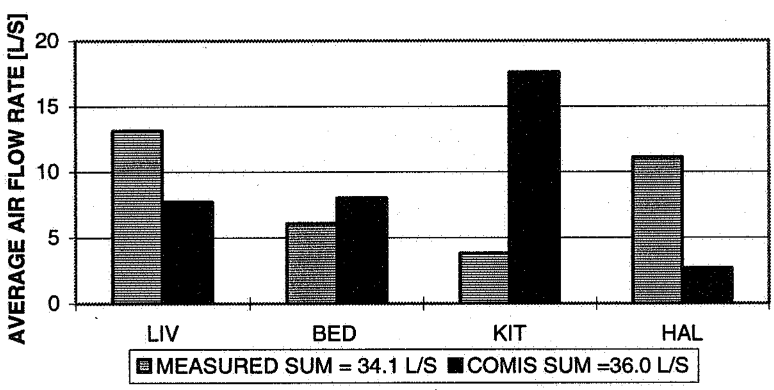

Figure 4.2b Measured and predicted outdoor air ventilation rates (individual rooms) for the passive stack ventilated apartment on the ground floor.

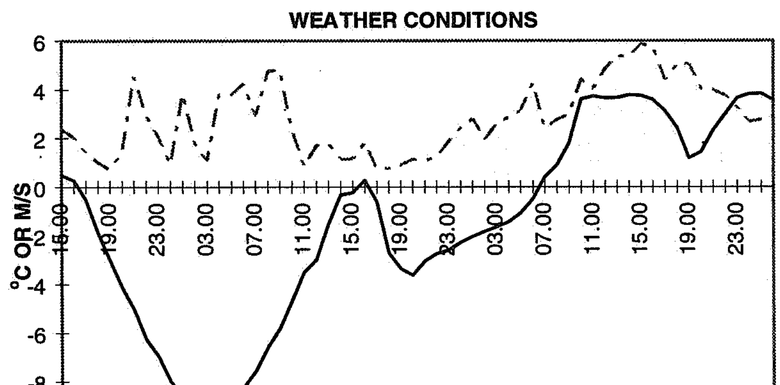

Figure 4.2c The weather for the measured and predicted outdoor air ventilation rates for the passive stack ventilated apartment on the ground floor.

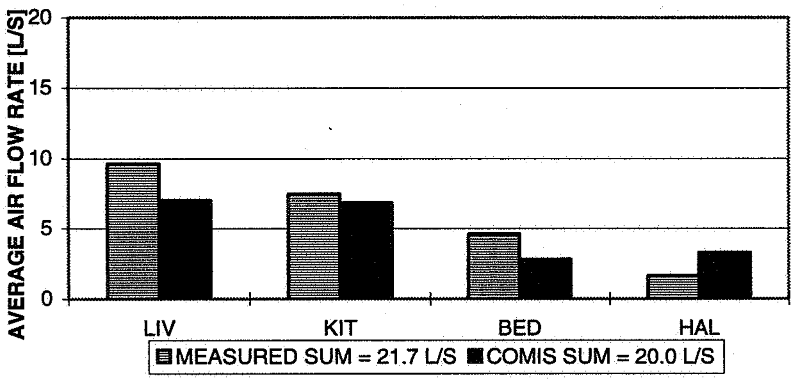

Figure 4.3a Measured and predicted outdoor air ventilation rates (individual rooms) for the passive stack ventilated apartment on the second floor.

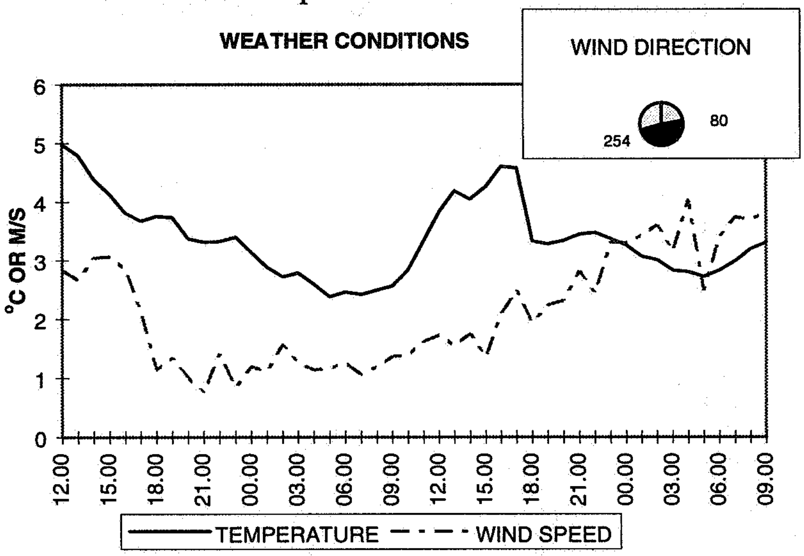

Figure 4.3b The weather for the measured and predicted outdoor air ventilation rates for the passive stack ventilated apartment on the second floor.

and is e. g. too low (< 2 x 4 l/(s and person)) in the bedrooms of the one-family house (see figure 4.4). The disagreement between predictions and measurements is fairly large for some individual rooms e.g. the kitchen in the apartment on the ground floor.

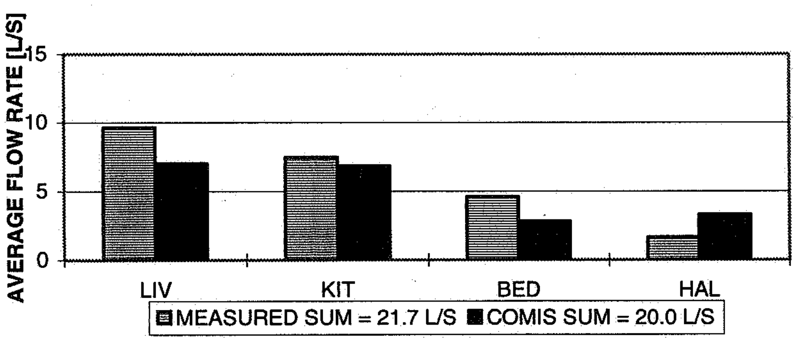

Figure 4.4a Measured and predicted outdoor air ventilation rates (individual rooms) for the passive stack ventilated one-family house.

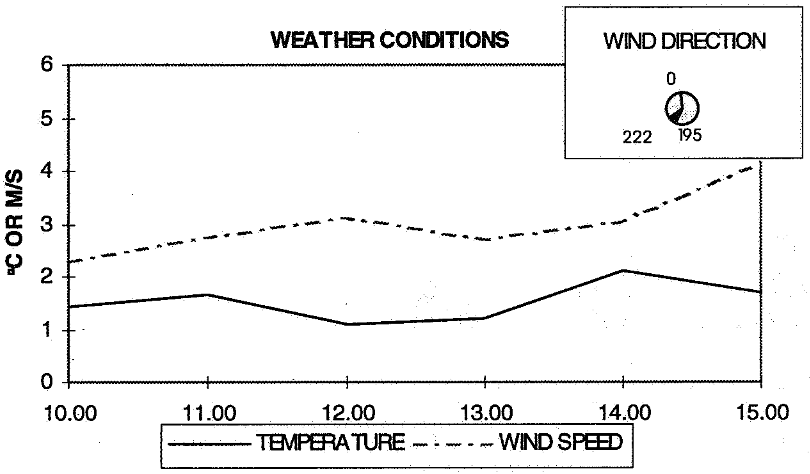

Figure 4.4b The weather for the measured and predicted outdoor air ventilation rates for the passive stack ventilated one-family house.

4.2.2 Exhaust ventilation

The agreement between simulations and measurements of the average total outdoor air ventilation rates for the measuring periods is good (see figure 4.5). The predictions vary between an overprediction of 8 % and an underprediction of 8 %.

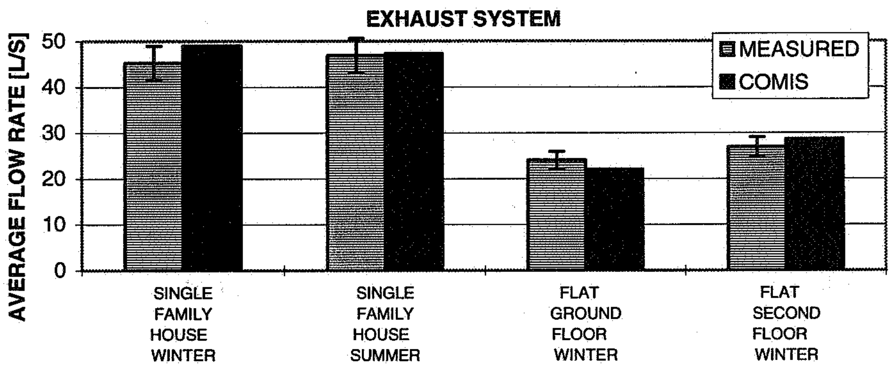

Figure 4.5 Comparison between measured and predicted average total outdoor air ventilation rates in the exhaust fan ventilated dwellings.

The continuous measurements of the overall outdoor air ventilation in two apartments show some variation over time in the ventilation rate (see figure 4.6). For the apartment shown this is probably due to fairly leaky exterior walls. The average measured ventilation rate was 27 l/s (0.75 ach). Both predictions and measurements show a slight variation in total ventilation rate. It is difficult to determine whether the variations are correlated to each other or not.

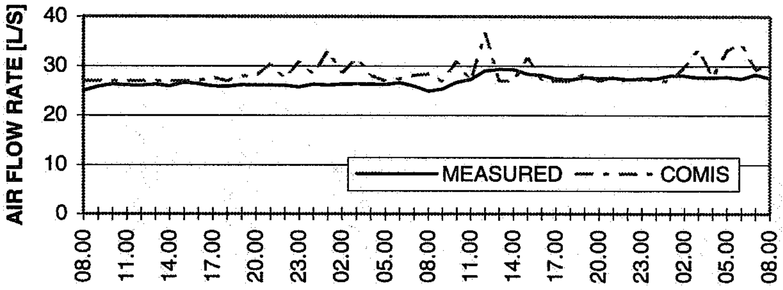

Figure 4.6a Measured and predicted outdoor air ventilation rates (total) for the exhaust fan ventilated apartment on the second floor.

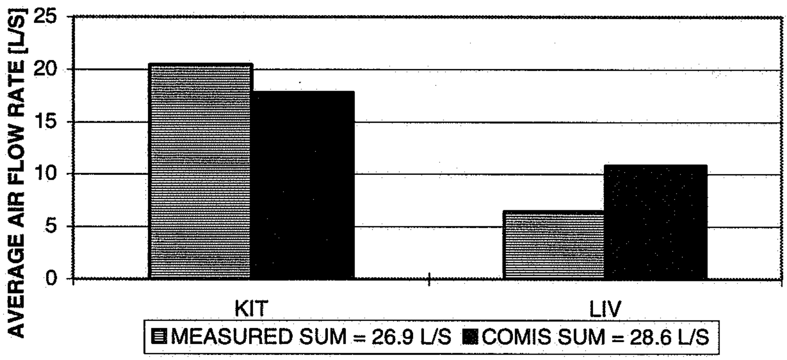

Figure 4.6b Measured and predicted outdoor air ventilation rates (individual rooms) for the exhaust fan ventilated apartment on the second floor.

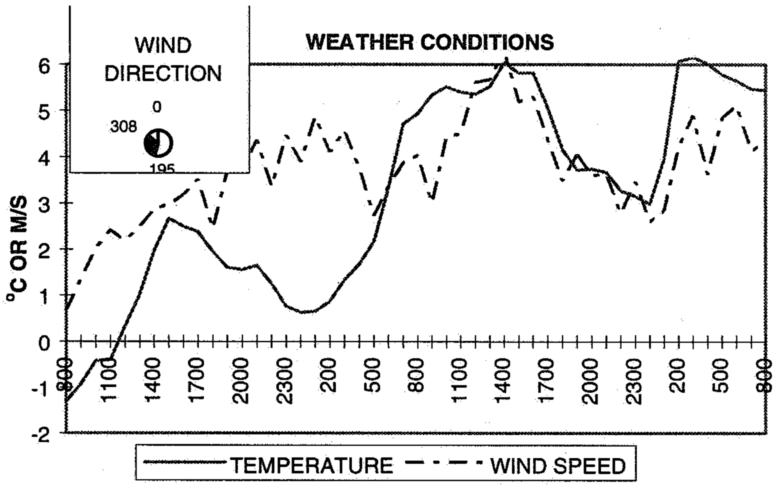

Figure 4.6c The weather for the measured and predicted outdoor air ventilation rates for the exhaust fan ventilated apartment on the second floor.

The outdoor air ventilation rates of individual rooms show some rooms to have too low (< 4 l/(s and person)) a ventilation rate e. g. bedroom 3 and 4 upstairs in the one-family house (see figure 4.7) and some rooms to have too high a ventilation rate e.g the kitchen in the apartment on the second floor (see figure 4.6). The agreement between predicted and measured ventilation rates is for most rooms good.

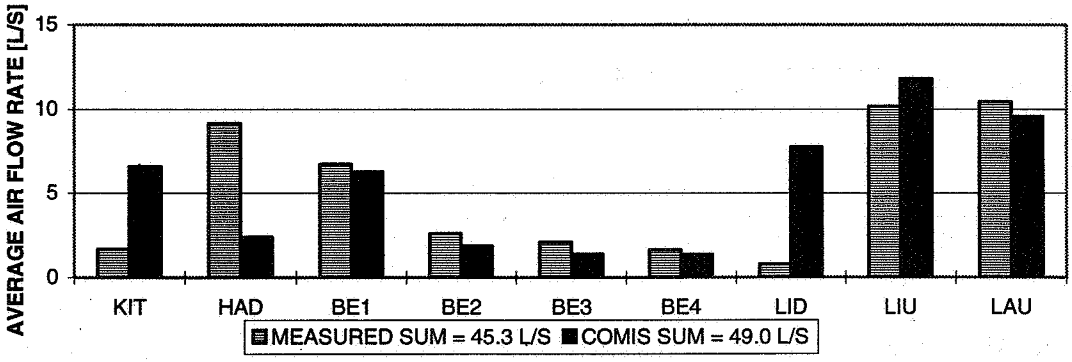

Figure 4.7a Measured and predicted outdoor air ventilation rates (individual rooms) for the exhaust fan ventilated one-family house.

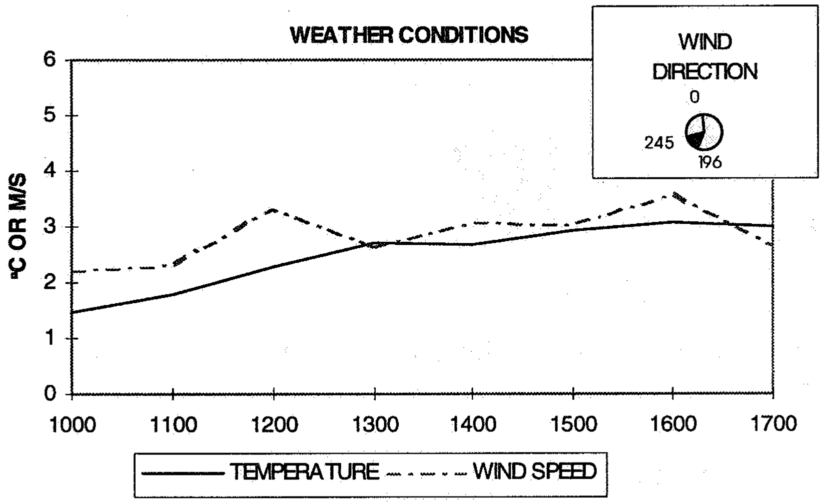

Figure 4.7b The weather for the measured and predicted outdoor air ventilation rates for the exhaust fan ventilated one-family house.

4.2.3 Balanced ventilation

The agreement between simulations and measurements of the average total outdoor air ventilation rates for the measuring periods is good (see figure 4.8) . The predictions vary between an overprediction of 26 % and an underprediction of 4 %. The measured (tracer gas) ventilation rates of individual rooms, in the dwellings with balanced ventilation, agree well with the measurements of the air flows through the supply air terminal devices (see figure 4.9 and 4.10). The exfiltration is very low, as the building envelopes have a good level of airtightness. Therefore the predictions should agree well with the tracer gas measurements and so they do.

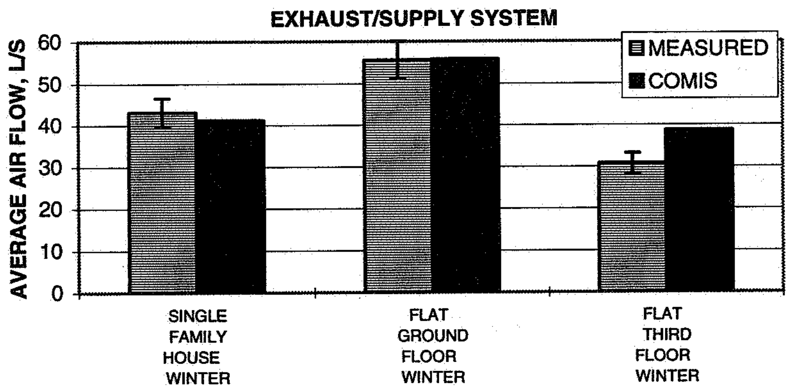

Figure 4.8 Comparison between measured and predicted average total outdoor air ventilation rates in the dwellings with mechanical exhaust and supply ventilation.

Both apartments have an almost constant outdoor air ventilation due to a high level of airtightness (see figure 4.9).

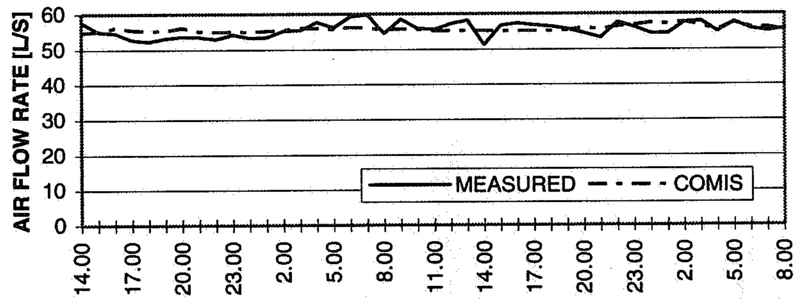

Figure 4.9a Measured and predicted outdoor air ventilation rates (total) for the balanced ventilated apartment on the ground floor.

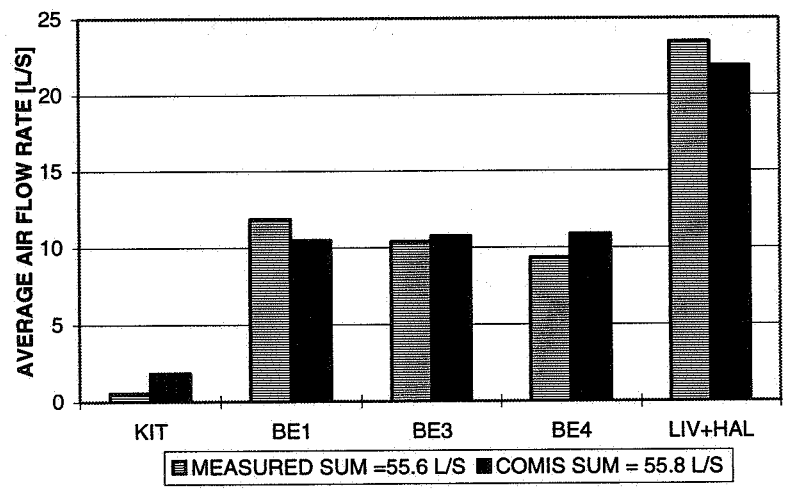

Figure 4.9b Measured and predicted outdoor air ventilation rates (individual rooms) for the balanced ventilated apartment on the ground floor.

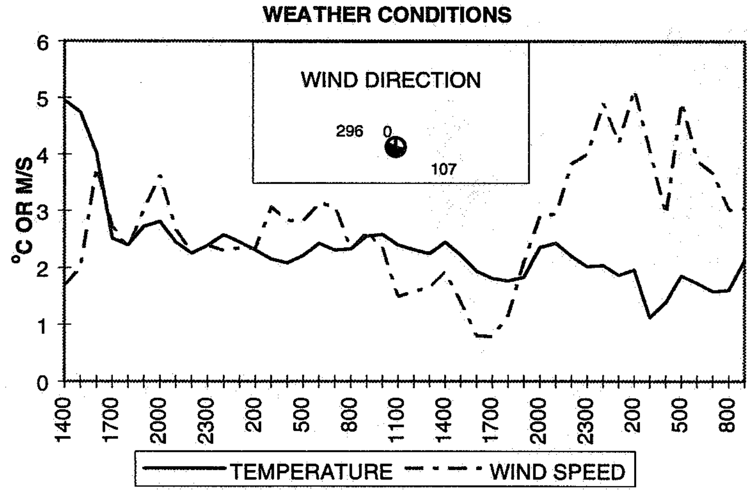

Figure 4.9c The weather for the the balanced ventilated apartment on the ground floor.

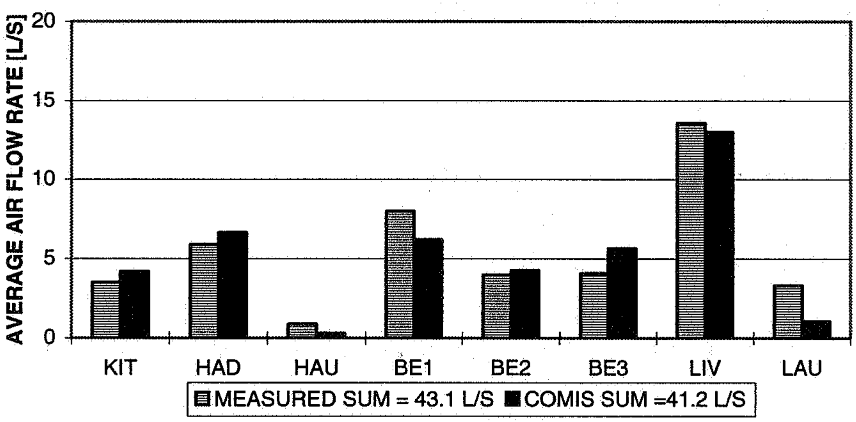

Figure 4.10a Measured and predicted outdoor air ventilation rates (individual rooms) for the balanced ventilated one family house.

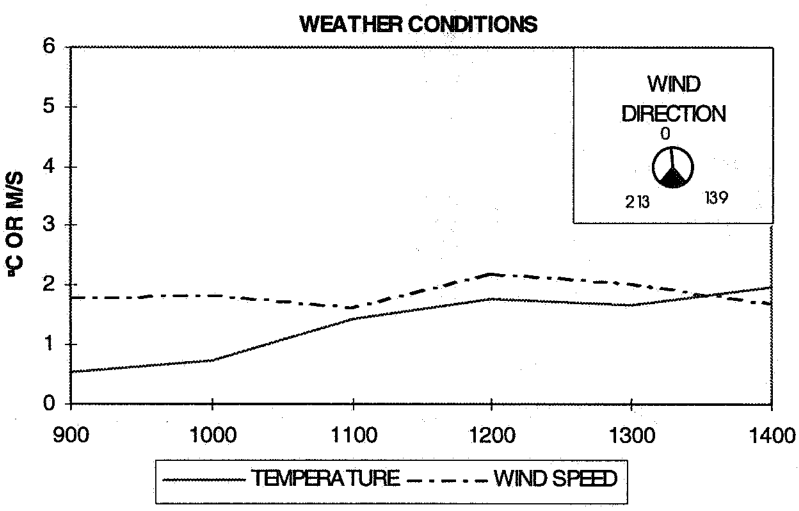

Figure 4.10b The weather for the measured and predicted outdoor air ventilation rates for the balanced ventilated one-family house.

4.3 Ventilation rates - long-term

The differences between measured and simulated average total ventilation rates are similar to the short-term values (see table 4.1). For the dwellings where airing took place and was simulated the discrepancy is often large, mostly COMIS overpredicts. When airing took place but no information about the extent of airing was available and therefore no airing was simulated, COMIS underpredicts, which is reasonable.

Table 4.1 Simulated and measured average total ventilation rates, l/s.

		Period	COMIS	Measured	Difference
Single family houses	**Natural ventilation**	Winter	57.7	37.2 ± 0.8	55%
		Spring	43.0	40.3 ±0.8	7%
		Summer	36.2	40.0 ±0.8	-10%
	Exhaust[1]	Winter	49.8	43.9 ±1.1	13%
		Spring	49.7	42.2 ±0.8	18%
		Summer	49.6	50.6 ±1.1	-2%
	Exhaust/supply[2]	Winter	48.9	52.5 ±1.4	-7%
		Spring	67.3	78.1 ±1.9	-14%
		Summer	67.2	73.9 ±1.7	-9%
Apartments					
Natural ventilation	**Ground floor**[3]	Winter	35.7	31.9 ±1.1	12%
		Spring	26.4	25.6 ±0.8	3%
		Summer	25.6	22.5 ±0.8	14%
	Second floor[3]	Winter	24.8	25.8 ±0.8	-4%
		Spring	18.9	27.5 ±0.8	-31%
		Summer	19.9	24.4 ±0.8	-18%
Exhaust	**Ground floor**[2]	Winter	26.0	34.4 ±1.4	-24%
		Spring	26.0	34.2 ±1.4	-24%
		Summer	25.9	31.4 ±1.4	-17%
	Second floor[2]	Winter	27.1	28.3 ±1.1	-4%
		Spring	27.0	26.9 ±1.1	1%
		Summer	27.0	38.3 ±1.7	-30%
Exhaust/supply	**Ground floor**[3]	Winter	56.7	80.8 ±2.5	-30%
		Spring	56.1	73.1 ±2.5	-23%
		Summer	56.2	53.9 ±1.9	4%
	Third floor[1]	Winter	51.9	48.1 ±1.7	8%

1 The dwelling has been inhabited during all three periods. The inhabitants have recorded the time of airing and running of the cooker hood. The airing area is assumed to be 10% of the open window area for all three period. One or two windows are open i. e. one window in the masterbedroom and one window in the living toom.
2 The dwelling has been inhabited during all three periods. No consideration has been taken to airing.
3 The dwelling has been unoccupied during all three periods.

The discrepancy between measured and predicted ventilation rates for one bedroom in each dwelling can be rather large (see table 4.2). For the dwellings where airing took place and was simulated the discrepancy is large only in one case. When airing took place but no information about the extent of airing was available and therefore no airing was simulated, COMIS tends to underpredict, which is reasonable.

Table 4.2 Simulated and measured average bedroom ventilation rates, l/s.

		Period	COMIS	Measured	Difference
Single family house	**Natural ventilation**[1]	Winter	4,6	3,5 ± 0.3	31%
		Spring	4,2	4,1 ± 0.4	2%
		Summer	12,5	5,4 ± 0.5	131%
	Exhaust[1]	Winter	2,7	2,7 ± 0.3	0%
		Spring	3,5	3,1 ± 0.3	13%
		Summer	3,7	4,1 ± 0.4	-10%
	Exhaust/supply[2]	Winter	5,8	5,1 ± 0.7	14%
		Spring	10,0	8,4 ± 0.7	19%
		Summer	10,2	8,5 ± 0.6	20%
Apartment					
Natural ventilation	**Ground floor**[3]	Winter	8,3	7,2 ± 0.7	15%
		Spring	5,9	6,0 ± 0.6	-2%
		Summer	5,8	5,5 ± 0.5	5%
	Second floor[3]	Winter	5,0	7,4 ± 0.7	-32%
		Spring	4,0	6,7 ± 0.6	-40%
		Summer	4,6	5,9 ± 0.6	-22%
Exhaust	**Ground floor**[2]	Winter	4,4	8,4 ± 0.9	-48%
		Spring	4,3	7,4 ± 0.7	-42%
		Summer	3,9	7,4 ± 0.7	-47%
	Second floor[2]	Winter	10,1	15,3 ± 1.4	-34%
		Spring	9,3	14,5 ± 1.3	-36%
		Summer	9,3	19,4 ± 1.7	-52%
Exhaust/supply	**Ground floor**[3]	Winter	11,2	11,2 ± 1.1	0%
		Spring	11,0	9,5 ± 0.9	16%
		Summer	11,2	6,2 ± 0.6	81%
	Third floor[1]	Winter	8,6	7,9 ± 0.8	9%

5. CONCLUSIONS

The outdoor air ventilation rates in the dwellings with passive stack ventilation varied over time, as could be expected. Some of the individual rooms e.g bedrooms had an outdoor air ventilation rate, which at times were too low (< 4 l/(s and person)). The predicted and measured average (1 - 6 days) total outdoor ventilation rates agree reasonably well. The individual hours can disagree. For indivual rooms the predicted and tracer gas measured outdoor air ventilation rate can be very different.

The exhaust ventilated dwellings had a reasonably constant outdoor air ventilation rate over time. The ventilation rate would have been more constant, if the dwellings had fullfilled e. g. the airtightness requirements of the Swedish Building Code. Individual rooms e. g. bedrooms sometimes have too low (< 4 l/(s and person)) an outdoor air ventilation rate. This was especially true for the leaky one-family house, which had no outdoor air vents. If the house had fullfilled e. g. the airtightness requirements of the Swedish Building Code and had had outdoor air vents, then the distribution of outdoor air to individual rooms would have been better. The agreement between predicted and tracer gas measured average total ventilation rates is good. For individual rooms the agreement is less good.

The dwellings with balanced ventilation had an over time almost constant outdoor air ventilation rate. This was due to the fact that the air leakage of the dwellings was very low. The dwellings fullfilled the requirements on airtightness as given in the Swedish Building Code. The ventilation systems in these dwellings were well adjusted, which meant that the individual rooms were supplied with a reasonable amount of outdoor air. The agreement between tracer gas measured and predicted outdoor air flows was good.

The difference between measurements and simulations can probably be explained by the inaccuracy in the tracer gas measurements of the air flows and the inaccuracy in the inputs to the simulations (distribution of leakage paths, characteristics of individual leakage paths, pressure coefficients, airing by occupant, local wind speed and direction). An important factor for the exhaust ventilated dwellings is that as input for the exhaust air flows through the air terminal devices measured values were used. The good agreement between predicted and measured (tracer gas) average total outdoor air ventilation rates can partly be explained by a good agreement between tracer gas measurements and measurements of exhaust air flows at the air terminal devices. For the balanced ventilated dwellings both measured exhaust and supply air flows through the air terminal devices were used as inputs.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

17th AIVC Conference, Gothenburg, Sweden,
17-20 September, 1996

A TOOL FOR EVALUATING DOMESTIC VENTILATION SYSTEMS' ABILITY TO PROVIDE AN ACCEPTABLE INDOOR AIR QUALITY

Lars-Goran Mansson*, Johnny Kronvall**, Jean-Robert Millet***, Jean-Georges Villenave***, Anders Holtsberg****

* LGM Consult AB, Adler Salvius vag 87, S-146 53 Tullinge, Sweden

** J & W Consulting Engineers, Slagthuset, S211 20 Malmo, Sweden

*** CSTB, 84 Ave Jean Jaures, Champs sur Marne BP 2-77421 Marne La Vallee, Cedex 2 France

**** Lund University, Dept of Mathematical Statistics, P O Box 118, S 221 00 Lund, Sweden

(Full paper not available at time of print)

Abstract

17th Annual Conference

Optimum Ventilation and Air Flow Control in Buildings

Title

A Tool for Evaluating Domestic Ventilation Systems' Ability to Provide an Acceptable Indoor Air Quality

Authors:

Lars-Göran Månsson*; Johnny Kronvall ⅄, Jean-Robert Millet ✦, Jean-Georges Villenave ✦, Anders Holtsberg ●

Affiliation:

* LGM Consult AB, Adler Salvius väg 87, S-146 53 Tullinge, Sweden.

⅄ J&W Consulting Engineers, Slagthuset; S-211 20 Malmö, Sweden

✦ CSTB, Ventilation and Air Conditioning Div; 84, Ave Jean Jaures, Champs-sur-Marne; BP 2-77421 Marne la Vallée, Cedex 2; France.

● Lund University; Dep. of Mathematical Statistics, P.O. Box 118; S-221 00 Lund; Sweden

The IEA Annex 27 "Evaluation and Demonstration of Domestic Ventilation Systems" formulated that one of the objectives is to develop simplified tools for evaluating the consequences of a specific chosen system in a particular dwelling. Assumptions have been made for parameters that cover most of the situations that can occur in dwellings. The combination of all parameters will result in unsurveyable results. The treatment was to use a statistical method to select combinations representing the whole spectrum. The method used is fractional factorial analysis. With this metod a selection of combination represents all combinations. Those selected combinations are simulated by using the semi-multizone program SIREN developed by CSTB, France. Also the multizone model COMIS will be used and the same procedure for analysing the same set of combination will be made so the results can be checked.

The simplified tool is made up by using an average value multiplied by factors depending on the situation to be evaluated. The parameters to be given factors are dwelling type, location in a 4 storey multi-family building, air leakage rate, number of residents, window airing pattern, climate, supply area, flow rate, if local fans are used. This is then applied on the four basic ventilation systems: Adventitious, passive stack, mechanical exhaust, and mechanical supply and exhaust. Totally this gives 174 combinations to be run. Simplified tools will be set up for the following indoor air quality parameters: area based pollutants (eg. building material), CO_2 expressed in ppmhours for the exposed person, pollutants from cooking products, tobacco smoke, humidity expressed as hours of condensation hours with RH>75 % (mould growth) and 4 consecutive weeks with water vapour above 7 g/kg (house dust mites), and outdoor air flow rate. The same approach will be used for ventilation heat needs and fan operation energy.

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

17th AIVC Conference, Gothenburg, Sweden, 17-20 September, 1996

(Title) **REDUCTION OF FLOW LOSS DUE TO HEAT RECOVERY IN PSV SYSTEMS BY OPTIMUM ARRANGEMENT OF HEAT-PIPE ASSEMBLIES**

(Authors) S B Riffat and L Shao

(Affiliation) Institute of Building Technology
University of Nottingham
Nottingham
NG7 2RD
United Kingdom

REDUCTION OF FLOW LOSS DUE TO HEAT RECOVERY IN PSV SYSTEMS BY OPTIMUM ARRANGEMENT OF HEAT-PIPE ASSEMBLIES

SYNOPSIS

Natural ventilation is being applied to an increasing number of new buildings to minimise reliance on mechanical ventilation and so reduce emission of greenhouse gases. However, passive stack ventilation (PSV) systems are currently designed without incorporating heat recovery leading to significant wastage of energy. Heat recovery systems have not been used in naturally-ventilated buildings because the pressure loss caused by a conventional heat exchanger is large compared to the stack pressure and could cause the ventilation system to fail. In addition, the stack pressure decreases owing to reduction of the temperature difference associated with the heat exchange, although this problem can be lessened by appropriate siting of the heat exchanger to maximise the effective stack height. In this study, natural convective flow through PSV stacks were computed using CFD to determine the effect of the layout of heat-pipe assemblies as well as the effects of spacing and length of fins on reduction of flow rate through the stack. Among the layout patterns examined, the arrangement where the assemblies are placed in a pattern of an arrow facing the flow direction produced the least insertion flow loss. The flow loss due to the insertion of the heat pipe assemblies (IFL) was found to increase sharply with the number of fins and reached over 30% when only 4 fins were used. IFL also increased with fin length but the rate of increase reduced for larger fin lengths. Therefore, for a given total surface area of fins, using fins with a larger length causes less flow loss than fins with a smaller spacing.

LIST OF SYMBOLS

g	Gravitational acceleration; 9.8 m/s^2
Gr	Grashof number; dimensionless
IFL	Insertion flow loss; %
l	Stack height; m
Q_{HP}	Volumetric flow rates in stack with heat-pipe assemblies; m^3/s
Q_{noHP}	Volumetric flow rates in stack without heat-pipe assemblies; m^3/s
T_w	Temperature of the stack walls; K
T_∞	Air temperature at the stack inlet; K
V_V	Vertical component of air velocity; m/s

1. INTRODUCTION

A wide range of buildings employ natural ventilation to minimise reliance on mechanical ventilation and so reduce emission of greenhouse gases. The passive stack ventilation (PSV) system which relies on the stack pressure created by the temperature difference between the indoor and outdoor air has been applied to various types of modern buildings, including offices, schools and houses [1, 2]. Natural stack ventilation consumes no power and so produces no harmful emissions, has no running cost, no noise of operation, requires little maintenance and because it involves no moving parts, operation is reliable. However, PSV systems are currently designed and constructed without incorporating heat recovery leading to wasteful heat loss. It has been estimated that this heat loss amounts to 3 - 15 GJ per annum for a small family residence and much more for larger buildings, e.g., offices [2]. There is obviously a need for appropriate and efficient heat recovery in PSV systems to minimise energy wastage. This paper presents results of a computer simulation which forms part of a feasibility study of a novel heat recovery system for use in PSV systems. The flow loss in the passive stack incorporating heat-pipe heat recovery was evaluated using computational fluid dynamics (CFD) to provide an insight into the mechanism and characteristics of the loss and an indication of its magnitude. Effects of spatial arrangement of heat-pipe assemblies and the spacing/length of fins were studied.

2. HEAT-PIPE HEAT RECOVERY SYSTEM

Heat recovery systems have not been used in naturally-ventilated buildings because the pressure loss caused by a conventional heat exchanger is large compared with the stack pressure and could cause the ventilation system to fail. In addition, the stack pressure decreases owing to reduction of the temperature difference associated with the heat exchange, although this problem can be lessened by appropriate siting of the heat exchanger to maximise the effective stack height. Research work on natural stack ventilation has been carried out by Schultz and Saxhof [3] using a counterflow heat exchanger consisting of channels separated by smooth corrugated metal sheets. This system had a high pressure drop, low capacity and large weight making it impractical to install in real buildings.

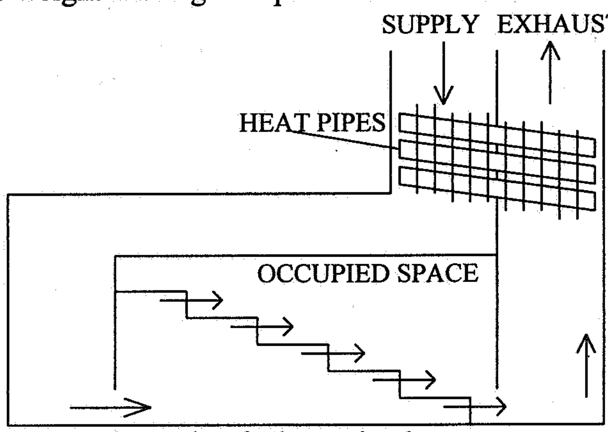

Figure 1. Schematic of a heat-pipe heat recovery system.

Heat pipes offer an alternative approach for heat recovery in naturally-ventilated buildings and have several advantages. The heat pipe consists of a sealed pipe lined with wick and partially filled with a working liquid. Installation is simple and involves locating one end of the tube in the warm exhaust ventilation stack and the other in the cool supply ventilation stack (see Figure 1). The liquid absorbs heat to evaporate at the warm end and condenses to release heat at the cool end, thereby performing the task of heat exchange. The condensed liquid is returned to the warm end by capillary forces in the wick fibres or by gravity. The heat pipe promises much higher capacity because it has much higher thermal conductance than the metal channels of conventional heat exchangers. In addition, it does not require complicated channels for supply and exhaust air. Individual heat pipes can be independently located in air ducts/cavities, making it easier to achieve a low pressure drop than using conventional heat exchangers which require the air path to be divided into narrow, flow resistant channels. Preliminary tests were carried out using a heat pipe system manufactured by Isoterix Ltd., UK. The results showed that for a heat recovery efficiency of 50%, stack flow rate of $0.056m^3/s$ and stack flow speed of 0.5m/s, the pressure loss across an existing type of Isoterix heat pipe assembly is about 1 Pascal, which would not cause significant reduction of stack flow (The driving force of stack flow depends on temperature difference and stack height but is typically 5-10 Pascal). This pressure loss could be further reduced by adopting a more streamline shape for the heat pipes. The features described above make the heat pipe suitable for heat recovery in natural-ventilation systems.

3. DESCRIPTION OF THE COMPUTATION

The CFD code FLUENT was used to simulate pressure/flow loss in ventilation stacks by solving the Navier-Stokes equations, the mass conservation equation and the energy conservation equation. The simulated natural ventilation stack and the computational domain of this study is shown in Fig. 2. This study focuses on the buoyancy flow in the exhaust stack and the flow in the room connecting to the stack is not considered and therefore not included in the computational domain. Although the flow in the room may affect the flow in the ventilation stack by influencing the inlet boundary condition of the stack, this effect is dependent on individual room configuration and ventilation conditions, which vary considerably. Obviously, it is impossible to study all room configurations and difficult to select a small number of them as being representative of the vast range of room shapes. Therefore it was decided that only the stack flow should be examined. This is acceptable, as the buoyancy flow in the exhaust stack is the key to the functioning of a passive stack system. An understanding of the mechanisms of thermal and aerodynamic processes in the stack forms the basis for understanding heat recovery in natural ventilation systems. The above simplification of the computation domain is necessary because it reduces the demand on computing resources to a more acceptable level.

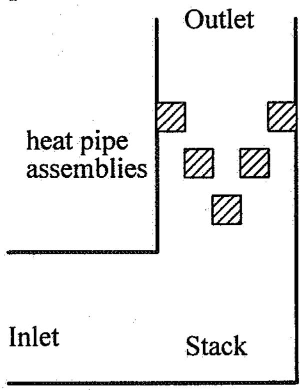

Figure 2. Schematic of the stack with heat recovery used in the simulation.

Computation of natural convective flow driven by small temperature differences such as those experienced in natural ventilation systems requires much more CPU time compared with computation of force flow in similar geometric set-ups [4]. In order to reduce the required CPU time to a manageable level, only two-dimensional computation was carried out. The inlet and outlet of the stack open to the outdoor atmosphere. The overall height and width of the stack (including the horizontal duct) are 3 m and 1 m, respectively. The width of the vertical stack and the height of the horizontal duct are both 0.4 m.

A major difference between the heat-pipe heat exchanger and more conventional heat exchangers is that for the former, heat is carried from the source to the sink by the refrigerants in the heat pipes instead of the wall separating the source and sink flows. To increase heat transfer capacity, the latter would resort to dividing the flow into smaller flow channels to increase the contact area while the former would simply require installation of multiple heat pipes spanning the source and sink flows. A typical heat-pipe has the shape of a cylindrical rod and several heat pipes are often bundled together and fitted with fins to form a heat-pipe assembly. As shown in Figure 2, heat-pipe assemblies are modelled in the simulation and placed perpendicular to the plane of the two-dimensional computational domain. The cross section of the assembly measures 0.08 m(w) × 0.09 m(h). It has been shown that the temperature of heat-pipe assemblies has little effect on the flow or pressure loss across the assemblies and therefore it was left at the default temperature (273K) set by the CFD code. As the fins are normally perpendicular to the heat pipes, they can not be simulated in a two-dimensional domain and the heat-pipe assembly would appear as a solid block. This limitation can be overcome by three-dimensional simulation, which will be carried out later. The limitation is not critical for this preliminary study, because the focus of the investigation is the loss of flow which would in any case normally flow around the assembly as if it were solid.

The computational domain is mapped by a Cartesian grid of 40×80 and the grid density in the axial direction of the stack is less than that for the spanwise direction to reduce the number of grid nodes required [5]. The walls of the stack have a temperature of 20°C and the inlet air temperature is 5°C. These were used to calculated the Grashof Number (Gr) to determine whether the flow is turbulent.

$$Gr = \frac{g(T_w - T_\infty)l^3}{T_w v^2}$$

where $g = 9.8\ m/s^2$, $Tw = 293$ K, $T\infty = 278$ K, $l = 3$ m and $v = 1.33 \times 10^{-5}\ m^2/s$. Based on the above values of the parameters involved, the resultant Gr is 7.75×10^{10}, which is greater than the transition value which is of the order of 10^8. Therefore the flow is turbulent and the k-ε turbulence model was used in the computation. The governing equations for the stack flow were discretised into finite volume equations, based on a Cartesian grid and the Power-law interpolation scheme. The equations resulting from the discretisation process were solved using the SIMPLE algorithm. Convergence acceleration methods including increasing the values of under-relaxation factors, multiple sweeps for the enthalpy equation and multigrid technique were used.

4. RESULTS AND DISCUSSION

Twelve cases of natural convective flow through the stack were computed, including cases without heat pipes and cases with various layout patterns for the heat pipes assemblies. The effect of the layout of the assemblies as well as the effects of the spacing between and length of fins of the heat pipe assemblies were also examined.

Figure 3 shows the flow field in the ventilation stack without a heat-pipe assembly. The average vertical velocity at the stack outlet and the volumetric flow rate through the stack are 0.112 m/s and 0.0447 m^3/s. Because the simulation is two dimensional, the flow rate reported is for an stack with a depth of 1m. The average vertical velocity is defined as

$$V_{V-average} = \frac{\sum_{i=1}^{N} V_{V-i}}{N}$$

where $V_{V\text{-}i}$ is the vertical component of air velocity at i'th cell in the outlet of the stack and N is the total number of cells in the inlet/outlet cross-section. The volumetric flow rate through the stack is the product of the average velocity and the duct cross section. This will be compared with the flow rate when the heat pipe assemblies are inserted in the stack to obtain the insertion flow loss (IFL) as shown in the following sections.

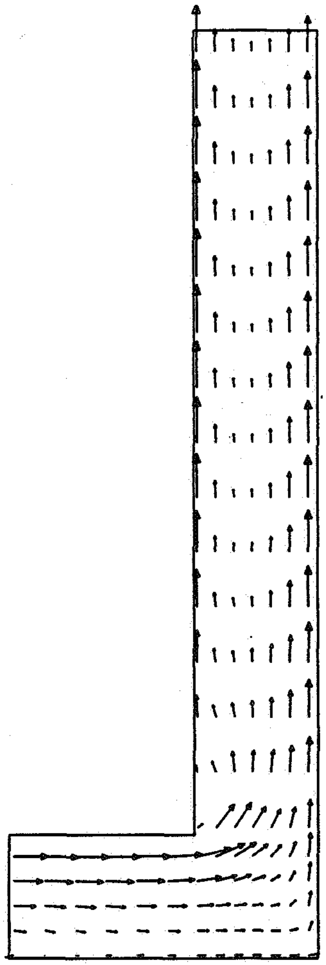

Figure 3. Flow field in the stack.

The effect of the relative location of the heat pipe assemblies was examined by computing six cases, each with a different layout pattern for the heat-pipe assemblies. , as indicated by Layout 1 - Layout 5 in Figure 4. In all six cases, the heat pipe assemblies are grouped around the centre of the vertical stack and the height of the lower horizontal surface of the bottom row of assemblies is 1.521 m. The spacing between neighbouring rows and columns of assemblies are 0.09 and 0.0 m, respectively. For Layout 1, the volumetric flow rate through the stack is 3.38 m^3/s. The reduction of flow rate caused by the insertion of the heat-pipe assembly (IFL) is 26.6%. This insertion flow loss (IFL) is defined as

$$IFL = \frac{Q_{noHP} - Q_{HP}}{Q_{noHP}}$$

where Q_{noHP} and Q_{HP} are the volumetric flow rates in the stack with and without the heat-pipe assembly, respectively. It has been demonstrated that IFL is the more appropriate indicator of flow loss produced by obstacles submerged in natural convective flow such as that in a natural ventilation stack. The traditional indicator of pressure loss due to the insertion obstacles is not suitable [6].

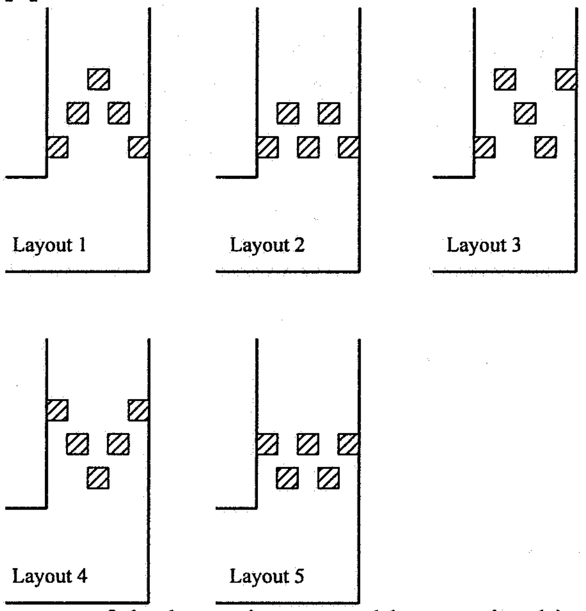

Figure 4. Layout of the heat-pipe assembly examined in this study.

The IFL's corresponding to the other four layouts of the heat-pipe assembly are listed in Table 1, together with that for Layout 1. As shown in the table the spatial arrangement of heat pipe assemblies that produced the least insertion flow loss is Layout 4 where the assemblies are placed in a pattern of an arrow facing the flow direction. Reversing the direction of the arrow in line with the flow direction lead to one of the highest flow loss found in this study. The second best arrangement producing low IFL is Layout 3 where the assemblies are located in three rows. The other two layouts (Layout 2 and 5) where the assemblies are placed in only two rows give rise to the highest insertion flow losses.

Table 1. IFL of for various layout patterns indicated in Figure 4.

	IFL (%)
Layout 1	26.6
Layout 2	33.6
Layout 3	19.0
Layout 4	8.1
Layout 5	28.9

The effect of finning arrangement on IFL was also studied by computing 6 additional cases. It is well known that reducing fin spacing and increasing fin length would increase heat transfer performance of a heat exchanger but at the cost of increasing resistance to the flow and flow loss. The extent of the flow loss was investigated by computing IFL in stacks with various numbers of uniformly spaced fins. This number was increased in stepwise fashion from 1 to 4. Although the number used is smaller than that normally found in heat exchangers, it nevertheless allows concrete conclusions to be drawn. Increasing the fin number further would require substantially more computing resources. The four-fin example is illustrated in Figure 5. The fin planes are perpendicular to the plane of the computational domain and have lengths of 0.314 m. The heat pipe linking the fins was not modelled as doing so in a 2-D simulation would block the stack. In addition, it omission is acceptable because the heat pipes are small in diameter and contributed little to flow resistance compared with the fins. The fins are located around half full-height of the vertical stack with their lower edges at 1.566 m above the floor of the horizontal stack. The stack geometry and thermal boundary conditions used are identical to those described in section 3.

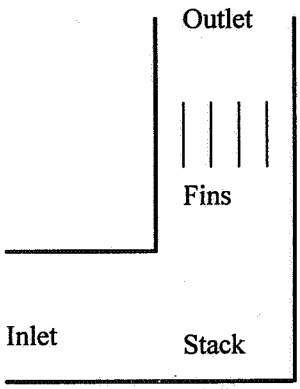

Figure 5 Schematic of arrangement of fins used in the computation.

Figure 6 shows the effect of fin spacing which is the distance between neighbouring fins. As the fin number increases from 1 to 4 the flow loss increases exponentially to over 30%. The IFL would be much greater if a larger number of fins were used as would be the case in a practical heat exchanger design. Obviously, having more than two fins in a 0.4 m wide stack would be undesirable.

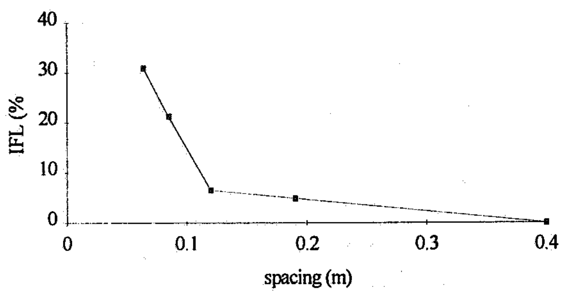

Figure 6. Effect of fin spacing on flow loss

The effect of fin length was examined by computing three cases which were identical to the three-fin case described above except their fins have different lengths. The fin lengths for the three cases are 0.179m, 0.314m and 0.493m, respectively. As shown in Figure 7, flow loss increases with fin length but the rate of increase in flow loss reduces for larger fin lengths. Therefore, for a given total surface area of fins, using fins with a greater length causes less flow loss than fins with a smaller spacing.

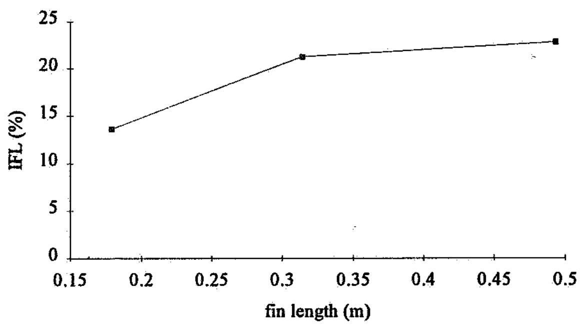

Figure 7. Effect of fin length on flow loss in the stack.

5. CONCLUSIONS

Twelve cases of natural convective flow through the stack were computed using CFD to determine the effect of the layout of heat-pipe assemblies as well as the effects of spacing and length of fins. Among the five layout patterns examined, the arrangement where the assemblies are placed in a pattern of an arrow facing the flow direction produced the least insertion flow loss. IFL was found to increase sharply with the number of fins and reached approximately 30% when only 4 fins were used. IFL also increases with fin length but the rate of increase reduces for larger fin lengths. Therefore, for a given total surface area of fins, using fins with a larger length causes less flow loss than fins with a smaller spacing.

6. ACKNOWLEDGEMENT

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

17th AIVC Conference, Gothenburg, Sweden,
17-20 September, 1996

The Effect on Ventilation Parameters of Various Ventilation Strategies

M W Simons, J R Waters

School of the Built Environment, Coventry University,
Priory Street, Coventry, UK

The Effect on Ventilation Parameters of Various Ventilation Strategies

1 Synopsis

The work described in this paper is aimed at predicting the local values of the ventilation effectiveness parameters of large industrial buildings by a technique which involves the use of computational fluid dynamics and multizonal modelling.

A modelling technique is described and applied to a typical modern industrial building equipped with both, mixing and displacement ventilation systems. The results of modelling each of the above systems are presented and discussed. They provide an interesting insight into magnitude and spatial variations in of local air change index that occur in the occupied space. The results also demonstrate how differences in ventilation strategy can result in distinctly different variations of ventilation effectiveness parameters.

It is concluded that the modelling technique described may be used to provide important information about the air movement characteristics of buildings in terms of local air change index and also that it could prove to be a very useful design aid.

2 List of Symbols

Symbol		Units
ε_p	local air change index at point p	
$\bar{\tau}_p$	local mean age of air at point p	s
τ_n	nominal time constant	s

3 Introduction

Ventilation effectiveness parameters provide a valuable insight into the air movement characteristics of enclosures. They are of particular importance in the case of buildings which contain large undivided internal spaces since the characteristics of such buildings are particularly difficult to predict. In addition, effective ventilation strategies to remove the contaminants associated with the industrial processes frequently carried out in such buildings are essential if risks to health are to be avoided. However, the measurement of ventilation effectiveness parameters is restricted to the relatively small number of points at which tracer gas can be sampled. It is therefore of value to be able to model the behaviour of air within large buildings in order to predict the magnitude and variation of the parameters. This work represents a development of that reported by M W Simons et al [1] which because of computational constraints was restricted to buildings subdivided, for the purpose of CFD analysis, into a small number of zones

4 Ventilation Effectiveness Parameters

Ventilation effectiveness parameters which have been devised to allow the variability of the air movement within a space to be studied may be subdivided into the categories of **air change efficiency** and **contaminant removal efficiency**. Air change Efficiency parameters have been devised to establish how effectively the air within an enclosure is replaced by fresh ventilating air which is the purpose of the present study. The most commonly used are:

i Local mean Age of Air, $\bar{\tau}_p$

This is defined as the average time taken for air to travel from the inlet to any point p in the room and may be written as:

$$\bar{\tau}_p = \int_0^\infty t.A_p(t).dt$$

where $A_p(t)$ represents the age distribution curve for air arriving at point *p*. The local mean age of the air is different at each point, *p* within the room.

ii Local Air Change Index (LACI), ε_p

This index provides a measure of the age of the air at a point relative to the overall supply rate being defined as:

$$\varepsilon_p = \frac{\tau_n}{\bar{\tau}_p}$$

where τ_n is the nominal time constant of the room. The nominal time constant is the reciprocal of the ventilation air change rate.

It will be observed that a value of LACI greater than 1 indicates that a point is receiving air more efficiently than the average and that the higher the index, the better the ventilation. The opposite applies to values less than 1. The result of the current study have been expressed in terms of LACI.

5 Experimental Procedure

5.1 The approach adopted has been to first undertake a CFD analysis of the internal space in order to establish intercellular flows. The CFD software used for this work was 'Flovent' by Flomerics. The intercellular flows were then post processed by inputting them as interzonal flows into software developed at Coventry University to undertake a multizonal analysis and hence establish the relevant ventilation effectiveness parameters.

A significant difference in the post processing software used here and that reported previously is that it has been necessary to replace the standard matrix inversion routines with a customised solver specially designed to handle the very large sparse matrices involved.

Software has been made available from 'Flomerics' to return the computed ventilation effectiveness parameters to Flovent for the purposes of presentation.

5.2 Investigation of several buildings has been undertaken, one of which has been selected for presentation here as an example of the information that can be derived.

The building chosen, which is shown in Figure 1, is representative of the type of modern building that could be used for engineering manufacture in a location where extremes of climatic conditions may be expected. The building was selected to closely resemble a Finnish factory unit which has been used as a case study for the International Energy Agency Annex 26 Project "energy Efficient Ventilation of Large Enclosures" by the Finnish Institute of Occupational Health [2].

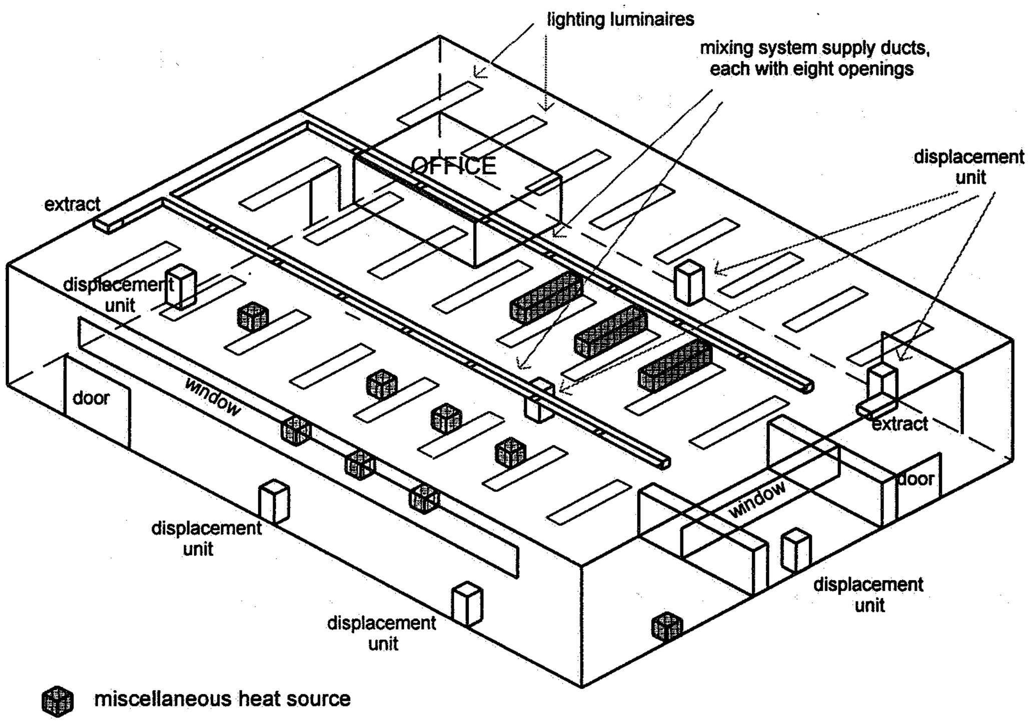

Figure 1 Factory with mixing and displacement ventilation systems

The model is of a factory 40m by 30m in plan and 8m high with wall, roof and window thermal transmittances of 0.45, 0.25 and 2.8 $Wm^{-2}K^{-1}$ respectively. Various incidental heat sources are present within the building as indicated in Figure 1.

Mixing and **Displacement** ventilation systems have been modelled assuming the environmental conditions described below:

i Mixing Ventilation System

An ambient temperature of 5^0C is assumed and air is supplied through 16 roof mounted duct openings each supplying $0.44kgs^{-1}$ ($0.37m^3s^{-1}$) at 18^0C. This gives an overall air change rate of 2 per hour and corresponds to winter heating in the UK.

ii Displacement Ventilation System

An ambient temperature of 22^0C is assumed and air is supplied through 7 floor mounted displacement flow diffusers at a velocity of $0.3ms^{-1}$ and a temperature of 14^0C. The data used for the performance of the diffusers is typical of currently manufactured products. This gives an overall air change rate of 2.2 per hour and corresponds to a low level, low turbulence displacement supply suitable for providing summer cooling in the UK.

6 Consideration of results

6.1 Mixing Ventilation System

Figure 2 illustrates the air velocity vectors and temperature gradients 1.5 m above floor level with the mixing ventilation system operational. This shows an acceptable velocity distribution and uniform temperature of between 18.5^0C and 19^0C, the temperatures above 19^0C corresponding to positions immediately above the heat sources.

If the mixing system was working perfectly, the value of the LACI would be unity throughout the space. However, Figure 3 clearly shows that despite the apparently satisfactory airflow patterns, the values of LACI at head height range from values in excess of 1.10 close to the supply openings down to values below 0.90 over a significant proportion of the working space. The areas with low values of LACI correspond to those areas where air movement is least. The variability of the LACI is also shown in Figure 4 where inspection reveals that on vertical planes remote from the supply openings, the value of the index follows a similar range, from 0.85 to 1.15.

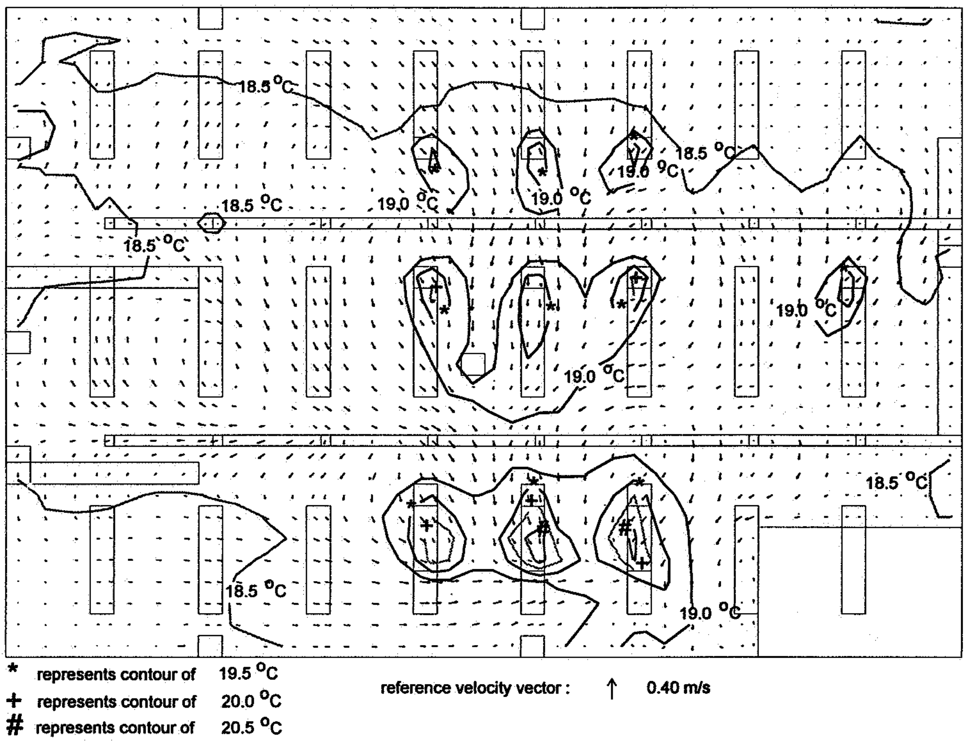

Figure 2
Mixing Ventilation Velocity Vectors and Temperature 1.5m Above Floor Level

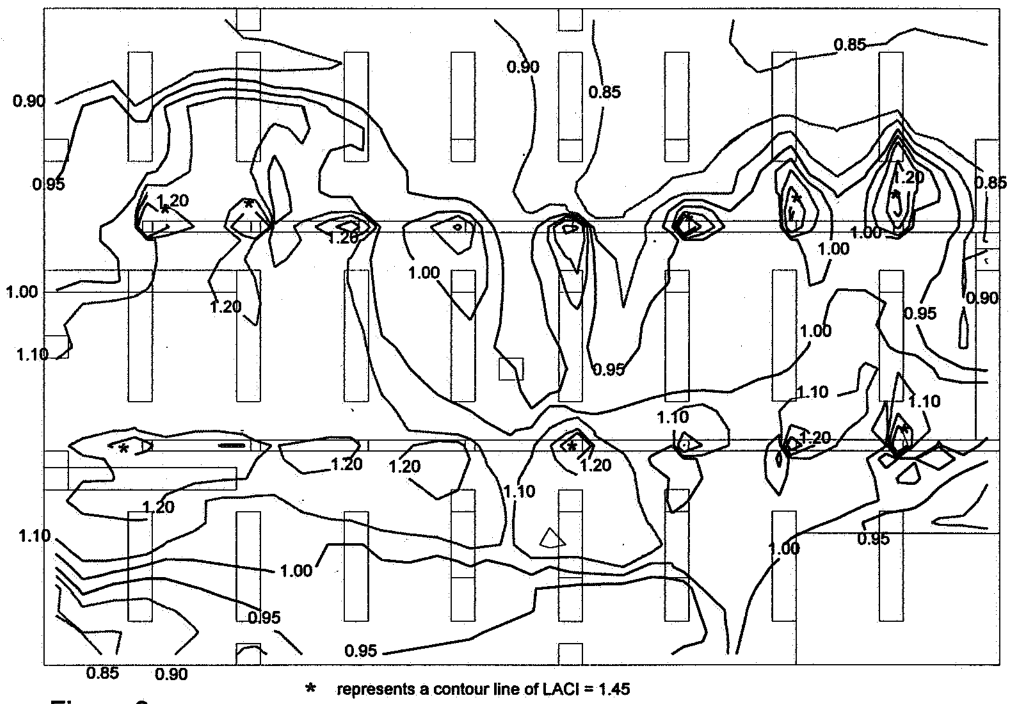

Figure 3
Mixing Ventilation LACI Contours on Plane 1.5m Above Floor Level

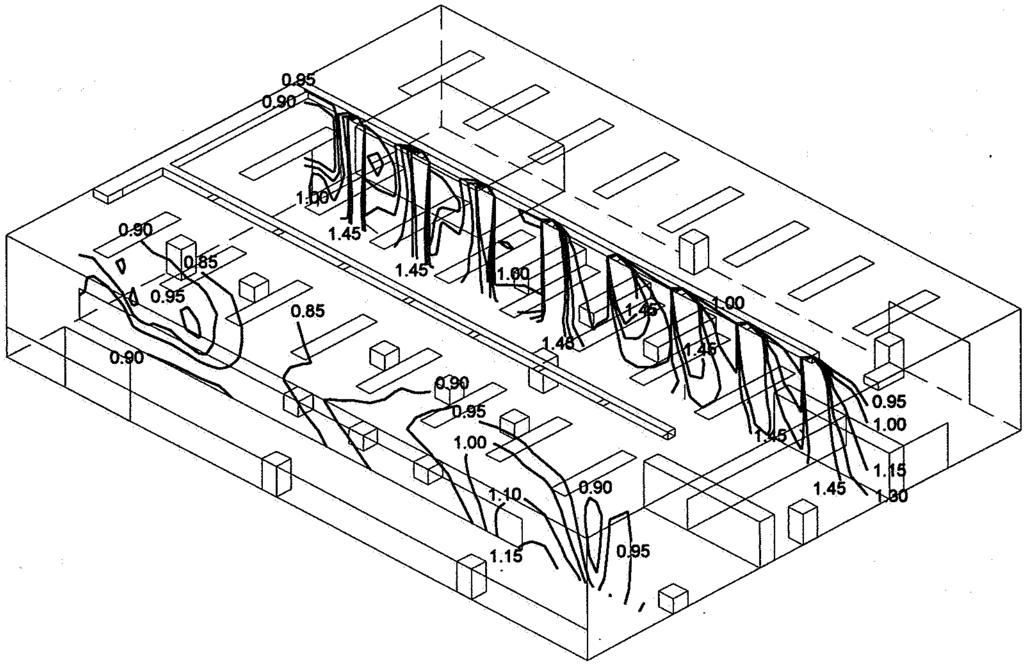

Figure 4 Mixing Ventilation
Contour diagrams of LACI on vertical planes

Figure 5 illustrates the magnitude of LACI close to the mixing ventilation supply outlets. It is interesting to observe how rapidly the index reduces in close proximity to the supply outlets and that values of less than unity are present within 2m of them, showing that high values of LACI are restricted to the immediate vicinity of the outlets

The overall conclusion is that the mixing system is behaving satisfactorily.

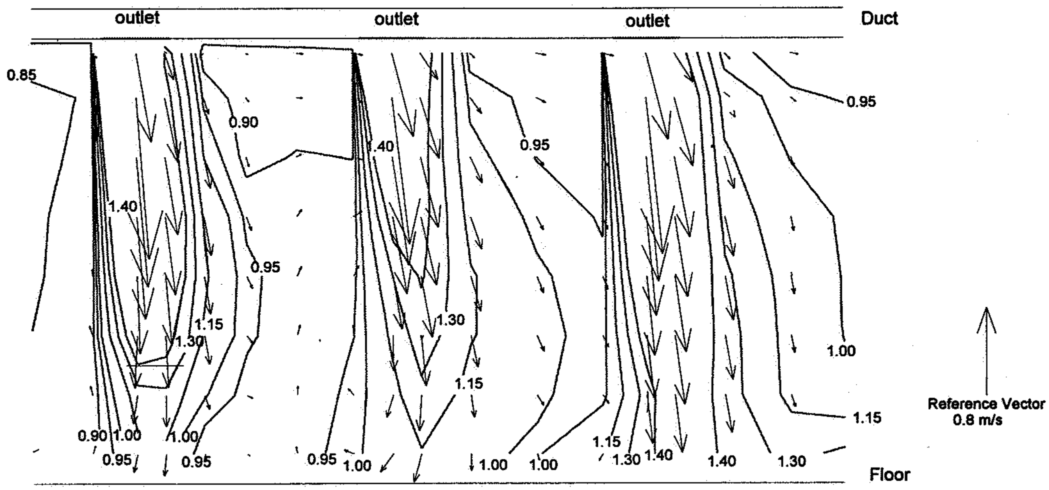

Figure 5 LACI Contours and Velocity Vectors in Vicinity of Mixing System Outlets

6.2 Displacement Ventilation System

Figures 6, 7 and 8 illustrate the LACI contours, at heights of 0.5m, 2.5m and 5.5m respectively above floor level, resulting from operation of the displacement system. These figures clearly show that the values of the LACI are high at the low level and decrease with height above floor level. However they also show a considerable variation across these horizontal planes, i.e.:-

Height(m)	Range of LACI	
	from less than	to greater than
0.5	1.3	1.9
2.5	0.8	1.3
5.5	0.7	1.3

with the proportion of the area at a high value decreasing with height. The variability of the LACI in the vertical plane is also clearly shown in Figure 9.

The interpretation of these contour diagrams is that whereas the displacement system is **on average** behaving as expected, there are notable variations, with a significant area at a height of 2.5m with an LACI of less than 1. It appears that in addition to the upward movement of air caused by the displacement units, there is a circulatory motion with air rising in some regions (high values of LACI) and falling in others (low values of LACI). The contours in Figure 8 show this particularly well.

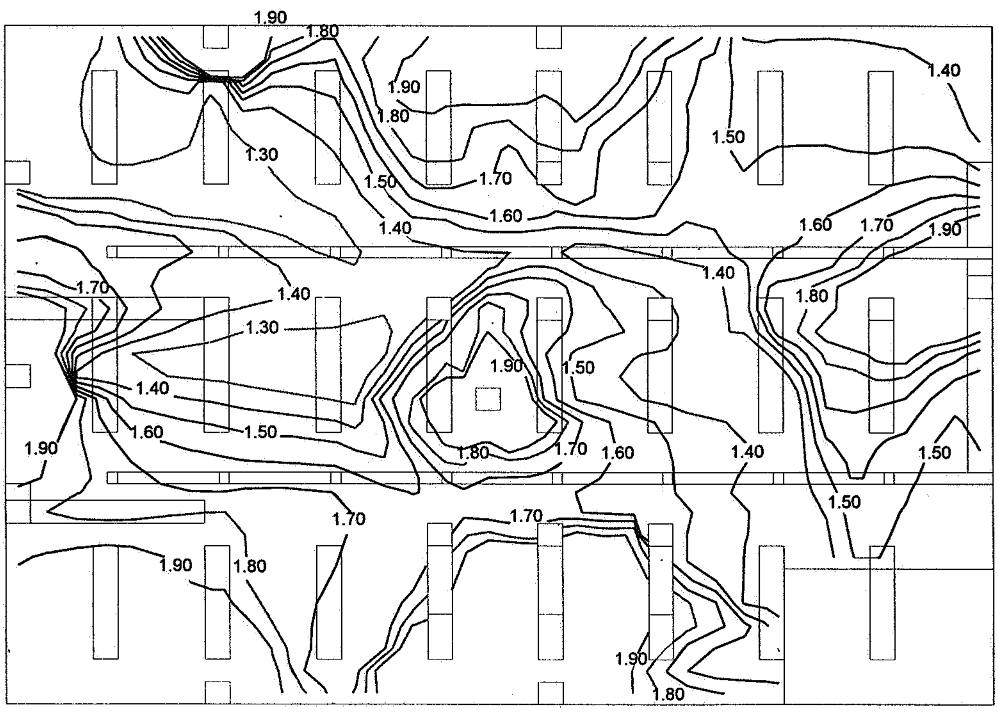

Figure 6
Displacement Ventilation LACI Contours on Plane 0.5m Above Floor Level

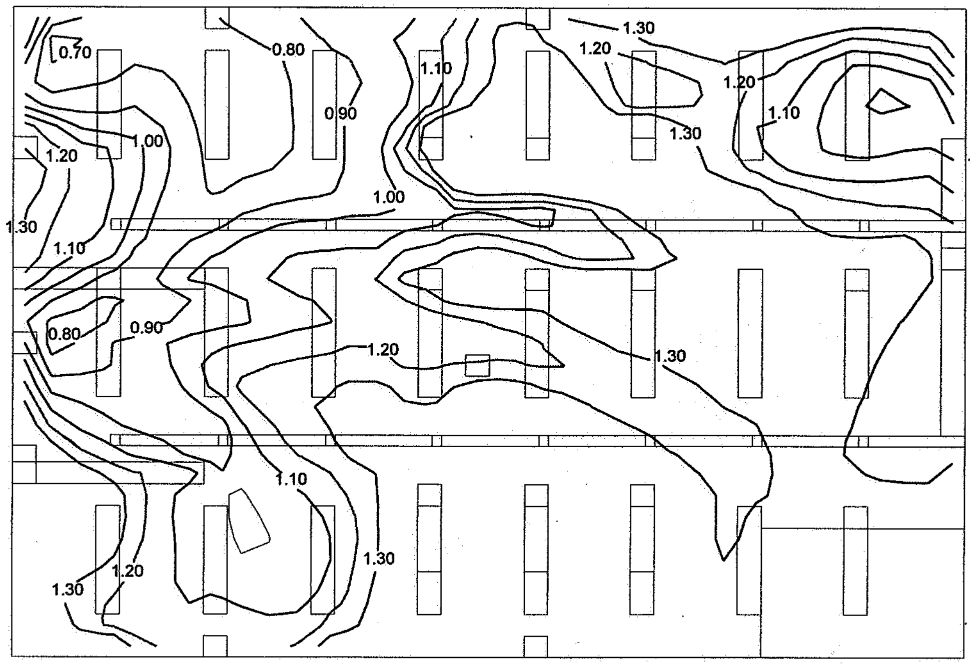

Figure 7
Displacement Ventilation LACI Contours on Plane 2.5m Above Floor Level

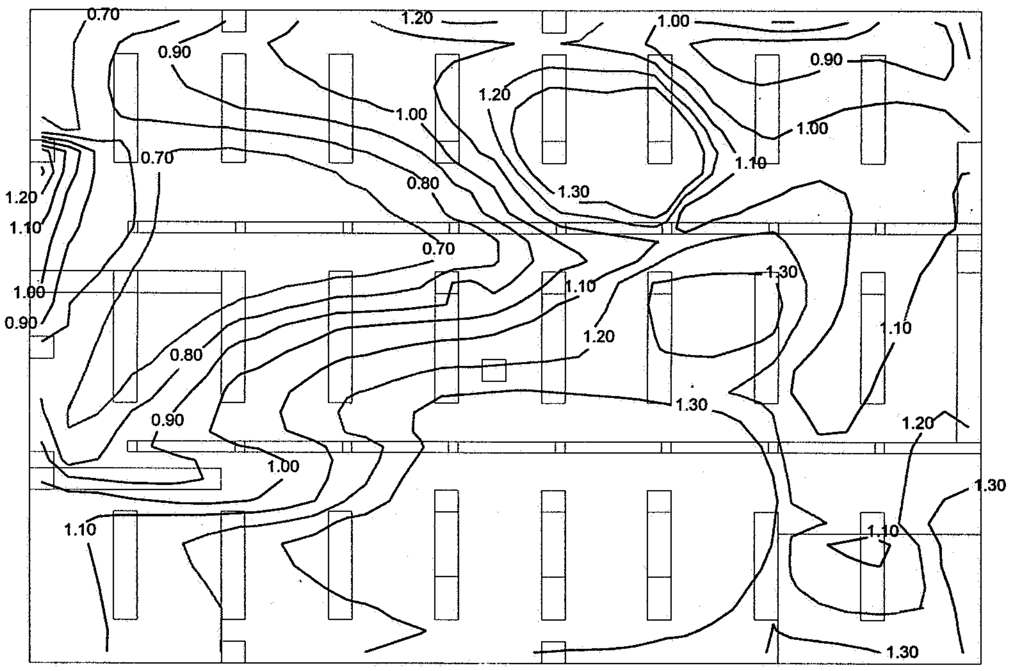

Figure 8
Displacement Ventilation LACI Contours on Plane 5.5m Above Floor Level

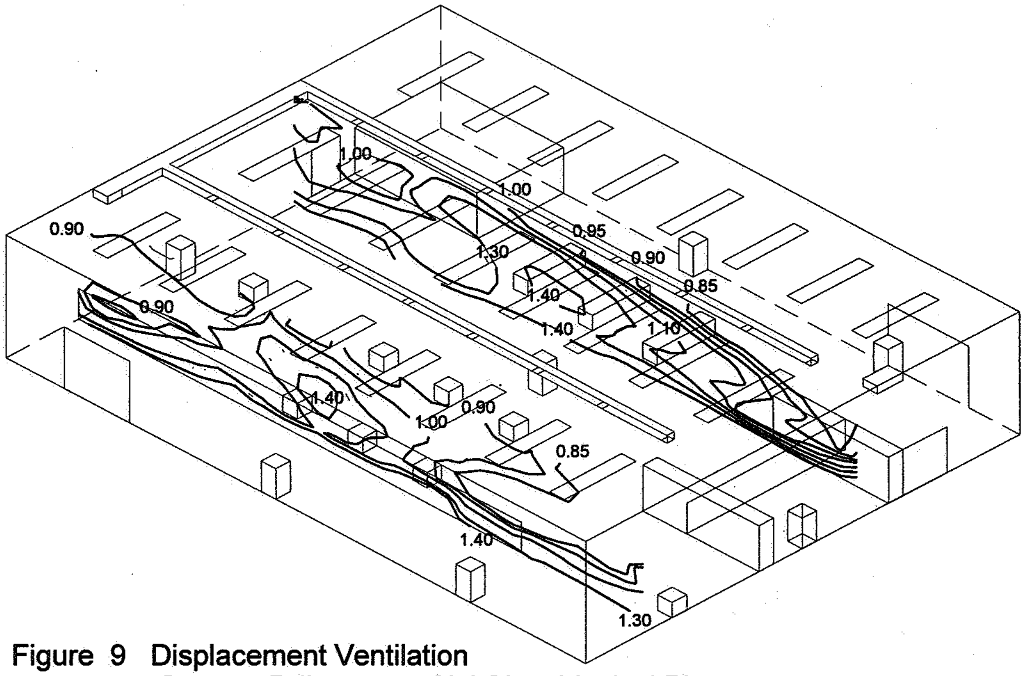

Figure 9 Displacement Ventilation
Contour Ddiagrams of LACI on Vertical Planes

7 Conclusions

The output from CFD analysis has been successfully analysed by the multizonal modelling technique to provide local values of ventilation effectiveness parameters. It has been applied to two distinctly different mechanical ventilation systems thereby demonstrating the usefulness of the technique as a design tool and the value of air change efficiency parameters in assessing the performance of buildings.

The results show that the two ventilation systems are working generally as expected. However the LACI contour diagrams for the displacement system show that the air distribution patterns are more complex than a simple analysis would suggest.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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NATURAL VENTILATION DESIGN FOR A CONCERT HALL

R R Cohen, R M Davies and M A Standeven

HGA Ltd, Burderop Park, Swindon, Wiltshire, SN4 0QD, UK

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HGA Ltd, Burderop Park, Swindon, Wiltshire, SN4 0QD, UK

SYNOPSIS

This paper describes the ventilation analysis undertaken during the design of a new music centre for which it was desired to avoid the use of air conditioning and conventional ducted mechanical ventilation. The main objective was to predict the thermal comfort of occupants in the centre's main auditorium during summertime performances. The analysis was done using computational fluid dynamics (CFD) and a dynamic thermal model.

The CFD results were used to decide the size and location of openings for natural ventilation, which led to the final design having a much better distribution of incoming fresh air than the initial design. The peak fresh air ventilation rate was reduced, but this did not significantly increase the risk of summertime overheating.

The dynamic thermal analysis predicted that the time when the temperature would be over 25°C ranged from 0.3 performance hours/year with a dense concrete roof construction and an orchestra of 30 to 3.5 performance hours/year with a lightweight roof and an orchestra of 100. Given that the larger orchestra would not be formal and so could wear lighter clothing, it was concluded that natural ventilation should be a viable strategy for controlling the risk of summertime overheating. However, given uncertainties regarding the usage of the space and UK summertime temperatures in the future, it was recommended that provision was made in the design to enable mechanical cooling to be added at a later date.

1 INTRODUCTION

1.1 Background

HGa were appointed by the Wiltshire Music Centre Trust as M&E consultants in a design team led by Feilden Clegg Architects to design a music centre in Bradford Upon Avon, Wiltshire. The focus of the building was a concert hall holding an audience of up to 410 people and a 100 piece orchestra. The 1750 m² facility also incorporated 3 large rehearsal spaces, a number of smaller practice and tutorial spaces, a recording studio, offices and a catering facility.

1.2 Objectives

The aim was to achieve a low energy building by an integrated design which incorporated high insulation levels, daylighting and avoided the use of air conditioning and conventional ducted mechanical ventilation. The proposed ventilation strategy was to provide fresh air by means of controlled natural ventilation in summer, and with fan assistance in winter. The benefits of a high thermal mass roof structure and night-time ventilation in terms of lower summertime overheating risk needed to be assessed. The daylighting scheme had to take account of the acoustic criteria and the need for blackout on some occasions.

1.3 Methodology

HGa undertook a detailed analysis of the concert hall auditorium to assess the likely thermal comfort of occupants (orchestra and audience) during summertime performances. The analysis comprised two integrated elements:

- Estimation using the CFD package FLOVENT of the air change rate induced by natural ventilation under worst case (no wind) conditions.
- Prediction of internal temperatures throughout the year using the dynamic thermal model APACHE and assessment of thermal comfort using these temperatures, the air speed predicted by the CFD and the anticipated activities and clothing of the orchestra and audience.

This paper describes the main results from the analysis and shows how the design developed to achieve the brief's low energy objectives.

2 BUILDING DESCRIPTION

The floor plan of the proposed auditorium is shown in Figure 1. It comprised a stage area, a small space for removable seating and permanent raked seating. A dominant feature of the auditorium was the roof lantern positioned above the stage (see the building sections in Figure 2). The lantern was intended to provide both daylighting and a route for stale air to leave the space via controllable dampers protected by louvres.

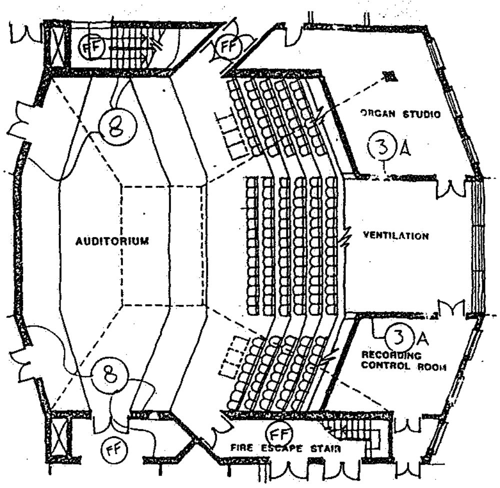

Figure 1 Floor plan of auditorium

In the proposed ventilation strategy, fresh air for the stage area was drawn from high level openings down two vertical builders work ducts at each end of the stage. From these the air would feed into a plenum running along the back of the stage and thence spill onto the stage through a series of low level grilles. Fresh air for the audience was designed to come primarily from a large louvred opening on the external facade which would supply a triangular cross section plenum running beneath the raked seating. The air would enter the auditorium through grilles in the step risers under each row of seats. Air extract from the auditorium was via louvred openings on each side of the roof lantern above the stage.

Section A - A

Section B - B

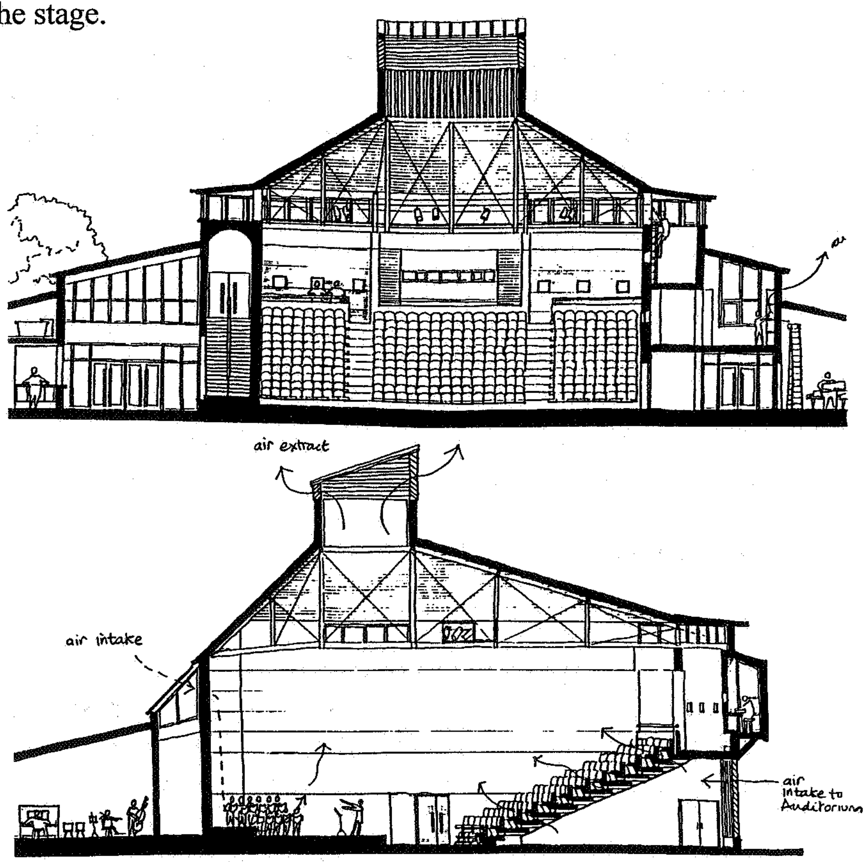

Figure 2 Building sections

3 AIR FLOW ANALYSIS

3.1 Initial Design Analysis

A full 3-D air flow computer model for the initial design of the auditorium was constructed using the FLOVENT CFD software. The objectives were to

- predict an overall bulk air flow rate for the space under worst case (no wind) conditions
- determine fresh air flow uniformity to the side banks of the audience
- assess the fresh air supply rate and uniformity to the orchestra
- predict air velocity around occupants (important for thermal comfort)
- identify any potential thermal hot spots

The model reproduced the geometry of the auditorium space and incorporated all the purpose built ventilation openings and pathways both to the outside and internally. Also included in the model were heat sources representing 100 musicians on the stage and 350 adults on the raked seating, with sensible heat emissions of 165 Watts and 80 Watts each respectively.

The results of this model predicted that the overall bulk air flow rate through the auditorium under worst case (no wind) conditions would be the equivalent of 8.3 ach (air changes per hour). With the given sizes of ventilation openings, some 90% of this total air flow went to the audience, giving each person on average 26 litres/second, and the remainder went to the stage giving each member of the orchestra an average of 9 litres/second.

The predicted air flow was very even between the centre and side banks of the same row, variations amounting to less than 10%. The air supply to the audience did however vary with the height of the row of seating, the lowest row receiving some 60% more fresh air than the top row ie about 34 litres/second per person compared with 21 litres/second per person respectively. The air flow pattern along the centre plane of the auditorium is shown in Figure 3. It confirms that there is no back-flow through the audience seating and that there is a strong plume removing the heat generated by the musicians.

The air flow model also predicted the air temperature distribution in the space. This was useful to see how much stratification there was and any potential 'hot spots', but because it is a steady state calculation (ie at a single point of time under fixed boundary conditions) the absolute values of temperature are not as meaningful as those predicted by a dynamic thermal model (see next section).

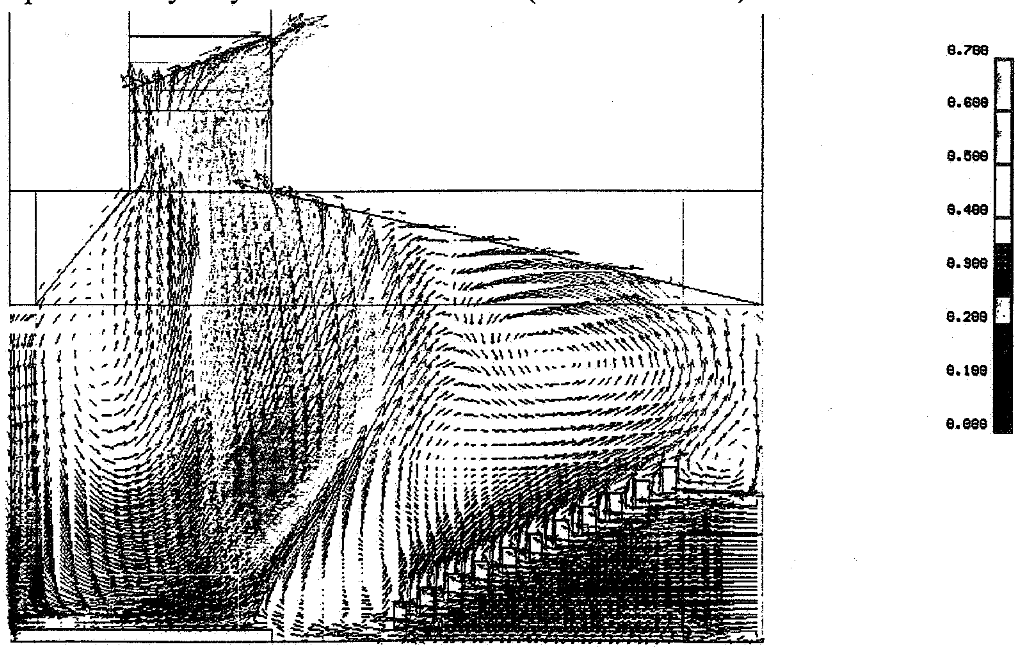

Figure 3 Air velocity vectors along vertical section through centre of auditorium.

3.2 Final Design Analysis

Subsequent to the analysis of air flow paths and rates in the initial design, the size of the ventilation openings for the auditorium, evolved due to two factors.

1. There was the desire to achieve a well daylit stage area for daytime rehearsals. With the initial design for the lantern, the skylight was completely occluded by noise attenuators suspended beneath the lantern to prevent noise breakout, and predicted daylight levels on the stage were very poor. This led to a proposal to divide the lantern into two halves, one to provide daylight and the other the pathway for air extract. To enable this proposal to achieve its aims successfully it was necessary to increase the size of the lantern throat from 6m by 4m to 6m by 6m (which also had structural advantages). The increased scale of the lantern throat enabled the daylight and air extract components of the lantern each to be 6m by 3m in plan. All the time it had to be borne in mind that the lantern design had to maintain a strict acoustic seal on the building and be able to achieve blackout without obstructing airflow.

2. There was a need to minimise the cost and therefore size of the acoustic attenuators on the auditorium air intakes and at the same time adjust opening areas to try to achieve a better balanced airflow between audience and orchestra.

The opening areas in the initial design were therefore adjusted and the new design was analysed in a 2D model using FLOVENT. (A 2D model of the initial design had been shown to give bulk air flows in good agreement with those predicted by the full 3D model.) The predicted bulk air flow rate for the revised design was 5.1 ach compared with the 8.3 ach predicted for the initial design which had considerably larger opening areas. In the new design it was predicted that some 76% of the total air flow went to the audience, giving each person on average 14.3 litres/second, and the remainder went to the stage giving each member of the orchestra an average of 15.8 litres/second.

These results indicated that the new proposed opening areas produced a much better distribution of the incoming fresh air between orchestra and audience than the previous design even if the overall fresh air ventilation capacity had been reduced. It is shown in the next section that the reduced ventilation capacity did not significantly increase the risk of summertime overheating.

The air supply to the audience still varied with the height of the row of seating, and it was recommended that the air distribution be made more uniform by adjusting the size of the free area of the openings in the risers of each step; all these openings were assumed to be the same size in the computer model (height 200mm, free area 20%).

4 THERMAL COMFORT ANALYSIS

4.1 Introduction

The dynamic thermal performance of the auditorium was simulated using the APACHE modelling package. The objective was to quantify the risk of overheating during summertime performances with different design options eg roof constructions and different possible occupancy levels. The computer model calculated the heat flows in and out of the auditorium and its structure through a whole year as they were driven by hour-by-hour changes in weather, lighting, occupancy, air flow rates etc.

4.2 Tested Options and Results

A summary of the different permutations analysed and the key result for each run is given in Table 1. The base case conditions were assumed to be a lightweight roof construction, a peak ventilation capacity of 8.3 ach, an orchestra of 30 and an acoustic curtain retracted during a performance.

Run no.	Roof construction	Orchestra numbers	Other variants	Performance hours/year >=25°C with performance every day	Probable performance hours/year >=25°C with performance every 10 days
WMCT1 (base case)	lightweight	30		11	1.1
WMCT2	lightweight	100		35	3.5
WMCT3	lightweight	100	Acoustic curtain during performance	42	4.2
WMCT4	Siporex	30		8	0.8
WMCT5	dense concrete	30		3	0.3
WMCT6	lightweight	30	peak ventilation 5.3 ach	11	1.1
WMCT7	lightweight	30	no rehearsal	9	0.9
WMCT8	Siporex	30	no rehearsal	6	0.6
WMCT9	Siporex	100		23	2.3

Table 1 Risk of overheating under various conditions

The simulations assumed that a full day rehearsal and an evening performance with a capacity audience occurred every day of the year. Given that only about 10% of all days will in reality have a performance, the predicted probability of the temperature exceeding 25°C is about one tenth of the values calculated by the computer simulations eg a result showing 11 performance hours per year over 25°C with performances every day, indicates a probability of only 1.1 performance hour per year over 25°C if there is one performance every 10 days. (The night purge assumed in the model effectively decouples one day from the next, so that the methodology employed avoids the possible effects of a build up of heat over a number of days.)

4.3 Discussion of Results

The threshold for summertime overheating was set at a resultant temperature of 25°C in view of the activity level of the performers, who are perceived to be the critical group in this regard. The value of 25°C was arrived at using the concept of predicted mean vote (PMV).

The PMV is the predicted mean vote, on a seven-point thermal sensation scale (-3 to +3), of a large group of persons when they experience a particular combination of air temperature, mean radiant temperature, humidity and air velocity with their metabolic rate and clothing insulation at given levels. An individual is deemed to find conditions acceptable if his vote is one of the three middle points of the scale (-1 to +1). Variance between individuals, and interestingly between the same individual on different occasions, means that even when the PMV is at the centre of the scale, statistically 5% of the group will have voted outside the central three points and are therefore considered to be dissatisfied. The International thermal comfort Standard ISO 7730, based on PMV, adopts the criterion that 10% uncomfortable is acceptable as a working maximum which statistically translates to the PMV lying in the range -0.5 to +0.5.

If members of the orchestra were wearing summer clothing ie the equivalent of long trousers and open-neck short sleeve shirts, and assuming an air speed of 0.4 metres/second which was predicted by the CFD analysis, their PMV at a resultant temperature of 25°C would be +1.15 given the assumed 'medium work' activity level. Wearing a jacket and tie under the same conditions would mean the PMV would rise to +1.7. Thus with summer clothing at this activity level, 25°C is on the border of the satisfactory range for an average person, albeit, on a statistical basis it translates to 33% of a large group being dissatisfied. With a jacket and tie at this activity level, 25°C is approaching the PMV level of +2 which is deemed "warm" in the ISO standard and translates to 60% of a large group being dissatisfied.

Performances are likely to entail either an orchestra of up to 30 who might normally wear the equivalent of jacket and tie or an orchestra of up to 100 who would find 'summer clothing' perfectly acceptable. It was concluded that with an orchestra of 30, a temperature of 25°C should represent a strict upper limit, whilst with an orchestra of 100 a small number of performance hours over 25°C would be an acceptable risk.

Under the base case conditions, the analysis predicted that there was a probability that only 1 performance hour in a typical year would incur a temperature over 25°C. This was considered to be an acceptable risk. With the size of the orchestra increased to 100, the risk of overheating increased to 3.5 performance hours per year, again an acceptably small risk given the associated lighter clothing level.

If the thermal capacity of the roof construction were increased by incorporating Siporex lightweight concrete, exposed on its underside, enabling it to absorb heat and moderate temperature rises, the risk of overheating would be reduced to 0.8

performance hours per year with an orchestra of 30 and to 2 hours per year with an orchestra of 100. With a dense concrete roof the risk of overheating with an orchestra of 30 would be reduced to 0.3 performance hours per year. The probable number of hours at or above 23°C during performances for the base case lightweight roof construction and with the same conditions except a dense concrete roof (runs WMCT1 and WMCT5) are presented in Figure 4. The results confirm that the dense concrete roof produces a general downward shift in warm temperatures of about 1°C.

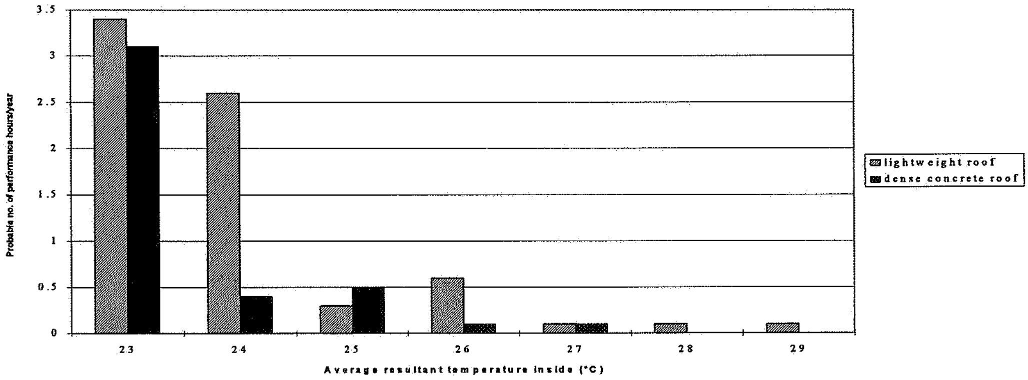

Figure 4 Frequency of internal temperatures at or above 23°C during performances

The run where the ventilation capacity was reduced from 8.3 to 5.3 ach showed no increase in overheating, which suggested that the lower overall fresh air ventilation capacity predicted for the new design with the reduced opening areas would not lead to a significant increase in overheating risk. The explanation for this result is that on those days when the internal temperature exceeds 25°C, the outside temperature is even warmer than the inside temperature, which means that the ventilation rate would be held back at its background level (to avoid introducing warmer air). Under these circumstances, a higher potential ventilation capacity is redundant.

5 CONCLUSION

CFD was successfully used to refine the design of a natural ventilation scheme for a concert hall. The size and location of openings were adjusted to obtain a uniform distribution of fresh air. Integrated analysis by a dynamic thermal model indicated that natural ventilation should be a viable strategy for controlling the risk of summertime overheating in the auditorium. However, the risk of overheating was not zero and there are a number of uncertainties regarding the usage of the space in the future. These factors, combined with the possibility that UK summers might continue to be hotter than the long term historical average, meant that it was recommended that provision should be made in the design to enable mechanical cooling to be added at a later date in case it proves to be necessary.

ACKNOWLEDGEMENTS

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

17th AIVC Conference, Gothenburg, Sweden, 17-20 September, 1996

A Technique for Controlling Air Flow through Modified Trombe Walls

Adrian C. Pitts and Stuart Craigen

School of Architectural Studies
University of Sheffield
Western Bank, Sheffield S10 2TN, U.K.

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School of Architectural Studies, University of Sheffield, Western Bank, Sheffield S10 2TN, U.K.

Synopsis

This paper describes an experimental investigation into the operation of a modified Trombe wall. The construction has been altered to include a layer of insulation material; two alternative positions for this insulation layer have been considered and tested. Air flow from the top of the Trombe wall has also been enhanced by the inclusion of a low power axial flow fan which was controlled to function dependent on measured temperature in the wall cavity. The construction with exposed concrete facing into the air cavity has a slower response than that with exposed insulation facing the cavity. The concrete option also produces higher outlet temperatures, though the fan was found to operate for longer periods with the insulation option. The controller used could be developed for employment in more sophisticated control strategies.

1. Introduction

An important aspect of passive solar design, and one which has a great influence over the ultimate level of success and efficiency of such designs, is the control of air movement. This is particularly the case with 'indirect gain' and 'isolated gain' categories, which often rely on air flows to redistribute solar heat gain. This paper deals with the design and experimental investigation of a modified indirect gain Trombe wall in which the air flow is of particular importance (though the general principles are of relevance to a number of other passive design options). Several related studies have been carried out at the University of Sheffield [1,2], and form the background to the work described here. The enhanced design employs automatic control of air flow and is thus specifically relevant to the theme of the conference - 'optimum ventilation and air flow control in buildings'.

2. Trombe Walls

Trombe walls evolved from the mass wall form of passive design. In the basic mass wall design, a high thermal mass wall is positioned behind a southerly facing expanse of glazing. Solar radiation passes through the glazing and is absorbed on the outer surface of the wall. The 'greenhouse effect' helps retain the heat which has been absorbed. Depending upon the thickness and thermal properties of the wall, the heat which has been absorbed will be released some hours later from the interior surface of the wall. In domestic situations, the wall can be designed to introduce a time lag suitable to delay heat gain to the interior until the most profitable time of the day (in terms of heating needs), that is until the occupancy period of the early evening. Figure 1 shows the main features of the mass wall.

An advance on the basic mass wall was the introduction of air flow openings at bottom and top of wall, as developed by Trombe. The air which has been warmed by solar heat gain in the cavity between wall and glazing, rises due to buoyancy effects and enters the room interior, whilst cooler air is drawn into the bottom of the cavity. Figure 2 illustrates the basic concept of the Trombe wall. Whilst the basic form relies only on natural air flow, there is the opportunity for a more sophisticated control system to be employed. In combination with a controller, there is also the potential for the wall to be used in a passive cooling, as well as heating, mode.

When initially developed, the concept of the Trombe wall was that air flow through the wall cavity (between glazing and absorbing surface), allowed heat absorbed in the cavity to be delivered more quickly to the interior of the building. Such a modification has been found to be particularly useful in climates with variable solar radiation intensity (the UK climate being a case in point). In these circumstances, the ability to extract and deliver the heat to the interior using the air flow is considered a more effective means of utilising the energy absorbed on the outer surface of the wall, (by comparison with the mass wall approach of allowing heat to slowly permeate through the solid wall). If a steady supply of solar heat throughout a substantial portion of the day is unlikely to occur, then it seems to be more effective to extract the heat using the air flow and deliver it immediately to the interior.

If alternatively, the system is to be used in a cooling mode, then provision must be made for the warm air to be vented directly to the exterior rather than allowing it to enter the occupied interior space. This can be achieved by providing an alternative outlet at the top of the wall, however this outlet needs controlling or positioning in some way relevant to the prevailing conditions, perhaps using some sort of automatic controller. Once a controller is installed, however simple it may be, then there is less reason not to develop the system a stage further with even better control options. This reasoning was fundamental for the study described here.

3. Wall Construction

In this study, variations on the basic wall construction have also been introduced. The reason for this was to attempt to optimise heat absorption and also to increase the level of thermal insulation between the interior and exterior. This lack of insulation has often proved to be the weak link in the thermal design of passive solar designs: whilst the sun is shining, some beneficial positive heat balance can be produced, at other times the high thermal heat transfer allows heat loss at a higher than acceptable rate (U value of the order of 1.0 $Wm^{-2}K^{-1}$, compared to currently acceptable wall U values of the order of 0.3-0.4 $Wm^{-2}K^{-1}$).

If the argument advanced in section 2 is accepted, that is that the main mode of solar heat extraction will be by using the air flow, then the heat gain through the wall itself becomes less significant and the extra insulation more acceptable and beneficial. In making this choice, the Trombe wall has been converted to act more in the fashion of an isolated gain collector.

Figure 1

Mass Wall Design

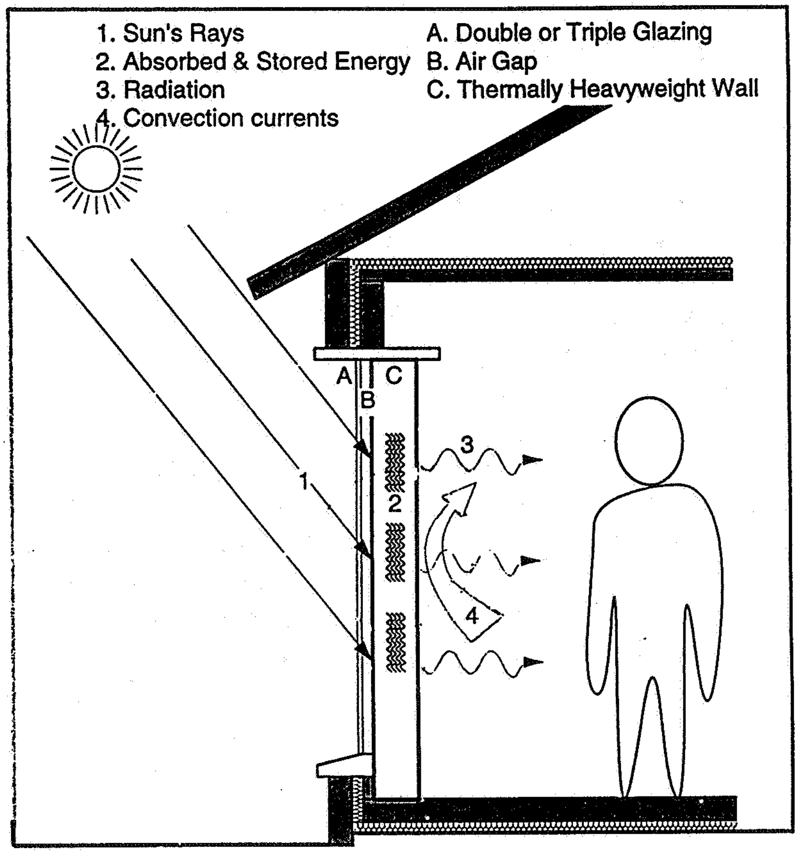
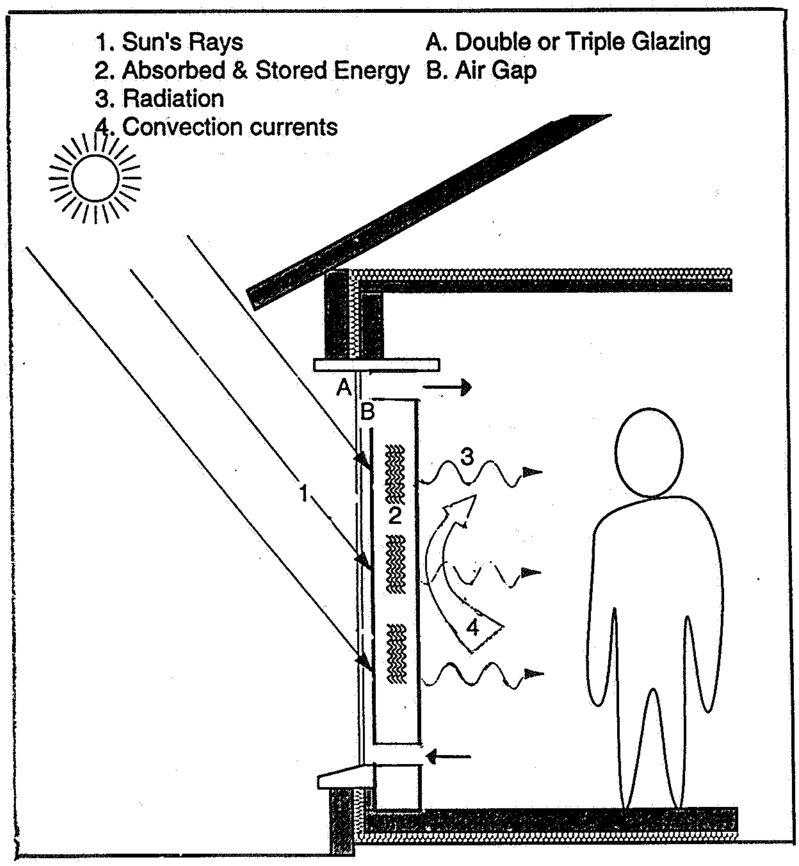


Figure 2

Trombe Wall Design



Two forms of wall construction were used for comparison in the tests performed. Each was of the same nominal U value (0.6 $Wm^{-2}K^{-1}$ approximately), however in one case the insulation layer was placed facing into the air flow cavity, whilst in the second it was placed facing into the room interior. The alternative face was concrete blockwork. Around the sides and above and below each construction, the wall was well insulated to reduce edge effects.

Simple, medium density concrete blocks, commonly available in the UK were used as supplied, with insulation (expanded polystyrene foam) attached to one surface. Unfortunately the absorptivity of each surface varied - the polystyrene being bright white, the concrete blockwork being fairly dark grey. despite this anomaly, the project continued with the blocks as they were supplied since the aim was to devise a simple system which required minimal specialist components.

4. Air Flow Enhancement

In a plain Trombe wall, natural buoyancy effects cause some air flow in the wall cavity, but it is largely uncontrolled. The heat gain to the air occurs from the solar absorption on the wall surface (and to a lesser extent from the interior surface of the glazing), the heat transfer is by natural convention. The attachment of a fan to the outlet from the wall permits control of flow, both time periods and flow rate. The heat transfer will be by forced convection from the interior cavity surfaces, and thus more effective for heat extraction. The air flow can also be controlled so as to deliver the heat either to the interior when this would be beneficial, or to the exterior when overheating would be likely.

Air flow from the modified wall design was enhanced and controlled using a low power (2.2W) axial flow fan attached to the top outlet of the wall section. The airflow generated by the fans was in the range 12 to 14 litres per second.

5. Experimental Investigation

In order to investigate the performance of the two designs a series of experiments were carried out. The two versions of the modified wall were used in a number of flow configurations - closed air flow loop; open without fan induced air flow; and with fan controlled flow. The results reported here deal with the last of these options. More details of the alternatives are available in Craigen[3].

Figure 3 shows the two test set-ups, the upper diagram is of the exposed concrete facing into the air gap, the lower is of insulation facing into the air gap. The position of fans, measurement points and solarimeter are all shown. the two wall configurations were positioned adjacent to each other in the south facing wall of the Arts Tower building at the University of Sheffield.

A data logger took measurements over a variety of weather conditions ranging from November 1995 to July 1996.

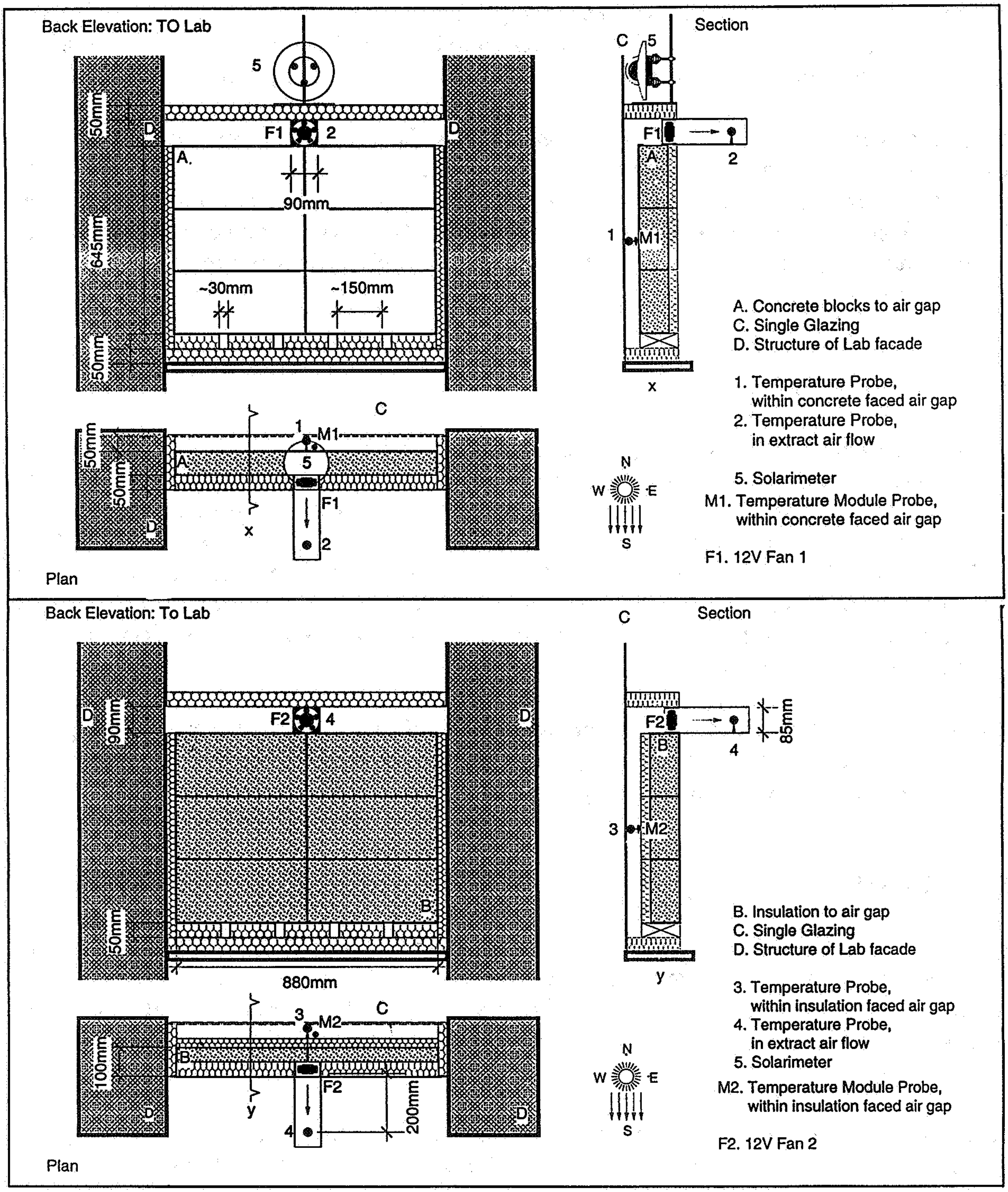


Figure 3 Experimental Arrangement
(Top: Concrete surface to air gap; Bottom: Insulation surface to air gap)

6. Control of Air Flow

A purpose designed controller was constructed using only easily available and relatively cheap components. Temperature sensors within the wall fed information to the controller which then decided to switch fans off or on. The switching point at which the fans operated occurred when the temperature measured within the wall air cavity exceeded 30°C, though any other value could have been used. There were two reasons for choosing this temperature. Firstly, the temperature of the air at the outlet needed to be above internal air temperature to derive any benefit, 30°C seemed a reasonable value to choose under these circumstances. Secondly, arising from initial tests on the walls, it had been noted that the temperature measured in the air cavity was above the air temperature due to radiant effects on the sensor which could not easily be eliminated.

7. Analysis

A range of comparative analyses were undertaken. In general it was found that the insulation to air gap construction produced a more rapid rise in temperature closely in phase with solar radiation intensity. The construction with concrete facing into the air gap exhibited a 1-2 hour delay in resulting temperature rise, however the maximum temperature was significantly higher. The switching sensor in the insulation to air gap wall appeared to be more affected by solar radiation intensity causing the fan to be switched on for longer periods, yet the temperature at the wall outlet was lower, and in some cases would have provided little useful heat gain.

Figure 4 illustrates comparative results for a typical day. The top diagram relates to the concrete facing into the air gap, and the bottom shows the insulation facing into the air gap.

Peak temperature rise (air temperature at outlet compared to room temperature) was 7.4°C. At the air flow rate in use this would deliver a heat transfer rate of abut 108W. The area of the absorbing surface was $0.57m^2$ and the solar radiation intensity $730Wm^{-2}$; this would indicate an available solar heat input rate of 416W and thus an operating efficiency of (108/416), that is about 26%.

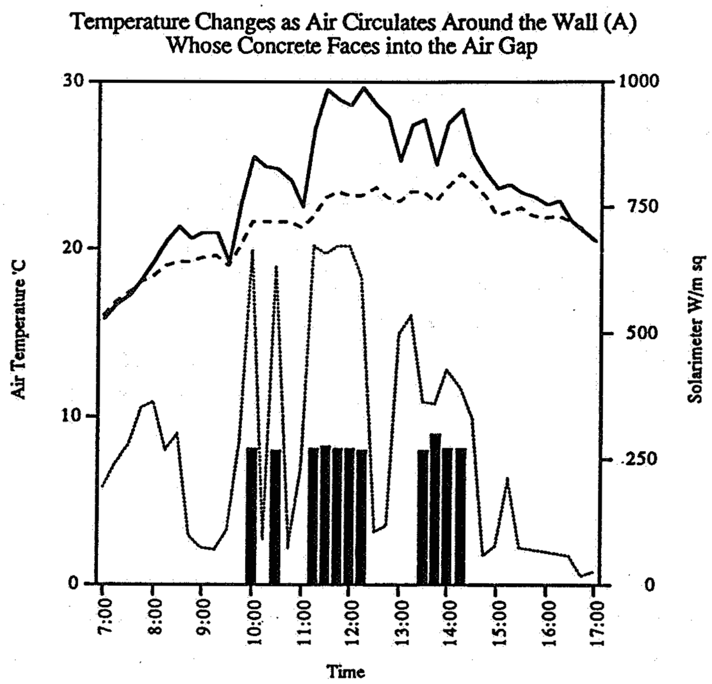
8. Control Strategy

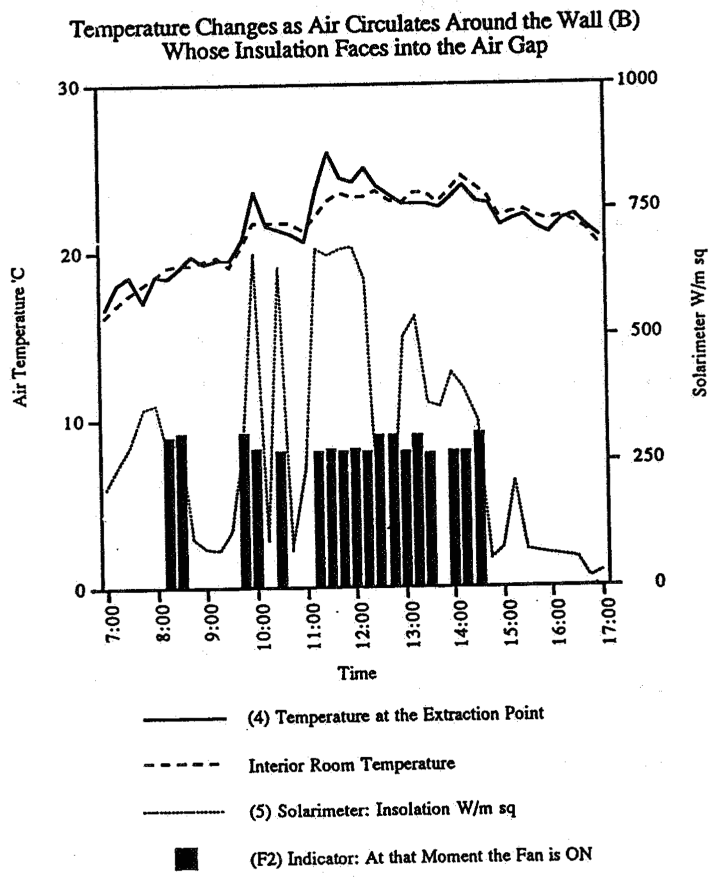
A relatively simple control strategy was employed in the system as tested, however a more sophisticated option would be quite feasible using a timed programmer to optimise heat extraction from the wall depending upon occupancy period and intended distribution (that is whether main function is as a heater or cooling mechanism).

Provision has also already been made within the controller to switch the fan not only in accordance with the temperature measured in the air cavity but also with reference to the room conditions.

Figure 4

Typical test results showing comparison of each wall type





9. Implications for Ventilation Design

A number of studies of passive solar design which have been carried out in recent years fail to demonstrate the expected benefits in terms of energy use reduction. One reason for this may be the over-reliance on natural energy and air flows. Work carried out at the University of Sheffield has shown that improvements in comfort and some reduction in energy use are possible if 'hybrid', or moderately controlled design options are employed. A level of control also engenders a more accepting response by residents.

On the basis of the work described in this and other papers it is suggested that controlled air flow is the key to successful functioning of passive solar design options. This has many implications for building ventilation design. The flow paths and the level of control to be applied must be considered at the sketch design stage. In particular the variations between winter heating and summer cooling operation must be included. The positioning of ventilation openings may have to be adjusted to match the overal 'system' requirements in order to optimise performance.

Of course, research is still necessary to improve designers' knowledge of the strategies, and the study described here has produced anumber of questions which deserve more attention.

The most critical aspect of successful design is the combining of commonly available components in relatively simple, yet robust, solutions which can be easily understood. It is hoped that this work makes some contribution to that goal.

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OPTIMUM VENTILATION AND AIRFLOW CONTROL IN BUILDINGS

17th AIVC Conference, Gothenburg, Sweden, 17-20 September, 1996

The Efficiency of Single-sided and Cross Ventilation in Office Spaces

by Martin White and Richard Walker

Building Research Establishment
Garston, Watford, WD2 7JR

THE EFFICIENCY OF SINGLE-SIDED AND CROSS-VENTILATION IN OFFICE SPACES

SYNOPSIS

This paper reports on work carried out at BRE to address the need for guidance on designing for natural ventilation via single-sided and cross-ventilation in office spaces and the limits of application in terms of plan depth. Present guidance suggests that natural ventilation will be adequate up to 6 m from the ventilating facade. This leads to the conventional design of offices up to 6 m deep on either side of a central corridor, giving as a rule of thumb a width of 15 m for a building with natural cross-ventilation.

The present work looks at the opportunities for going beyond these rules of thumb. The implications for thermal comfort and draught risk are also assessed. In the conclusions issues such as, local ventilation rates, ventilating air penetration from a facade, the use of artificial mixing (eg ceiling fans) on hot days, the position of windows, and means of enhancing internal air speeds and air change rates are discussed.

1. INTRODUCTION

In designing for natural ventilation, existing guidance[1] advises that air distribution will be 'reasonable' in naturally ventilated buildings with 6 m wide rooms either side of a central corridor. This results in the rule of thumb that 15 m is an effective limit for the width of the building. The same guidance may be interpreted as setting a limit of 6 m to the depth of a naturally ventilated space with single-sided ventilation. The work described below suggests that, depending on circumstances, this limit may be qualified or revised.

Effective natural ventilation must satisfy the following requirements:

- Overall air change rate must be adequate in winter for occupant health and safety, contaminant removal etc. and, in summer, to maintain thermal comfort.
- Distribution of fresh air must be even so that all occupied parts of the space receive an adequate supply.
- Air movement should be sufficient for thermal comfort in summer, but draughts causing discomfort or nuisance must be avoided.

If these can be satisfied, natural ventilation would be possible in deep open plan style rooms and buildings wider than 15 m.

The following key issues affecting the distribution of natural ventilation in deep office spaces are addressed:

- Single-sided ventilation compared with cross-ventilation
- Effect of window location
- Effect of partitions
- Possible impact of wind shelter

Measurements were carried out over a wide range of realistic conditions in several deep office rooms. Heated cylinders (approximately 100 Watts) were used to represent the influence of occupants and equipment on local ventilation rates. Window opening was usually set to maintain broadly typical indoor temperatures.

2. MEASUREMENTS AND RESULTS

Adequacy of air distribution with single-sided ventilation

Tests were carried out to assess air distribution in a deep office, 10.4 m deep normal to the windows, with single-sided ventilation[2]. It was found that local ventilation rates with single-sided ventilation were similar throughout the space. The variation was generally within about 15% of the mean, Figure 1 shows typical graphs of logarithmic tracer gas concentration against time for even and uneven air distribution. Wind direction appeared to be a key factor affecting the distribution of local ventilation rates. Distribution of air was more uniform with openings in the lee of the building, and less so when winds were generally light and towards the openings.

Comparison of air distribution between cross and single-sided ventilation

Tests were carried out to compare air distribution for single-sided ventilation with that for cross-ventilation[3]. The office was on the upper floor of a two-storey building and measured 9.7 m deep, normal to the windows. Alternate tests with single-sided ventilation and cross-ventilation were carried out in pairs on each day.

Cross ventilation and single-sided ventilation achieved similarly even distributions of local ventilation rate throughout the space. Room-average ventilation rates were generally less for single-sided ventilation when compared to cross-ventilation with the same open area, but similar when purely buoyancy-driven. In two cases the observed room-average ventilation rate was less than that expected due to buoyancy alone, illustrating that wind and buoyancy forces can combine in a way which is, as yet, not fully explained.

Comparison of single-sided ventilation in 'sheltered' and 'exposed' offices

Parallel tests were carried out in a ground floor office in a sheltered location[3] and the second-storey office. The sheltered office was 9.65 m deep normal to the windows, with the same window open area and orientation as the second storey office.

Ventilation flow rates in the sheltered office were either slightly greater or similar (within 19%) to those measured in the more exposed office in 83% of tests. This may be a consequence of low wind speeds during the tests and ventilation rates being dominated by temperature difference between inside and outside (ie buoyancy forces). In the remaining tests (17%), ventilation rates in the sheltered office were less (by approximately 10%) than in the exposed office . In these cases the wind speeds were much higher (4.0 m/s), and would be expected to dominate over stack induced ventilation, thereby increasing ventilation rates in the exposed office compared to those observed in the sheltered office.

Effect of window location (height)

There is evidence[4] to suggest that draughts may be avoided, and air distribution improved if windows are located higher above working level. Tests were carried out, in pairs in the second storey office, to assess the possible effect of the height of the opening above the office floor[3]. One test was with 'standard' window openings (ie as found) where the lower edge was 1.6 m above the floor, and about 0.85 m above desk level. The second test had 'high-level' windows, the lower half being blocked off to a height of 2.1 m above the floor, about 1.35 m above desk height. The same total open window area was used in a given pair. Air speeds were measured, at a height of 1.1 m, along the centre line normal to the windows. Figure 2 shows typical air speed profiles down the room for single-sided and cross ventilation with 'standard' and 'high level' windows.

Local ventilation rates throughout the space were similar for tests with either window type. With single-sided ventilation, air speeds varied only slightly along the depth of the room, with either high-level or standard windows. During these tests, wind speeds were quite significant, in the range 2.1 m/s to 6.1 m/s (typically around 4.0 m/s). In two of the test pairs, higher air speeds were recorded near the window (1 m into the room) with the standard window opening. The reason for this was not clear, although it was noted that in these cases the wind speeds were quite low, less than 2.0 m/s, and within one pair of tests there was a change of wind direction of some 120 degrees.

With cross-ventilation, air speeds near to windows located at high-level were lower than for the standard windows. This indicates that, for cross-ventilation, high level windows may reduce the risk of draughts nearby at working level. This may be particularly beneficial in deep, multi-occupancy rooms, by providing a more even 'cooling' effect for all occupants, and reducing the likelihood of the window being closed by the nearby occupant.

Effect of partitions

In many large open plan offices, work areas are marked out by partitions, usually lightweight panels approximately 1.5 m high. A pair of identical adjacent offices, 9.3 m deep normal to the windows, were used to study the possible effect of such partitions on air distribution and internal climate[3]. One office was partitioned into four equal areas, the other was left unobstructed. Partition sizes were based on commercially available partitions. Tests were carried out in pairs, one with standard windows the other with high level windows and repeated for both single-sided and cross-ventilation. It was found that local ventilation rates were similar in both rooms. Air speeds were generally low whether with or without partitions, over a wide range of external air speeds.

3. ANALYSIS OF DRAUGHT RISK AND THERMAL COMFORT IN DEEP OFFICE ROOMS

An assessment of draught, air movement and thermal comfort in deep offices with single-sided or cross-ventilation was made based on the above work.

Draught

A maximum air speed of 0.8 m/s[5] may be assumed above which nuisance may be expected (papers blowing) and thermal comfort may be difficult to maintain. Mean air speeds were generally below this; those measured at 1 m from the windows were below 0.8 m/s in 96% of tests. In a typical test, speeds exceeded 0.8 m/s for only about 5% of the time. Results show that at depths beyond about 1 m draught annoyance would not be expected to be significant for occupants. However, the 5% occurrence rate of intermittent higher speeds may indicate the likelihood of short duration gusts, which could cause annoyance or thermal discomfort if the incoming air is several degrees Celsius below room temperature.

Thermal discomfort may arise following the sudden opening of a window when ventilation is driven by temperature difference between inside and outside[6]. A possible approach to reduce these potential problems may be to provide a greater number of opening windows, at higher level, preferably opening in unison, for more even distribution of incoming air.

The applicability of the draught risk equation developed by Fanger et al[7] was assessed. Generally the calculated draught risk was low, due to the low air speeds. In some instances draught risk was high (eg 75 %) 1 m away from the window even though air speeds were moderate, ie in the range 0.1 - 0.34 m/s. This was primarily due to low air speeds in combination with relatively high turbulence levels. This indicates that the current draught risk expression may be inappropriate for natural ventilation in summer, especially when cooler incoming air may be regarded as pleasant. The underlying problem may be due to the original formulation of the draught risk expression from tests in an environmentally controlled chamber, in which subjects were exposed to forced air supply conditions.

Thermal comfort

The tests were not designed to directly assess the ability of a typical office room to maintain thermal comfort in summertime. However, the following results do give a reference perspective.

For tests in summer (ie. all except those in the partitioned rooms), external air temperatures ranged from 11 °C to 27 °C, while most internal air temperatures were in the range 17 °C to 27 °C, ranging from slightly cool to slightly warm on the ASHRAE standard scale[8] of thermal comfort (after Fanger[9]). The exceptions to the above were a case where internal air temperature reached 34 °C when the external temperature was also 34 °C; and two cases where the internal temperatures rose above external (eventhough this could have been avoided by opening more windows).

A desk study was carried out using existing design guidance procedures. Mean and peak internal temperatures were calculated using the Environmental Design Manual[10], which is based on the Admittance Method[5]. The manual is limited to maximum total internal heat gains of 15 W/m^2, which is low by today's standards. Currently internal gains in naturally ventilated buildings are typically [11] in the range 10 - 40 W/m^2, allowing for increased use of IT equipment. This highlights the importance of solar shading, for offices without external shading, solar gains are potentially the most dominant heat input to the room.

The desk study considered three categories of thermal mass, ie; 'light', 'medium' and 'heavyweight' (admittance values 8, 16 and 24 W/m^2/°C resp.) with windows shaded by internal Venetian blinds. Calculations showed, in medium to heavyweight buildings with such blinds, an 'intermediate' level of thermal comfort could be maintained for most of the summer, ie mean temperatures not greater than 24 °C and 'swings' not greater than 4 °C for 50 days in 10 years. External blinds significantly improved this to 30 days or less depending on thermal mass.

4. CONCLUSIONS

The key findings of the above measurements can be summarised as follows:

Ventilation and its distribution: Local ventilation rates are broadly even throughout deep spaces up to 10 m deep, for single-sided and cross-ventilation. Implying that, if adequate overall ventilation rates can be achieved, then ventilation will be adequate throughout the room. However, it is noted that other factors need to be considered when assessing the potential thermal comfort of occupants, eg proximity to windows, and local air currents.

Air movement and draught: Mean air speeds were low (unnoticeable) in about 75% of tests. Significant air speeds were measured in about 25% of tests, rarely exceeding the level which would cause draught nuisance (papers blowing, difficulty maintaining thermal comfort). Higher air speeds did occur occasionally during a typical test.

Air speeds at depth - 'rule of thumb': The measured results were dependent on many possible influences, not all of which could be isolated. Even so, results tend to indicate that air speeds fall from a maximum near the opening (measured at 1 m from the opening) to a minimum for cross-ventilation, or a uniform background level for single-sided ventilation, at or near the room centre 5 m into the space. This supports the existing rule of thumb of 6 m for the depth of penetration of fresh air, in that it appears to describe small scale locally unmixed air currents, not the distribution of local ventilation rates. The rule is more applicable to achieving adequate air movement through natural ventilation in summertime when we wish to provide rapid ventilation.

Proximity to windows and enhancing air movement: In offices deeper than 6 m, occupants not adjacent to windows may not share the benefit of cooler incoming currents and, on warm days, may find air movement inadequate for thermal comfort. One strategy to overcome this potential problem may be to mix incoming air more uniformly and enhance air movement, using (say) ceiling fans. An additional benefit of increasing air speeds artificially is that the range of thermal comfort may be extended by up to 3 °C, without the need to increase ventilation rates. Another strategy, mentioned below, is to optimise the window opening design.

Draught: Where air speeds were significant, they could cause thermal discomfort if combined with low outside air temperatures, or possibly gusts. Improved window design (described below), and possibly artificial mixing should help reduce these problems. Draught risk was calculated using the expression developed by Fanger. Calculations were dominated by low air speeds combined with relatively high turbulence levels, and would appear to undervalue the beneficial cooling effect of air movement in summer conditions. This suggests that there is an underlying problem in applying the Fanger draught expression to natural ventilation in summer, for which it was not formulated.

Window design: With cross-ventilation, higher level windows may reduce draughts nearby at working level. Results suggest that neither the overall ventilation rate nor its local distribution is significantly affected by such small changes in the height of the window opening above the floor. A greater number of smaller openings, placed at high level, may be expected to produce smaller 'jets' which will dissipate more quickly and allow incoming air to warm-up by increased "contact" with internal air. This may help to reduce possible thermal discomfort caused by large 'unmixed jets' of cool air entering through a few large openings.

Effect of partitions: For simple partition layouts, and two different window heights, local ventilation rate distribution was relatively even over a 10 m deep narrow office space, for both single-sided and cross-ventilation. Air speeds were generally low with or without partitions for a wide range of external air speeds.

Thermal comfort: With few exceptions the measured internal climate stayed within conventional comfort limits. Calculations showed that, in medium to heavyweight buildings with internal venetian blinds, an 'intermediate' level of thermal comfort could be maintained for the greater part of the summer, and was significantly improved with external shading.

In conclusion the following design guidance is proposed based on this research:

- Rule of thumb that ventilating air penetrates to about 6 m from an open window applies only to air speeds and unmixed incoming air currents.
- Local ventilation rates are generally evenly spread in deep office rooms (up to 10 m deep) either with cross- or single-sided ventilation
- Higher level windows may reduce draughts at working level and adjacent to windows particularly when cross ventilating.
- Distributed openings may avoid thermal discomfort problems due to draughts.
- Artificial mixing (eg ceiling fans) may avoid unmixed air currents local to an opening, producing more evenly distributed conditions for comfort; in warm conditions, increased air movement also extends the range of thermal comfort.

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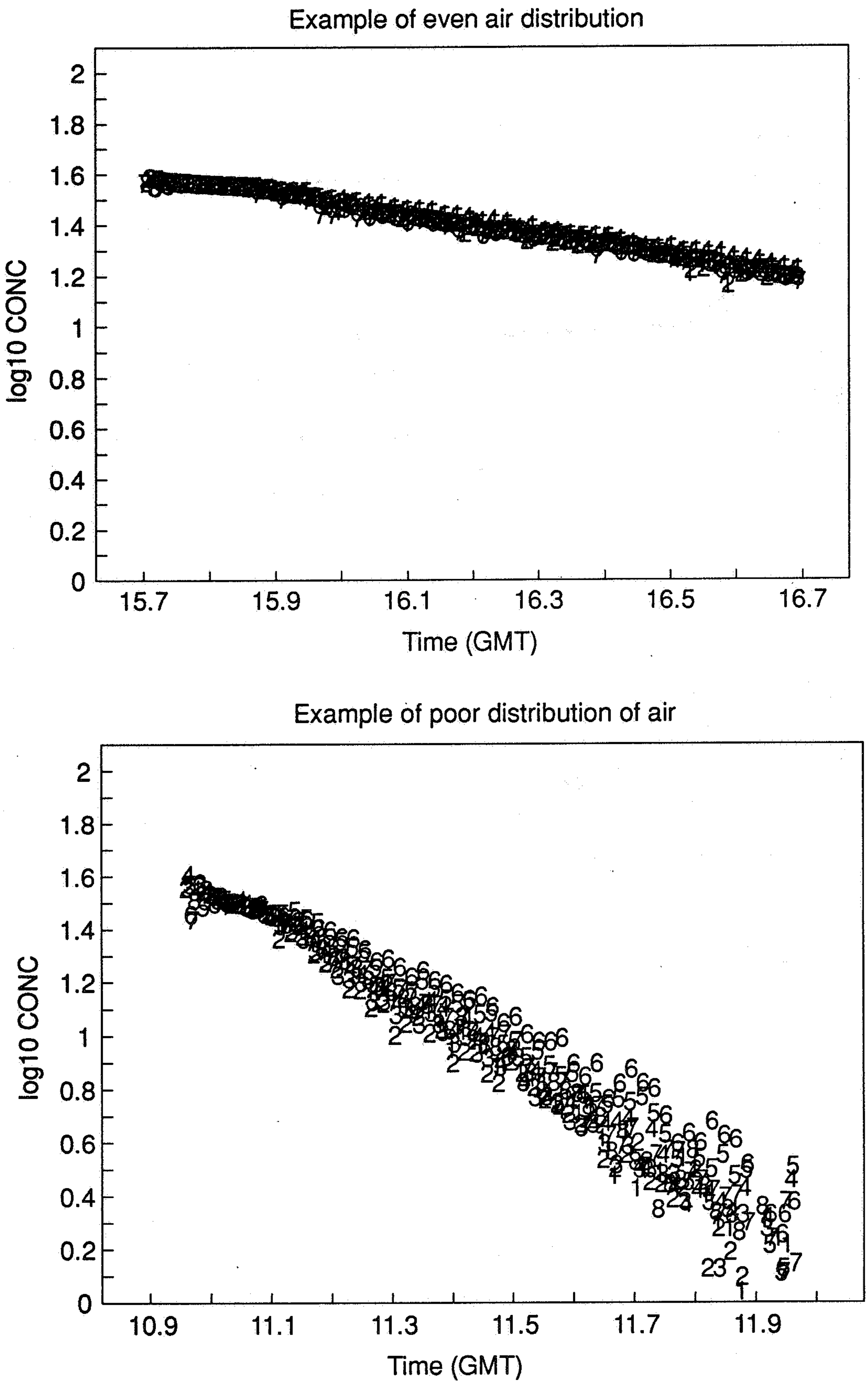

Figure 1. Examples of typical tracer gas decay curves

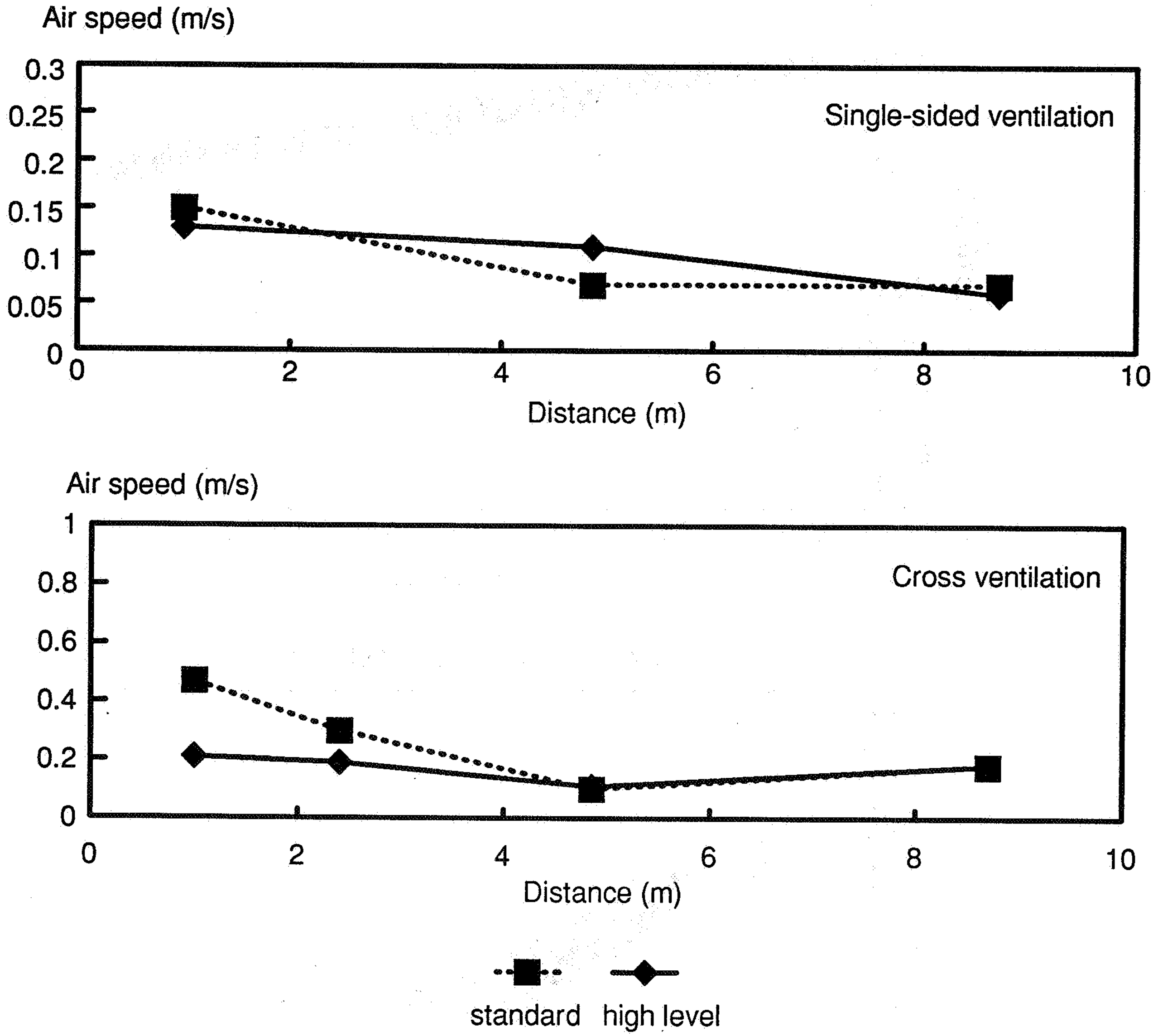

Figure 2. Air speeds at increasing room depth for standard and high level windows

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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The Evolution of Ventilation in Manufactured Housing in The Northwestern United States

D. Stevens*
M. Lubliner**
B. Davis***

*Stevens and Associates, PO Box 398, Keyport, WA 98345 USA
**Washington State University Energy Extension, POB 43165, Olympia, WA 98504 USA
***Ecotope Inc., 2812 E Madison, Seattle, WA 98112 USA

Synopsis:

Electric utilities in the Pacific Northwest have spent over $100 million to support energy efficiency improvements in the HUD-code manufactured housing industry in the Pacific Northwest over the past several years. Over 65,000 manufactured housing units have been built since 1991 that exceed the new HUD standards for both thermal performance and mechanical ventilation that became effective in October, 1994. All of these units included mechanical ventilation systems that were designed to meet or exceed the requirements of ASHRAE Standard 62-1989. This paper addresses the ventilation solutions that were developed and compares the comfort and energy considerations of the various strategies that have evolved in the Pacific Northwest and nationally. The use and location of a variety of outside air inlets will be addressed, as will the acceptance by the occupants of the ventilation strategy.

Key words/terminologies

BPA -- Bonneville Power Adminstration, a United States Department of Energy power marketing agency in the Pacific Northwest

DAPIA -- Design Approval Primary Inspection Agency -- Technical agency which reviews manufacturer designs to determine compliance with FMHCSS

FMHCSS -- Federal Manufactured Home Construction and Safety Standards -- Federal safety and performance standards for manufactured homes

IPIA- - In-Plant Primary Inspection Agency -- Inspect home construction process (in factory) to ensure compliance with DAPIA-approved designs

RCDP -- Residential Construction Demonstration Program -- Research and demonstration program funded by the Bonneville Power Adminstration and operated by the state energy offices in the Pacfic Northwest

Spot ventilation -- Removal of air from bathrooms and kitchens during bathing and cooking.

Ventilation effectiveness (temporal) -- Time-weighted ventilation: getting the ventilation to the occupants when they need it.

Ventilation effectiveness (spatial) -- Zone-weighted ventilation; getting ventilation to zone(s) occupants occupy.

Whole house ventilation -- Devices designed to provide fresh air to all home zones and remove stale indoor air from these zones.

Introduction

Pacific Northwest electric utilities have invested over $100 million to improve energy efficiency in HUD-code manufactured housing. Over 65,000 manufactured homes have been built which exceed the 1994 Federal Manufactured Housing Construction Safety Standard (FMHCSS) thermal requirements and which rely on mechanical ventilation systems similar to those required by the 1994 FMHCSS. Research projects have analyzed ventilation system performance, energy costs, and occupant acceptance of various systems. These data have been used to develop ventilation approaches that meet the needs of both the housing manufacturers and the occupants.

The paper describes the evolution of manufactured home ventilation systems in the Pacific Northwest. System hardware is discussed, as are controls, occupant comfort, and operation issues. Performance of ventilation systems included in recently constructed homes is reported. The 1994 FMHCSS ventilation requirements use some of the information developed in the Pacific Northwest, but may still fail to meet ASHRAE Standard 62-1989. The research results include discussion of whole house air leakage, exhaust fan performance, HVAC system and building envelope interactions, and occupants' acceptance and operation of ventilation systems.

HUD-Code Housing Background

Manufactured housing is constructed on permanently affixed metal frames in factories, and shipped on wheels to sites where they are set up as single or multiple section units. The technology evolved out of World War II, combining cheap and plentiful surplus steel frames with the inexpensive, yet functional, interior design found in boat interiors. Mobile homes grew in size, features, and popularity throughout the '60s and '70s. Currently, manufactured homes comprise about 10% of all single family housing in the U. S. (Krigger 1994). In parts of the rural U. S., manufactured homes account for over 50% of single-family housing.

Prior to 1976, manufactured homes were more commonly referred to as "mobile homes" or "trailers" and were often built with minimal insulation and low-cost HVAC systems and provided low ($<\$20/ft^2$) first-cost new housing. Insulation and safety standards for these homes were codified primarily in the Uniform Building Code (UBC) and various ANSI standards. In 1974, in response to growing concern that the industry be regulated by a single body, Congress enacted the National Manufactured Housing Construction and Safety Standards Act. Two years later, HUD adopted national industry building standards known as the Federal Manufactured Housing Construction and Safety Standards (FMHCSS) and became the regulator of the industry. Day-to-day regulation of the industry is carried out with the cooperation of state inspection agencies which are under supervision by HUD's technical contractor, the National Conference of States on Building Codes and Standards (NCS/BCS).

In 1994, the FMHCSS thermal and ventilation requirements were updated at the request of Congress. A national research laboratory developed revisions to thermal standards, which they estimated would provide about $300 million in annual national energy savings to new homebuyers over current practice, and about $400 million compared to the 1976 FMHCSS. (Conner et al., 1992)

Infiltration-induced heating and cooling were estimated to account for approximately 20% to 30% of the total load in a typical manufactured home. The lab analysis did not attempt to evaluate energy conservation measures that would lower infiltration rates. Thus no changes were made in the 1994 FMHCSS for infiltration control options. This decision was based on several assumptions:

1. No requirement was clearly present in the enabling federal legislation to include additional infiltration reduction measures in the 1994 FMHCSS.
2. New manufactured homes were relatively airtight, so very low natural infiltration rates would result from further tightening. HUD has estimated the average natural infiltration rate in new manufactured housing to be 0.25 air changes per hour (ACH). Various studies of airtightness in Pacific Northwest homes (reported later in this paper) confirm that newer manufactured homes are indeed built to this level of airtightness.
3. In the absence of mechanical ventilation, very low infiltration rates can have significant impacts on occupant health. A ventilation standard was not proposed in the national lab's recommendations because "standards to mitigate health effects of very low levels of infiltration are difficult based on the current state-of-the-art, and would require further study." (Conner et al., 1992). There were also the practical concerns of specifying a target infiltration rate and assigning responsibility in the event of noncompliance. Manufacturers have no direct control over the home's site set-up. Inadequate structural support or air-sealing can increase infiltration levels significantly, especially at the marriage line between halves of a double-wide unit. Occupant control of mechanical ventilation systems is also a major determinant of ventilation effectiveness.

Ventilation effectiveness is more concerned with pollutant removal from the home rather than just the time-weighted average rate of building infiltration. Depending on the rate of natural infiltration, the interactions of the natural and mechanical ventilation, and the duty cycle of the mechanical ventilation system, the average ventilation rate and effective ventilation rate may be very different. The lab study suggested that "Although there is significant potential energy savings associated with reduced infiltration, this recommendation must be accompanied by ventilation measures that mitigate potential IAQ problems." (Conner et al. 1992) It further says "A reduction in infiltration levels should be accompanied by:

1. A clear definition of the minimum ventilation/infiltration rates required for occupant health and moisture control.
2. A practical and economical method for determining the maximum ventilation/infiltration rate and minimum ventilation/infiltration rate in commercially produced homes.
3. A clear definition of the ventilation characteristics other than rates (such as ventilation control and distribution) required to assure a healthy environment in a low infiltration home." (Conner et al., 1992)

Based on recommendations from National Institue of Standards and Technology (NIST) and their consultants (TenWolde and Burch, 1993), HUD ultimately decided to require that occupant-controlled, whole house, mechanical ventilation systems be installed, without requiring any additional airtightening measures beyond the 1976 FMHCSS.

FMHCSS Ventilation Requirements

The FMHCSS ventilation requirements include short-term spot ventilation specifications and whole house volumetric and distribution specifications.

Bath Spot Ventilation

The 1976 FMHCSS required bathrooms to have minimum size operable windows (1.5 ft^2) or mechanical "spot" ventilation using bathroom exhaust fans. This gave occupants the ability to exhaust indoor pollutants, such as moisture generated during bathing, by opening a window and/or switching on the bath fan. In 1994, the FMHCSS began requiring mechanical bathroom fans for spot ventilation in rooms with tubs or showers, but still allows only an operable window to ventilate toilet compartments.

Kitchen Spot Ventilation

The 1976 FMHCSS required kitchen spot ventilation to exhaust indoor air pollutants created during cooking, such as moisture and combustion by-products. Requirements included one of the following:

Kitchen range hoods

or

Passive stack ventilation with 12.5 square inches of area within 10 feet of the range

or

An operable window within 10 feet of the range

In 1994, the FMHCSS eliminated the use of operable windows to satisfy kitchen range spot ventilation requirements. Kitchen fans in the HUD homes required 90 CFM (42 L/s) installed capacity. Standard industry range hoods were installed which either exhaust through the roof (similar to the bath fan) or exhaust directly through an adjacent exterior wall. These fans have backdraft dampers installed.

Whole House Ventilation (1976 FMHCSS)

Whole house ventilation is designed to dilute and remove indoor pollutants which may not be associated with cooking and bathroom use. The 1976 FMHCSS required a home to have either 4% of floor area in operable windows or mechanical ventilation installed to provide "whole house" ventilation. Many manufacturers built 1976 FMHCSS homes using systems sold by the furnace manufacturers that integrate ventilation into furnace operation by using a fresh air duct from the outside to the furnace. These systems had the potential to pressurize the home and drive moisture into the attic, causing condensation.

Similar systems also sold by furnace manufacturers are commonly used by manufacturers to provide both whole house ventilation and mechanical attic ventilation. These systems introduce outside air into the homes interior and the attic cavity. A high-volume fan (about 300 CFM or 142 L/s) pushes air through the attic cavity and out attic vents. In newer homes with predominantly vaulted ceilings and hence limited free volume in the ceiling assembly, the usefulness of these fans may need further investigation.

Whole House Ventilation (1994 FMHCSS)
The 1994 FMHCSS ventilation standards require a mechanical occupant-controlled ventilation system. The system cannot be a dual spot/whole house fan located in the bathroom because of HUD's concern that spot ventilators are not designed for long duty cycles. The new requirements also specify that the occupant must be able to control the system with an accessible on/off control, and that the ventilation system must not create a net positive or negative pressure across the building envelope, in an effort to minimize envelope structure damage from elevated moisture levels.

The revisions in the FMHCSS ventilation requirements were an attempt by HUD to improve both temporal and spatial ventilation effectiveness in manufactured homes. A good faith effort was made to adopt some of the ventilation system ideas used in the Pacific Northwest, followed by a consensus and rule making process. These processes often result in requirements which may not accomplish all the original intent. The FMHCSS ventilation requirements do not address fan noise level, operating cycle, or specific thresholds of pressurization to avoid. The final FMHCSS requirements should improve ventilation in manufactured housing to the extent they are correctly designed and installed by the housing manufacturer and are used by occupants as intended. However, the lack of specificity in these requirements has raised considerable debate in the Pacific Northwest. Some of this debate is recounted in the discussion section of this paper.

BPA Ventilation System Specifications

BPA's specifications were designed to ensure adequate spot and whole house ventilation. A variety of whole house systems were authorized by BPA. By far the most common system installed in the early SGC and MAP manufactured homes was the integrated spot/whole house option based on two nominal 50 CFM (24 L/s) bathroom fans controlled by 24-hour timers.

Bath fans in most Pacific Northwest manufactured homes built to the 1976 FMHCSS were generally supplied by one manufacturer as a ceiling-mounted fan. The bathroom exhaust air was ducted to the vicinity of a roof jack or connected directly to an exhaust termination fitting provided by the fan manufacturer as a package with the bath fan. The bath fans had a plastic sponge-type backdraft damper which opens when the exhaust fan operates, and which tend to leak air when the bath fan is not operating. These backdraft dampers also tended to disintegrate and be blown away over time. When they fail, these devices often become seven inch passive stack ventilators.

Until changes in the FMHCSS caused manufacturers to consider improved fan products, the fans installed for dual duty (spot and whole house ventilation) were very inexpensive units which were not necessarily intended for long periods of daily use. This fact is important to remember when considering the long-term effectiveness of this ventilation system.

Kitchen spot ventilators were required to deliver 90 CFM (42 L/s) and were ducted either vertically or horizontally. Kitchen ventilators are not a primary focus of this discussion.

BPA-certified ventilation systems are required to meet both volumetric specifications and distribution specifications. The National Environmental Protection Act (NEPA) requires federal agencies to conduct an environmental impact statement (EIS) and ensure "findings of no significant impact" associated with actions or programs that could cause indoor air quality problems. BPA conducted a series of studies under the EIS process which resulted in requirements that mechanical whole house ventilation systems be installed in any homes involved with BPA programs. These programs have indeed resulted in construction of more air-tight homes.

The BPA whole house ventilation specifications were designed to provide for introduction of outdoor air to occupied zones (primarily bedrooms) in accordance with ASHRAE Standard 62-1989 and exhaust of stale indoor air to attain the seasonal average ventilation rate specified by Standard 62 (the greater of 0.35 ACH or 15 cfm or 7 L/s per person).

The problem with standards is that they are simplifications of a more complex interaction between building tightness, occupancy effects, and ventilation system effectiveness. Many indoor pollutants are associated with the level of occupancy. For CO_2 levels or biocontaminants that transmit disease from one person to another, ventilation systems should be governed by the number of occupants in a building. Relative humidity levels also correlate with occupancy level as more occupants bathe, cook, wash and emit water vapor. In the RCDP home occupant and site survey analysis (Palmiter et al., 1992), the volume of the building has little or no correlation with the number of occupants living there. The size of the home tends to be strongly correlated with family income but only weakly correlated with the number of occupants.

Under a volume-based specification, a very large home with two occupants must be ventilated at a higher level than a small manufactured home with four occupants. Although the intent of the specifications is to ensure 0.35 ACH, the SGC and MAP specifications are not directly based on volume. Instead, they are based on the number of bedrooms in the home, which is more closely correlated with number of occupants. BPA requires a whole house mechanical ventilation system with a measured installed capacity of providing 45-90 CFM (21-42 L/s) in 1-4 bedroom homes, respectively, as shown in Table 1. An additional 15 cfm (7 L/s) beyond the normal ASHRAE 62 level is specified for the main living area because of high occupant levels.

Table 1
SGC/MAP Performance Specifications for Whole House Ventilation

# Bedrooms	Measured Flow
1	45 CFM (21 L/s)
2	60 CFM (28 L/s)
3	75 CFM (35 L/s)
4	90 CFM (42 L/s)

To ensure a minimum exhaust capacity exists for bathroom spot ventilation (50 CFM or 24 L/s) and whole house fans, system flow rate testing is provided by staff from the state energy offices to manufacturers whenever any design changes are made to the spot and/or whole house fans, ductwork or exhaust termination devices. Manufacturers use small flow hoods provided by state energy offices to test flow rates and determine if they comply with the performance specification flow rates as part of the design and commissioning process.

A maximum noise rating of 1.5 sones is required on all whole house exhaust fans. This is reduced to 1.0 sone for fans operating continuously. Sone ratings were specified in hopes that occupants would operate bath fans for whole house ventilation on a duty cycle of sufficient length to actually increase the effective ventilation rate of the home. This was born out in a recnet survey by the Washington State Energy Office of 50 MAP home owners. The systems used by Pacific Northwest manufacturers in 1995 are summarized in Table 3 and described in more detail below.

Table 2

Ventilation Systems used by Pacific Northwest Manufacturers in 1995 to meet MAP ventilation specifications

System Type	Number of Manufacturers	% of Manufacturers
(1) Combination bath/whole house fan	7	33%
1.5 sone		
Window inlet vents		
24-hour timer control		
(2) Dedicated whole house fan	8	38%
Continuous operation		
0.5-1.0 sone		
Window inlet vents		
On-off switch control of whole house fan*		
(3) Dedicated whole house fan	1	5%
Intermittent operation		
1.5 sone		
Outside air to furnace		
Timer controls furnace and whole house fan		
(4) Dedicated whole house fan	3	14%
Continuous operation		
0.5-1.0 sone		
Outside air to furnace		
On-off switch control of whole house fan		
(5) Balanced flow HRV system	2	10%
Connected to furnace		
Timer control of HRV system		
Total =	21	100%

* One of these uses a timer instead of on-off switch
Options 2-5 also require spot ventilation fans in the bathrooms

Combination Spot/Whole House & Dedicated Whole House Exhaust (Systems 1 & 2)
The most common ventilation systems employed in these manufactured homes under the SGC program were (1) combination bathroom and whole house exhaust fans and (2) dedicated whole house fans. These systems are designed to provide general ventilation at least during periods of occupancy. The combination system provides the bathroom "spot" ventilation needs of occupants when using the bathroom, with whole house ventilation provided via operation of a 24-hour timer. Fresh air inlet vents are supposed to provide the necessary fresh air in each bedroom and in the main living space (even though the performance of these vents , as currently sized, probably adds little additional ventilation to the home, as discussed below.)
This system ran afoul of the 1994 FMHCSS because it relied on a spot ventilator for whole house ventilation. HUD's issue was the use of inexpensive bath fans, but the FMHCSS mandated that the whole house fan could not be located in the bathroom. Many manufacturers switched to a dedicated, continuous fan for whole house ventilation when the federal specifications changed. By doing this, they also met MAP specifications.

The dedicated system cannot be located in a bathroom and must use a separate ventilator for whole house ventilation. Generally the dedicated whole house fan is located in a hallway, operates continuously, is rated at 1.0 sone or less, and uses an on-off switch rather than a 24-hour timer to secure the fan during periods of long vacancy. The 0.5 and 1.0 sone exhaust fans typically use 15-17 watts to run the fan motor, which is considerably less than the 1.5 sone combination fans' typical 60-75 watts. On an annual basis, the parasitic energy cost of the quieter continuously-operating fan is less than $10 in most Pacific Northwest locations. This dedicated fan is also of considerably higher quality than most bath fans and therefore expected to last much longer.

Furnace-Based Systems (Systems 3 & 4)
Some manufacturers use an outside air intake system running from the roof to inside the furnace cabinet. This duct supplies outside air which is distributed by the furnace fan. In newer systems, a timer on the furnace fan allows for occupant- controlled ventilation independent of the furnace thermostat. While these systems reduce construction costs by eliminating need for passive air inlet window vents, they rely on the furnace fan (generally drawing at least 350 watts), to distribute air, consuming significantly more parasitic energy than fan/air inlet systems. No evaluation of the newer furnace based systems that ventilate independently of the thermostat has been conducted.

Several of the manufacturers have used "flapper dampers" to relieve possible pressure differentials caused by the operation of furnace-based ventilation. The 1994 FMHCSS require the use of a device to relive this pressure, but give no standards for what level of pressurization is acceptable and what size the damper needs to be. More research is needed in this area.

Balanced Flow Systems (System 5)
A few manufacturers have tried an innovative balanced flow system which preheats outside air to the furnace with exhaust air using a heat recovery ventilator (HRV) core and fan. These systems are attractive since they are designed to integrate into the furnace ductwork but work independently from the furnace operation. They do not cause pressure imbalances like other

systems and they provide some heat recovery and tempering of outside air. Evaluation of field installations of this system has been limited and most manufacturers have dropped this approach.

Through-the-wall HRVs which are designed to provide ventilation independent of the furnace ductwork are not approved for the BPA program unless units are installed in the main living area and each bedroom (to meet fresh air distribution requirements). It is unclear if an HRV installed in one zone of the home can provide adequate spatial ventilation effectiveness.

Air Inlet Vents

Air inlet vents as employed in the Pacific Northwest were designed to improve the distribution of outdoor air to occupants if a furnace-based distribution system is not used. Window sash vents or through-the-wall vents are installed in each bedroom and in the main living area. These vents are designed to provide small amounts of outdoor air to bedrooms and living zones to improve the ventilation of the living zones typically occupied. Occupants are instructed to leave these vents open and to only close them for brief periods when cold drafts or dust storms are encountered.

Outdoor air intake occurs whenever a negative building envelope pressure exists. Manufactured homes commonly are depressurized slightly (0.5 to 1.5 Pa) when interior fans operate. Tests run on window inlet vents in laboratory settings have achieved air flows close to that recommended by ASHRAE Standard 62 (15 CFM (7 L/s) per bedroom), but only at pressure differentials far above those commonly encountered across exterior bedroom walls; that is, the test had to be conducted at 15-20 Pa (depending on the mesh size of aluminum screens installed with these vents) to produce air flow of 15 CFM (7 L/s).

One of the authors has conducted tracer gas measurements of air flow through a window vent installed in a typical MAP home bedroom (Davis, 1996). The vent measures roughly 1/2" by 14", and has a net free venting area estimated at 2 in^2 (Luoma et al., 1988). With two exhaust fans running (combined exhaust flow of 110 CFM (52 L/s)) and the interior bedroom door closed, air flow into the bedroom increased by 3.65 ±0.16 CFM (1.72 +/-0.076 L/s). The test was conducted with a single tracer gas and therefore does not estimate the amount of air entering the room through interior wall penetrations. Another study (Palmiter et al., 1996), which measured ventilation in new Pacific Northwest multifamily buildings, found the vents admitted an average of 3.1 CFM (1.46 L/s) of outside air when the building was subjected to negative pressures resulting from exhaust fan operation. This study utilized a multi-zone tracer gas injector/sampler designed by Lawrence Berkeley Laboratory which has been used in a number of ventilation research projects in the Northwest.

In summary, the window vents do admit some amount of outside air into homes when they are open and fans operate; however, flows through these vents (as currently sized) are very modest

Cost Comparison of Whole House Ventilation Systems Used to Meet 1994 FMHCSS

Analysis of annual operating costs of the systems which can be used to meet the 1994 FMHCSS reveals a dramatic difference between the two most common whole house ventilation systems currently installed in Pacific Northwest manufactured homes: continuous exhaust-only systems and intermittent furnace-based systems.

The systems are compared on the basis of their ability to ventilate an identical double-section home to an effective ventilation rate of 0.35 ACH. Air at standard density is used in the calculation, along with TMY weather data for the sites mentioned. The analysis is done for electric heat. It assumes no extra energy penalty for duct losses. If such losses were included, which can be quite large and are difficult to calculate, the furnace-based system would be even more costly. The furnace-based system is assumed to use a 375 watt air handler fan to distribute outside air and an exhaust fan rated at 70 CFM (33 L/s) at 0.1" W.G. (and drawing 84 watts) to exhaust stale indoor air. Partial credit is given to the furnace-based system for addition of fan motor heat energy to the incoming fresh-air stream. The exhaust-only system uses a 0.5 sone exhaust fan rated at 70 CFM (33 L/s) at 0.10" W.G. and drawing only 14 watts.

Table 3
Cost Comparison of Most Popular Ventlation Systems in the Pacific Nrothwest
(Seattle TMY weather)

System	Annual energy to run system (kWh)	Annual heating energy for introduced ventilation air	Annual total cost (assuming $0.08/kWh)
Continuous exhaust fan	112	1213	$106
Furnace-based system	2848	835	$295

The furnace-based system must be interlocked with an adequate exhaust fan and run approximately 17 hours a day even during times of the year when heating or cooling may not be required to meet the ASHRAE Standard 62 recommended ventilation level of 0.35 ACH. Parasitic energy costs from using a furnace air handler fan to circulate outside air are significant. Comfort concerns are also significant, depending on where the home is sited. In practical terms, it is unlikelythat many homeowners will use such a system for maintaining indoor air quality.

Conclusions

Summary

Manufactured home ventilation systems can supply effective ventilation on a whole house basis, but only if the system is designed so that occupants will use it. Systems that achieve the intended installed capacity and are "user friendly" enough in terms of noise, drafts, and controls will encourage homeowner acceptance.

Results from the Pacific Northwest indicate past attempts at mechanically ventilating new manufactured homes have not succeeded in meeting ASHRAE Standard 62. Systems based on intermittent operation are sized adequately, but run times are short (generally less than 4 hours per day for each fan), so that combined natural and mechanical ventilation rates are still modest. Newer systems designed to run continuously with minimal fan operation costs may prove more successful in raising effective ventilation rates.

Since environmental effects from indoor air pollutants are not well understood by most home occupants, it is likely that many people will not perceive the need to operate ventilation systems very frequently. If the high failure rate in meeting ASHRAE Standard 62 in newer manufactured homes is viewed as a problem, more work should be done to investigate improvements to ventilation systems in these homes.

For more discussion of the current research and an extended version of this paper, see "Mechanical Ventilation in HUD-Code Manufactured Housing in the Pacific Northwest" to be presented by the authors at the 1997 ASHRAE Winter meeting, Philadelphia, PA, January, 1997

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

17th AIVC Conference, Gothenburg, Sweden, 17-20 September, 1996

EVALUATION OF VENTILATION PERFORMANCE IN PUBLIC SPACES.

John Currie, Graham Capper

Building Performance Centre, Napier University, 42 Craiglockhart Avenue, Edinburgh, Scotland EH14 1LT

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ABSTRACT

Public spaces in the UK have become one of the last bastions of the smoking populace, driven from the workplace by smoking restrictions.

A study has been undertaken to evaluate the performance of ventilation strategies, allied to the use of internal partitioning, intended to reduce the effects of ETS migration in public spaces. Research indicates that very little is known about the likely extent of the problem, where relatively large areas are intended to accommodate both smokers and non-smokers.

The internal environment in a number of public spaces where smoking is still permitted, including public houses and restaurants, were monitored. Levels of the ETS surrogate Carbon Monoxide were measured together with Carbon Dioxide as a general air quality indicator at a number of discrete sample points throughout the occupied zone. The study clearly identified potential problems with a simplistic design approach to shared occupancy spaces both in terms of partitioning and ventilation effectiveness. In some cases retrofit installation of ventilation equipment to mitigate perceived problems resulted in an adverse redistribution of the 'smoke'.

A CFD code was used to model airflows, with the aim of identifying inefficiencies in the ventilation systems, and the results compared with the empirical findings.

Public concern about the exposure to ETS in shared spaces may be justified by the results from this survey and improved ventilation strategies are required to alleviate this situation and maintain a general indoor air quality.

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

17th AIVC Conference, Gothenburg, Sweden, 17-20 September, 1996

EVALUATION OF VENTILATION SYSTEM IN VERY LOW ENERGY HOUSES

J. Pittomvils, H. Hens

Katholieke Universiteit Leuven
Department of Civil Engineering, Laboratory of Building Physics
Celestijnenlaan 131, B-3001 Heverlee (Leuven)
Belgium

F. Van Bael

Batech n.v.
Antwerpsesteenweg 120, B-2390 Malle
Belgium

0. Synopsis

Since 1985 more than 170 very low energy houses, all of the same type and structure, were built in the Flemish Region, Belgium. Because conduction losses are very low, mean U_m-value 0.30-0.35 $W/(m^2{\cdot}K)$, ventilation losses become very important, up to 45% of the heat losses if no heat recovery is utilised. Three of the houses were monitored in detail for energy consumption, energy and ventilation efficiency.
All houses are equipped with the same ventilation system: balanced mechanical ventilation with heat recovery. Tracer gas measurements, pressurisation tests, multipoint temperature measurements and on site and laboratory tests of the heat recovery system, give us a complete scope of the ventilation system and its energy and ventilation efficiency.
Pressurisation and depressurisation tests revealed the main air leaks in the construction: the different connections wall-floor and wall-roof, the window perimeter, even the sockets. Extra care in construction practice changed the n_{50}-value from an average of 4.5 AC/h to 3.5 AC/h, still high for a house with controlled ventilation.
After testing the airtightness, we carried out tracer gas measurements in whole dwellings and between the different zones. Real ventilation rates and interzonal flows were derived. Questions like: Are the airflows in accordance with the design values? Do they match the requirements? How can we measure interzonal flows with one tracer? were answered. The paper gives a mathematical description of the tracer gas flow patterns (solution of the differential flow equations) and compares the results with the measured data.
To complete the evaluation we carried out laboratory tests and field measurements on the heat recovery system. In laboratory the flat plate cross-flow heat exchanger showed a thermal efficiency up to 65% under specific climate conditions. In the dwellings, insulation and airtightness of the ducts appeared to be very important. The temperature efficiency decreased to values less than 45%.
As a conclusion, one may stress that a global evaluation of the ventilation efficiency has to include different tests. The medium or poor airtightness has the greatest impact on the energy efficiency of heat recovery and on the ability to control the system. Detailed testing also showed some flaws in the ventilation system and in the building construction.

1. Introduction

All 70 low energy houses are of the same type and structure. They have high insulation levels, with a mean U_m-value of 0.30 - 0.35 $W/(m^2{\cdot}K)$ (see Table 1).

	U-value	insulation thicknesses
cavity wall	0.19 $W/(m^2{\cdot}K)$	15 cm EPS + 4 cm MW
sloped roof	0.18 $W/(m^2{\cdot}K)$	min. 14.5 cm PUR
flat roof	0.16 $W/(m^2{\cdot}K)$	min. 16 cm PUR
floor	0.5 $W/(m^2{\cdot}K)$	7 cm EPS
glazing	1.8 or 1.3 $W/(m^2{\cdot}K)$	

Table 1: Major U-values and insulation thicknesses of the low energy building [ref. [2]]

A survey of 42 of these dwellings revealed their very low energy use. The average measured total energy use is about 62 kWh/(m^2.a). The energy use for heating varied from 40 to 58 kWh/(m^2.a) with an average of 45 kWh/(m^2.a) in the Belgian climate (1987 degree days, base 15°C) [ref. [3]].

Because conduction losses are very low, ventilation losses become very important, up to 45% of the heat losses if no heat recovery is used. A heat exchanger is included in the balanced ventilation system.
The goal of this research work is to get a clear description of the ventilation and energy efficiency of the ventilation system in this type of low energy house. The strategy we used here is also applicable and useful for other well-insulated dwellings with a controlled ventilation system. First the ventilation and energy efficiency are discussed separately. Then, as a conclusion, they are coupled.

2. Ventilation efficiency

Following items are measured: the airtightness of the building envelope, the airtightness of the ventilation ducts, the supplied and extracted air flows. To derive the actual ventilation rates and the interzonal airflows, tracer gas measurements were performed.

2.1. Airtightness of the building envelope

The airtightness of the building envelope is accurately tested in six houses. The results in Table 2 show the overall airtightness. Differences between the different houses are clear. Firstly the more recent the house the more airtight it is. The values with pressurisation are higher than with depressurisation. Cellar and garage are not airtightly separated from the protected building envelope: houses with cellar and garage in the house are less airtight than the others.

	depressurisation	pressurisation	year of construction	cellar or crawl space	garage in house
1	4.3	6.3	1990	Yes	Yes
2	4.0	5.4	1991	Yes	No
3	4.4	5.3	1991	No	Yes
4	4.9	4.5	1992	Yes, with door	Yes
5	3.6	3.6	1995	No	No
6	3.8	3.8	1995	Yes	Yes

Table 2: Measured n_{50}-values for 6 low energy houses

By visualisation with smoke and by taping of some leaks, the air leaks are qualitatively identified. The major air leaks are a consequence of the building construction itself.
The construction is built up as follows: on the foundation walls and pillars, prefabricated 3.9 × 3.9 m large concrete slabs with EPS insulation are positioned. Then the main load bearing structure, a skeleton of steel profiles and laminated timber joists and trusses, is erected. Finally the lightweight wall, roof and window elements are filled in and the dwelling is finished with a brick outer leaf and a tiled roof surface.

The roof segments are large lightweight prefabricated elements. They just fit into the load bearing structure, but all the seams and junctions remain possible air leaks, especially the connection wall-roof and the connection at the top of the roofs.
The same problem is investigated with the built in lightweight wall elements. Formerly the EPS insulation plates were placed directly on the floor slab, nowadays a space is left to be filled with PUR-foam.
Another interesting air leak is the floor slab itself. Some airopen junctions between the concrete slabs make it possible that air flows from within the crawl space through the floor into the light weight double plated inner walls. Because of this defect in the building construction even the sockets in inner walls are possible air leaks. (In the worst case all interior walls are filled with crawl space air, because all inner walls are interconnected.)
The doors to the garage, the crawl space and cellar are not airtight enough and the window perimeter is an other air leak.
None of these air leaks is dominant but together they make the building envelope too airpermeable for an efficient mechanical ventilation with heat recovery.

2.2. Airtightness of the ventilation ducts

The ducts of the ventilation system consist of a combination of two materials, stiff PVC-tubes and flexible, insulated ducts (2 cm MW). Both the supply and extract ducts are tested by depressurisation with a small power controlled fan. Table 3 and Figure 1 show the results for a pressure difference of 50 Pa. These values are extremely and intolerably high. When the system is working on medium speed at 50 Pa, 33% of the air escapes through air leaks in the ductwork.

	airflow [m^3/h] at ~~50 Pa~~	
air leaks		
Supply circuit	57	50 Pa
Extract circuit	55	
ventilation fans		
low	79	
medium	151	?
high	265	
enhanced	320	

Table 3: Leakage airflows compared with the airflows through the ventilation system (different speed levels) at a pressure drop of 50 Pa

2.3. Measurement of the supplied and extracted air flows

Due to the great leakage flows in the ventilation system one can conclude in advance that the ventilation system will not supply the amounts of air it is dimensioned for. Nevertheless we did the measurement in 2 dwellings (dwelling 4 en 6). In both houses the airflows are measured in two conditions:

- all interior doors closed and the ventilation system on low level (worst scenario)
- all interior doors opened and the ventilation system on high level (most favourable)

The airflows were investigated with a rotor anemometer at the ventilation openings. In the worst scenario, almost none of the air inlet or extract openings delivers or extracts enough air according to the Belgian regulations. Even in the most favourable case some rooms are not provided well in one house.

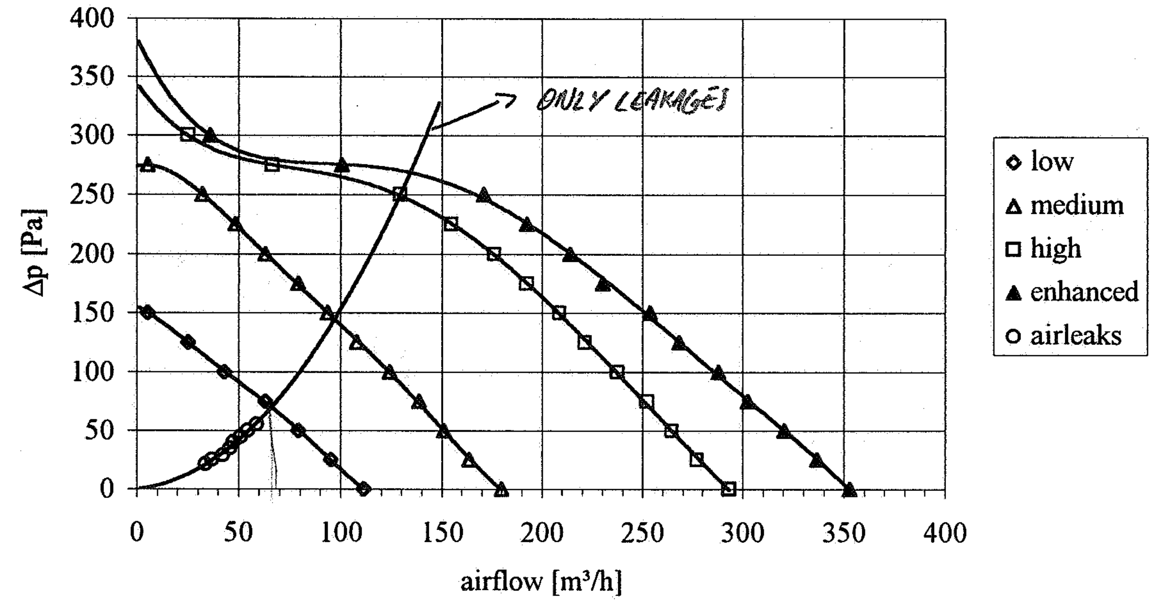

Figure 1: fan characteristics of the ventilation system compared with the duct leakage flows

2.4. Ventilation rates for each room

Instead of calculating mass balances between rooms, starting from the extracted and supplied airflows, one can measure real ventilation rates with tracer gas measurements. So for some of the rooms in the different houses the ventilation rates are measured using "the decay method". Ventilation rates from 0.35 AC/h (sleeping room house 1) up to 0.75 AC/h (bathroom house 2) are measured. In most of the cases there is no constant decay. This is obvious because of the lack of airtightness of the building envelope. The climate still plays an important role in the ventilation of the dwellings. But thanks to the lack of airtightness (high infiltration rates), most of the rooms receive a higher and sufficient ventilation rate.

2.5. Interzonal airflows

The ventilation rates calculated from the single zone measurements have been compared with the regulations. But one also has to examine if the airflows go from inlet to outlet and with the right amounts of fresh air. Therefor we tried to measure and calculate the interzonal airflows. The first and very important hypothesis, is the assumption that the ventilation rate is constant. Using this assumption, one can easily derive, from the mass balances, the differential equations describing the airflows between two different rooms and outside (in appendix).

Combining these equations and the tracer gas measurements, it is possible to calculate the interzonal airflows, so that the best fit is achieved between measured and calculated tracer gas concentrations.

Results for dwelling 1: The airflows were found by iteration of a two-zone model with the tracer gas measurements. Figure 3 shows the similarity of the measured and theoretical (best fitted) tracer gas concentrations.

from	→ to	airflow [m³/h]
living room	→ outside	95
living room	→ adjacent	17
outside	→ living room	109
adjacent	→ living room	3
outside	→ adjacent	62
adjacent	→ outside	76

Table 4: calculated interzonal airflows

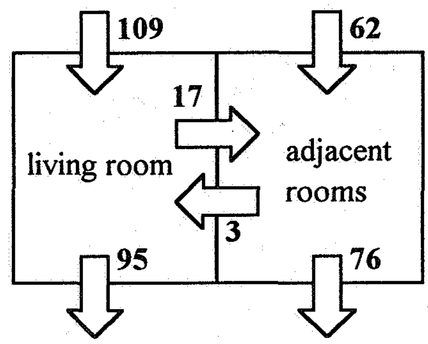

Figure 2: calculated interzonal airflows

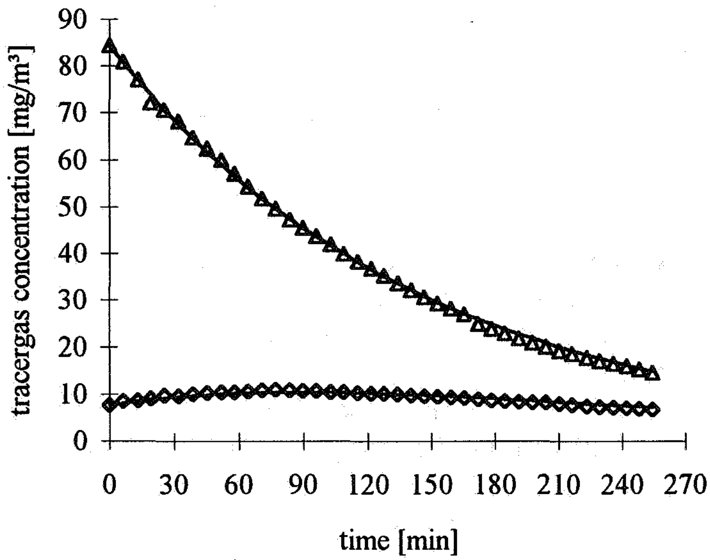

Figure 3: measured and theoretical (best fitted) tracer gas concentrations.

There are some remarks on this method. Some measuring cycles give insufficient results because of different reasons:

- the ventilation rate is not constant
- the measuring period is too short
- the tracer gas concentrations in the adjacent rooms are very low
- the tracer gas concentrations are supposed to be homogeneous in each room
- the tracer gas concentrations are not measured in the extract and supply ducts

Nevertheless the method gives information about the airflow patterns within the house. Some flaws in the ventilation system have been detected. For example: the bathroom in house 4 has a return flow through the hall to the toilet.

3. Energy efficiency of the heat recovery system

All the houses are equipped with a mechanical ventilation system with a static air-to-air heat exchanger. We will not proceed here with an economic analysis of this choice, we only look to the energy efficiency. First some results of the laboratory tests of the heat exchanger are given. Further the energy efficiency of the heat exchanger in the low energy houses is considered.

3.1. Laboratory tests of the air-to-air heat exchanger

The heat exchanger used is a flat plate air-to-air heat exchanger with cross-flow.
During several weeks the heat exchanger was tested under very different laboratory conditions in over 60 test-cycles (temperature, temperature difference, relative humidity, mass flow rates, pressure drops). The overall conclusion is that the tested heat exchanger retains a very constant temperature efficiency of about 65 to 67 % under different conditions of temperature difference and air flow rates.

3.2. On site testing of the air-to-air heat exchanger and the heat recovery system

In house 1 the efficiency of the heat exchanger was tested on site. The heat exchanger itself revealed a very high efficiency of about 73 %. However, the main problem is that, due to the lack of insulation and airleakage of the ducts, the temperature difference between outside and inside, is not active over the heat exchanger. As a consequence, temperature efficiency in situ decreases to 45 %.

4. Global evaluation of the mechanical ventilation system in the very low energy houses

The most important aspect for an efficient ventilation system for this type of houses is AIRTIGHTNESS of the envelope and the ducts, as well from the energetic point of view, as for ventilation efficiency. Lack of airtightness of the building envelope disturbs the ventilation balance and makes the airflows in the different rooms weather dependent. It also withdraws the energy efficiency of the heat recovery system. All air which infiltrates or exfiltrates through the envelope is lost for heat recovery. Lack of airtightness of the ductwork short-circuits the balanced ventilation system and, due to this, the efficiency of the heat recovery in the investigated dwellings decreases to less than 45 %.
Despite or sometimes thanks to the lack of airtightness most of the rooms are ventilated sufficiently.

5. Appendix

5.1. Calculation of interzonal flows with one tracer gas, using the decay method [ref. [1], [5]]

- conservation of mass: hypothesis: the air density is constant

$$\sum_j \left(-q_{ij} + q_{ji}\right) = 0 \qquad (1)$$

for $i = 1, n$ with n = number of zones
$j = 1, n$ + outside, $j \neq i$
q_{ij} and q_{ji} = airflows from i to j and from j to i, [m^3/h]

- conservation of mass of the tracer gas

$$V_i \frac{dc_i}{dt} = \sum_j \left(q_{ji} \cdot c_j - q_{ij} \cdot c_i\right) \qquad (2)$$

for $i = 1, n$ with n = number of zones
$j = 1, n$ + outside, $j \neq i$
c_i = tracer gas concentration in zone i, [mg/m^3]

The equations (2) form a system of n coupled linear first order differential equations with n independent variables c_i and with the airflows q_{ij} and q_{ji} and the volumes V_i as parameters. The solution of this system is a sum of n exponential functions of the form:

$$c_i(t) = A_{i1}e^{\lambda_{i1} \cdot t} + A_{i2}e^{\lambda_{i2} \cdot t} + \ \dots \ + A_{in}e^{\lambda_{in} \cdot t} \qquad (3)$$

with A_{ij} and λ_{ij} functions of the airflows q_{ij} and q_{ji}

The differential equations are solved numerically with the method of central differences. With the tracer gas concentrations at time t, the tracer gas concentrations at time $t+\Delta t$ are calculated, for a specific set of the parameters q_{ij} and q_{ji} and V_i.

$$c_{i,t+\Delta t} = c_{i,t} - \sum_j \left(q_{ji} \cdot c_{j,t} - q_{ij} \cdot c_{i,t}\right) \cdot \frac{\Delta t}{V_i} \qquad (4)$$

By solving the minimisation problem between calculated and measured tracer gas concentrations the unknown airflows q_{ij} and q_{ji} are derived iteratively.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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A TECHNIQUE TO IMPROVE THE PERFORMANCE OF DISPLACEMENT VENTILATION DURING COLD CLIMATE CONDITIONS

Peter Hansson*, **Hans Stymne**

The Royal Institute of Technology
Dept of Built Environment, Box 88, S 801 02 Gävle, Sweden

* Affiliated to: University College of Gavle-Sandviken,
S 80176 Gävle, Sweden

A technique to improve the performance of displacement ventilation during cold climate conditions

Peter Hansson* and Hans Stymne
The Royal Institute of Technology
Department of Built Environment
Box 88, S-801 02 Gävle, Sweden

*Affiliated to:
University College of Gävle-Sandviken
S-80176 Gävle, Sweden

Synopsis

Ventilation by displacement is a type of ventilation where the air flow is thermally driven. By this arrangement one obtains two zones in the room - a lower zone with supply air conditions and an upper recirculation zone with extract air conditions. Cold climate causes downdraught from windows and external walls and results in a mixing of air from the upper into the lower zone. To avoid this problem during cold climate a new principle for ventilation by displacement is tested. Excess heat from the upper zone of the room is used for heating cold surfaces. The principle involves creation of a narrow space in front of the exterior wall, separating the cold wall from direct contact with room air. Extract air from the ceiling level is forced down through this space by an extraction fan. This method can advantageously be applied in large buildings where an external wall mainly consists of glass panes. In this case it might be appropriate to utilise the space between the inner couple of glasses for extract air. Tracer gas and temperature measurements were carried out in a test-room. The result shows that the ventilation efficiency improves when using the new principle. The thermal climate also improves due to less down-draught and higher surface temperatures.

1 Introduction

The main purpose for a ventilation system is to provide good indoor air quality for people. Ventilation by displacement is one of the most effective principles considering the ability to remove contaminants emitted from humans and other heat sources. The ideal function of displacement ventilation in a room is showed in figure 1. The air flow is thermally driven by buoyancy forces (induced by heat sources like occupants and machines). Two different air zones develop in the room -a lower zone with supply air conditions and an upper recirculation zone with extract air conditions, where extract air is warm and contaminated by compounds originating from humans, appliances, building materials etc.

The interface between the upper and lower zone is stabilised due to a temperature stratification. It is important to maintain this stable interface, otherwise the lower zone gets contaminated and the advantage of ventilation by displacement is lost. Cold climate causes down-draught from windows and external walls and results in a mixing of air from the upper into the lower zone. The air in the lower zone will thus be contaminated and the ventilation efficiency will decrease (figure 2).

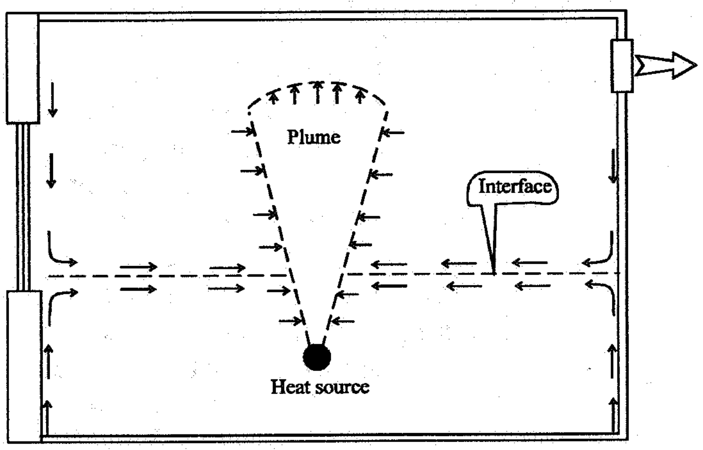

Figure 1 Ideal displacement ventilation.

To avoid this problem during cold climate a new principle for ventilation by displacement is tested. Excess heat from the upper zone of the room is used for heating cold surfaces. Normally extract air leaves the room by the extract air device at ceiling level. The new principle involves creation of a narrow space in front of the outer wall, separating the cold wall from direct contact with room air. Extract air from the ceiling level is forced down through this space by an extraction fan.

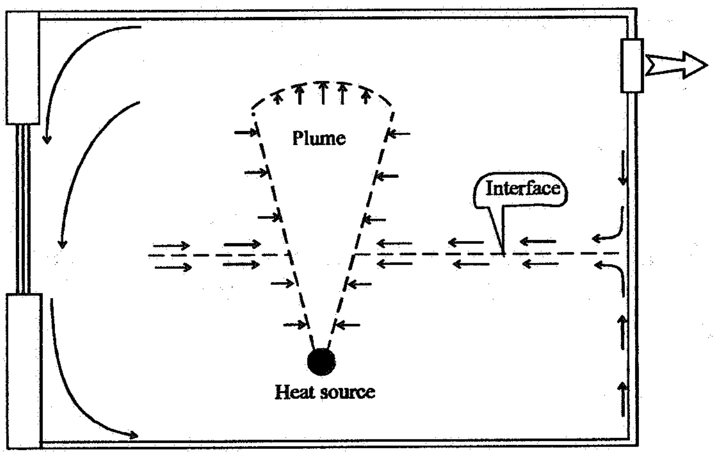

Figure 2 Ordinary displacement ventilation under cold climate conditions.

2 Experimental set-up

A full-scale test room with the dimensions $4.2 \times 3.6 \times 2.8$ m ($L \times W \times H$) is used (figure 3). The outer wall is a common well insulated (U-value = 0.27 W/m^{2}K) wall in the Nordic countries equipped with two triple glass windows (U-value = 2.0 W/m^{2}K). A narrow space was built in front of the external wall of the room, the space was constructed by use of a polyethylene film, spanned over wood strips. At ceiling level of the construction there is an opening so that air from here can be forced down through the space in front of the wall by an extraction fan connected to a perforated metal tube mounted at the bottom of the space.

The wall is connected to a cooling chamber ran at a temperature of -20 °C and 0 °C . At the middle of the test room a dummy was positioned. The dummy is made of metal tubes and has an internal heat production of 100 W, simulating a human. The supply air temperature was held at 17 °C and the air flow rate was 16 l/s. Air was distributed to the room via a low velocity device at floor level. Tracer gas and temperature measurements were carried out in order to investigate the performance of the new principle. Temperature gradients were measured at five different locations. Tracer gas was released at constant mass flow rate through a tube fixed to the dummy at the height of a mouth of a human. Thus, it should simulate a contaminant originating from a human (e.g. a smoker). The tracer gas concentration was measured at four different heights and locations in the room. The ventilation efficiency was calculated from tracer gas steady state values (steady state occurs after approx. $4 \times \tau_n$, where $\tau_n = \dfrac{\text{Room\_volume}}{\text{Air\_flow}}$ [h]). The ventilation efficiency describes how well a ventilation system can reduce pollutants in the indoor environment. Calculations of the height of the interface separating the two zones was made in order to verify tracer gas concentration gradients. Furthermore indoor climate parameters such as the effective under-temperature and the air velocity of downdraught were calculated.

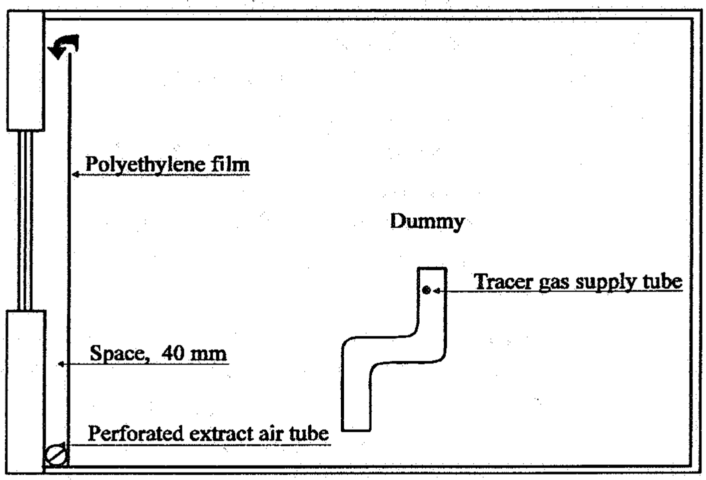

Figure 3 The test room.

Note: When no separating space is used extract air leaves the room via an ordinary extract air device at ceiling level.

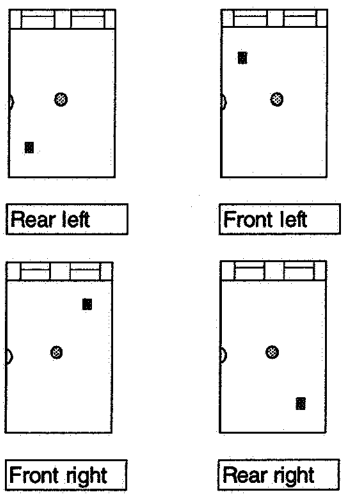

Figure 4 Top view of the test room. The squares represent tracer gas and temperature gradient measuring points in the occupied zone. The circle represents the heated dummy.

3 Results and discussion

3.1 Ventilation effectiveness

The ventilation efficiency can be defined as:

$$\bar{\varepsilon} = \left(\frac{\dot{m}/q}{\bar{C}} \right) \times 100\% \qquad (1)$$

Where $\dot{m}$ = mass flow rate of tracer gas into the room, q = supply air flow rate and $\bar{C}$ = average concentration of tracer gas in the room measured at 0.5, 1.0, 1.5 and 2.0 m height.

The ventilation efficiency with the uppermost measuring point in the room as a reference can be defined as:

$$\bar{\varepsilon}_{2.0} = \left(\frac{C_{2.0}}{\bar{C}} \right) \times 100\% \qquad (2)$$

Where $C_{2.0}$ = concentration of tracer gas at 2.0 m height.

Table 1 Ventilation efficiency

		Ventilation efficiency $\bar{\varepsilon}$ [%]			
Cooling chamber temperature		**0 °C**		**-20 °C**	
Case		Space	No space	Space	No space
Measurement point	*Front left*	132	124	130	134
	Front right	149	129	138	114
	Rear left	218	163	-	-
	Rear right	209	152	192	-
Mean value for the room		177	142	153	-

Table 2 Ventilation efficiency with the uppermost measuring point in the room as a reference

		Ventilation efficiency $\bar{\varepsilon}_{2.0}$ [%]			
Cooling chamber temperature		**0 °C**		**-20 °C**	
Case		Space	No space	Space	No space
Measurement point	*Front left*	121	122	120	130
	Front right	130	119	118	115
	Rear left	183	160	-	-
	Rear right	164	143	153	-
Mean value for the room		150	136	130	-

The result shows that ventilation efficiency improves when using the new principle. The ventilation efficiency is significantly higher in the rear region of the room. The heat production in the room approximately covers heat losses at 0 °C cooling chamber temperature.

3.2 Tracer gas concentration gradients

Figure 5 shows that the concentration gradient is more ideal when using the space. It is evident that contaminated air has moved from the upper zone of the room to floor level due to down draught from the external wall without space. A person sitting at that place would inhale contaminated air because air from floor level rises up along the human body to the nose. Furthermore, the interface has raised and is located at a higher level compared to the case with a space.

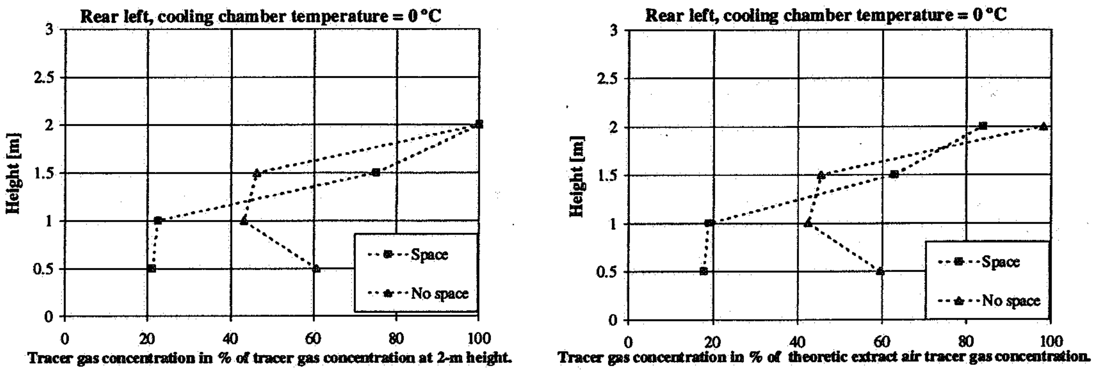

Figure 5 Tracer gas concentration gradients measured at rear left in the room.

3.3 Temperature measurements

Figure 6 shows one example of temperature gradients in the room where the measuring points for air temperature are at rear left. The result is evident; The surface temperature of the polyethylene film is higher in all measurement points when using the new principle. Note that even if downdraught is decreased, the ideal temperature gradient is not obtained with the new principle. The ideal case would be higher polyethylene film temperature than room air temperature below the interface (height level approx. 0.9 - 1.0 m, section 3.5). Thus above the interface, surface temperatures should be lower than room air temperature. The tested space had the same width from bottom to top. In order to provide more heat exchange at the lower part of the film one could make the space more narrow there.

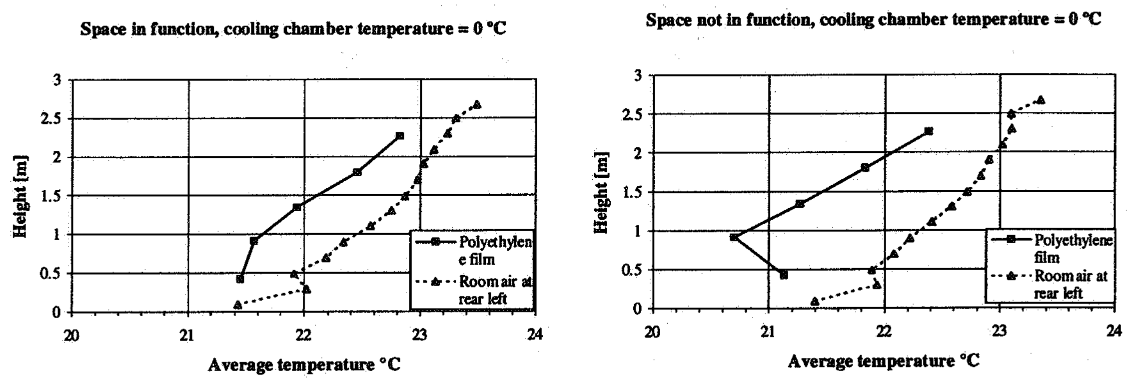

Figure 6 Temperature gradients (polyethylene film respectively room air).

3.4 Thermal climate

Cold climate causes down draught from windows and external walls. In order to calculate the maximum velocity of the air stream one can use:

$$u_{max} = 0.10\sqrt{x(t_r - t_v)} \quad [m/s] \tag{3}$$

(Rydberg 1963 based upon the work of Eckert and Thomas 1951). Where x = distance from top, t_r = room temperature and t_v = average surface temperature. A measure of influence of temperature and velocity on the human body can be expressed:

$$\vartheta = 0.4(t_r - t_v) + 0.8\sqrt{x(t_r - t_v)} \quad [K] \tag{4}$$

(Rydberg 1963). Where delta is called "the effective under-temperature". The effective under-temperature tells how large temperature difference still air needs, to cause the same cooling effect on the human body as air in movement. The downdraught flow is according to (Rydberg 1963) and (Mundt 1994):

$$\upsilon = 10(t_r - t_v)^{0.4} x^{1.2} \quad [m^3/h \text{ and m wall width}] \tag{5}$$

Table 3 Thermal climate parameters (measurement point, front right).

Cooling chamber temperature	**0 °C**		**-20 °C**	
Case	**Space**	**No space**	**Space**	**No space**
***u_{max}* [m/s]**	0.07	0.21	0.14	0.31
***Effective under- temperature* [K]**	0.65	2.26	1.38	3.79
***Downdraught flow* [l/s]**	16.83	40.05	28.22	54.97

The lowest maximum velocity appears when the space is used. The effective under-temperature should be in the range of 0.5 - 1.0 K for a sitting office worker (Rydberg). Only the case with space and 0 °C cooling temperature fulfils the demand. The downdraught flow is approximately doubled for the case without space compared to the case with space, this verifies the appearance of the tracer gas concentration gradients in figure 5.

3.5 The interface

The height of the interface separating the contaminated and clean zones in the room depends on: a) The supply air flow, b) the plume flow above heat sources and c) the convection air flows along vertical surfaces. The continuity equation (Sandberg 1990) is

$$Q_v = Q_p(z) \pm Q_b \tag{6}$$

where Q_v is the supply air flow rate, $Q_p(z)$ is the plume flow rate (up) from a point source (increases with the height z). Q_b is downdraught (-) and convection air flow rates upwards (+). The plume flow above the dummy can be expressed as:

$$Qp(z) = 55(Pk)^{1/3}(Z + Zv)^{5/3} \quad [l/s] \tag{7}$$

(Mundt 1994). Where Pk is the convective heat transfer of the dummy. [kW], Z is the height above the heat source [m] and Zv is the virtual height of the heat source [m]. Equation 5 is used for Q_b. The interface height (z) is found where the supply air flow equals the sum of plume flow and convection flows. Result of a calculation for 0°C cooling chamber temperature shows that the interface height is **1.0** m without space and **0.9** m with space. This follows the results in section 3.2 and 3.4. Without space, contaminated air flows down and penetrates the interface near the exterior wall into the cleaner lower zone. Thus the interface height increases.

3.6 Application

The new method could advantageously be applied in large buildings where an external wall mainly consists of glass panes, which often is the case in waiting rooms, air terminals etc. It might be appropriate to utilise the space between the inner couple of glasses for extract air , figure 6. The extract air device should then be equipped with a filter or/and the inner couple of glasses should be able to clean.

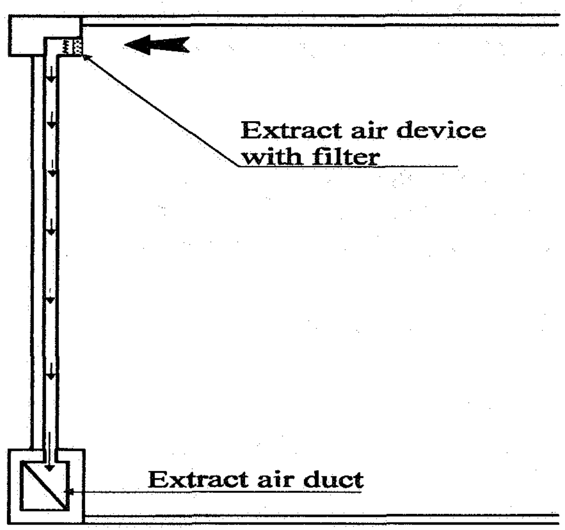

Figure 6 Application of the new principle

4 Conclusions

A suggested technique to utilise warm air at the ceiling level to heat cold surfaces has been demonstrated to improve the ventilation efficiency and the thermal climate in a test room equipped with displacement ventilation during cold climate. There is good agreement between theory and practice , the interface between the two zones in the room is displaced and downdraught is decreased when the new principle is used. The tracer gas concentration gradients verifies this fact.

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EVALUATION OF A CONTROLLED NATURAL VENTILATION SYSTEM

17th AIVC Conference, Göteborg, Sweden
17-20 September, 1996

Thomas Carlsson

Swedish National Testing and Research Institute
Box 857
S-501 15 BORÅS
SWEDEN

1. INTRODUCTION

The most commonly used form of ventilation of residential buildings in Sweden is passive stack ventilation. Its driving power is the temperature difference between internal and external temperatures and the wind. Other factors that affect ventilation rate are the airtightness of the building, the sizes and numbers of the outdoor air inlets and the sizes of the ventilation ducts.

Common shortcomings or problems in buildings with natural ventilation are:

* Varying ventilation flow rates over the year;
* Major variations in air flow rate from room to room;
* A significant effect of wind on the ventilation flow rate;
* Air leaks in the building envelope can cause uneven distribution of indoor ventilation.

When renovating older apartment buildings with passive stack ventilation, the choice of ventilation method is generally a compromise between various requirements and what is practicable at a reasonable cost. The costs of converting older passive stack systems can be considerable, as it is often decided to convert the system to a technically more complicated one with mechanical exhaust.

There is considerable interest today in finding simple system arrangements for use when renovating older passive stack systems. One such arrangement that has been discussed in Sweden is the fan-assisted passive stack ventilation system. This is based on the use of an auxiliary fan that maintains ventilation when the natural drive forces are insufficient. In order to minimise the effects of temperature on ventilation flow rate, systems are often fitted with temperature-controlled outdoor air inlets.

The project described in this paper has performed simulations using a multi-zone air flow model [4 (COMIS)] of three different passive stack ventilation systems. The objective of the simulation calculations was to evaluate system performances and to make suggestions for possible improvements of the systems.

2. THE VENTILATION SYSTEMS TESTED

Three different systems have been investigated, based on three renovation projects of passive stack ventilation systems in apartment buildings in Stockholm. The systems were inspected and the results recorded. A questionnaire survey [5] was performed in the three buildings, although the results are not reproduced here.

The first system (1) is a conventional passive stack ventilation system for residential buildings, with exhaust air ducts and controllable outdoor air inlets under windows.

The second system (2) was a fan-assisted passive stack ventilation system, based on the use of the existing passive stack ventilation ducts without extensive renovation. Reducing ventilation as a resulting of stack decreasing effect is steplessly and automatically compensated by an axial-flow fan at the top of the duct, speed-controlled by the ambient temperature (see Figure 2). The fan enclosure is designed to reduce the effect of the wind and can also be modified for heat recovery from the exhaust air. Fresh outdoor air is admitted through temperature-controlled and self-adjusting inlets in sitting rooms, bedrooms and larders. As the ambient temperature falls, the amount of outdoor air admitted is reduced. Ventilation air is exhausted from kitchens and bathrooms as normal.

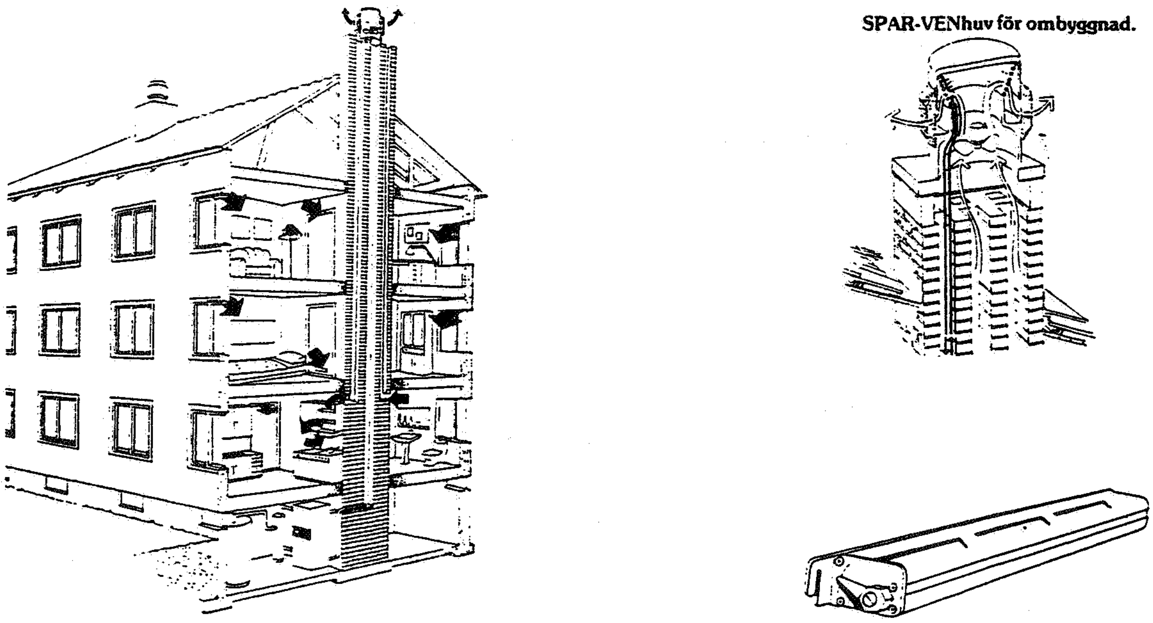

Figure 1. Components of the fan-assisted passive stack ventilation system.

Fan Speed vs Outdoor Temperature

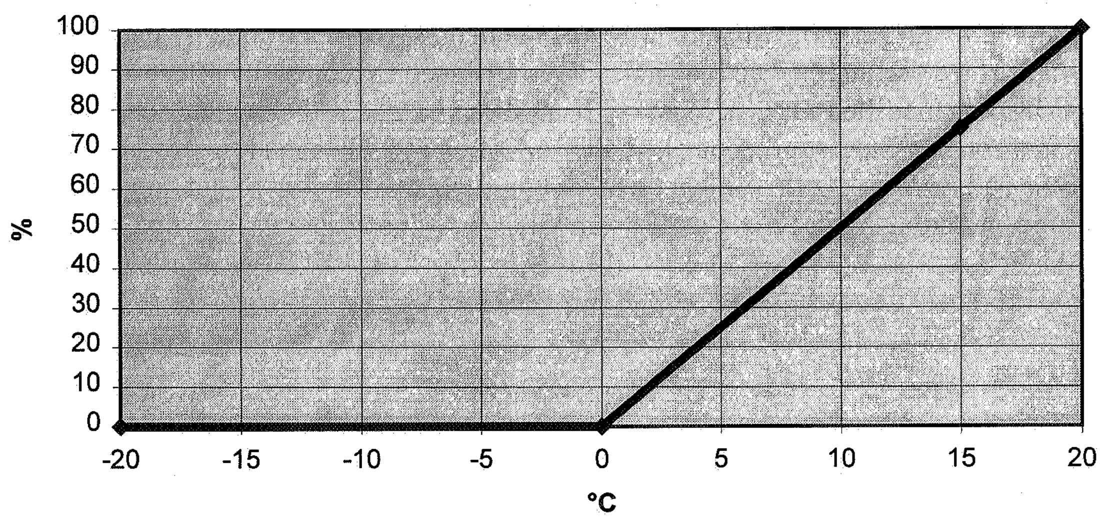

Figure 2. Fan control characteristic for systems (2) and (3).

The third system (3) that was investigated had outdoor air inlets as in a conventional Swedish system (1) and a fan and control system as in (2).

The apartment is in a residential area in a suburb of Stockholm, surrounded by buildings of the same height. The building is one of 13 in the area, containing a total of 189 apartments, with the distance to the nearest building being about 25 m. The buildings, which have three storeys and a cellar, date from the beginning of the 1940s. Indoor ceiling height is 2.7 m.

The simulations were performed for an apartment having the following characteristics.

Table 1. Differences between main features of the three systems

	Conventional passive stack system without fan (type 1)	**Fan-assisted passive stack ventilation system (type 2)**	**Conventional passive stack system with fan (type 3)**
Outdoor air inlets in rooms	Beneath windows: 1 in sitting room 1 in bedroom	Above windows: 2 temp. controlled in sitting room 1 temp. controlled in bedroom	Beneath windows: 1 in sitting room 1 in bedroom
Outdoor air inlet in larder	Disc-type inlet	1 temp. controlled	Disc-type inlet
Extractions	Kitchen and bathroom	Kitchen and bathroom	Kitchen and bathroom
Exhaust duct, pressure coefficient	-0.3	0	-0.3

3. WORKING METHODS

The objective of the investigation was to evaluate and compare the performance of a commercially available fan-assisted passive stack ventilation system with that of a passive stack system of the type that is conventional in Swedish residential buildings. The simulations compared the following parameters for both systems:

- Flow variations over the year
- Differences in ventilation performance in individual rooms
- The effect of wind on ventilation performance in individual rooms
- The effect of height above ground

A representative apartment, as in one of the buildings, was used as the model for all the simulations, being adapted in the model to incorporate the different types of ventilation systems as shown in Table 1. For the purposes of comparisons between the different systems, the apartment was assumed to be on the ground floor (level 1), but for the effects of height it was assumed to be on the second floor (level 3).

4. RESULTS

General input data for the simulations

Apartment's airtightness	1.9 air changes/h at 50 Pa negative pressure
Building pressure coefficient	As described by [1]
Internal duct dimensions	120 x 250 mm
Friction coefficient/for ducts	0.0042 [2]
Air leakage through closed larder door	As described by [3]
Inner doors:	Bathroom door closed, others open
Weather data	Stockholm, 1971

Flow variation over a year

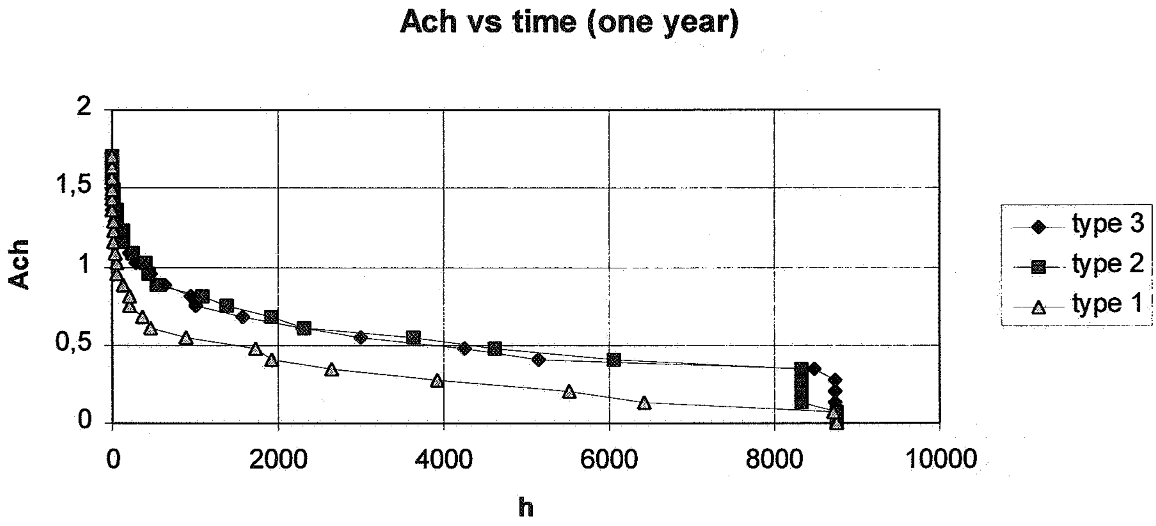

Figure 3. Air changes per hour over a year with the three different systems

Table 2. Air change rates per hour for the different systems

System type	Mean annual hourly change rate	Proportion of the year during which air change rate is within 20 % of annual mean value
Fan-assisted passive stack (type 2)	0.56	32
Passive stack with fan (type 3)	0.54	38
Passive stack without fan (type 1)	0.29	26

Despite the fact that System 2 (see Figure 3) incorporates temperature-controlled fresh air inlet fittings, the flow rate varies during the year just as much as it does in System 3, which does not have temperature-controlled inlets. The temperature-controlled inlets, in other words, have not succeeded in stabilising the air flow rate.

It can be seen from Table 2 that the proportion of the year during which the air flow rate is within ±20 % of the mean value is somewhat higher for System 3 than for System 2. This, too, shows that the temperature-controlled fresh air inlets have not had any effect in terms of stabilising the flow rate over the year.

The temperature-controlled fan, on the other hand, has increased the flow from 0.29 air changes/h to 0.54-0.56 air changes/h.

Differences in ability to ventilate individual rooms

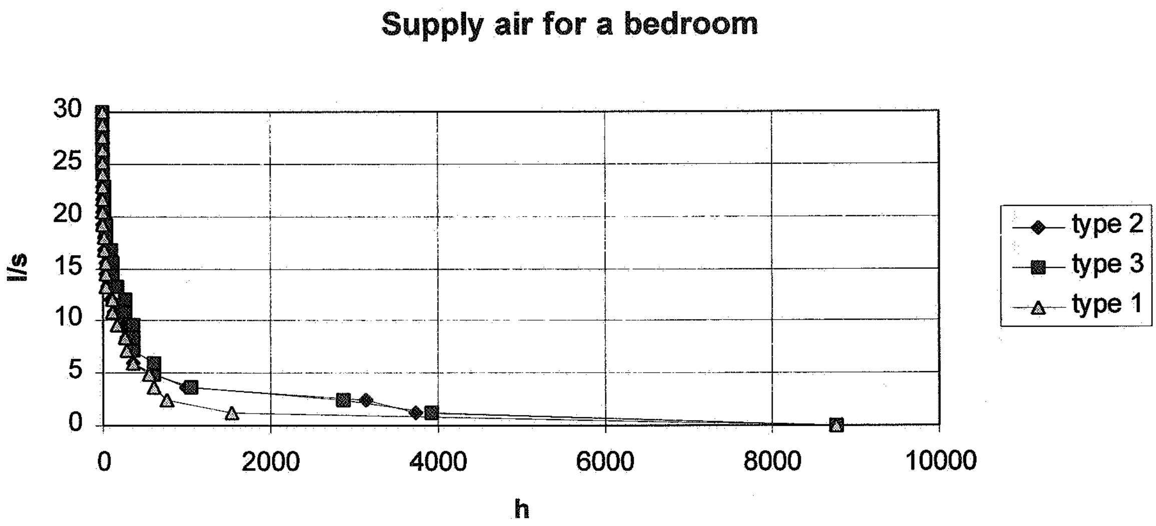

Figure 4. Supply air flow rate to the bedroom in the three different systems

Table 3. Average supply air flow rate over a year to bedroom for the three different systems

System type	Fresh air flow rate, l/s (mean value over the year)
Fan-assisted passive stack (type 2)	1.8
Passive stack with fan (type 3)	2.2
Passive stack without fan (type 1)	1.3

Swedish building regulations require the fresh air flow rate to a bedroom to be at least 4 l/s per person. It can be seen from Table 3 that none of the systems investigated meets this requirement. Systems 2 and 3 fulfil it for a lesser part of the year (< 1000 hours), as shown in Figure 4. For most of the year (> 4600 hours), there is no greater difference between the three different types of systems.

The effect of wind on ventilation of individual rooms

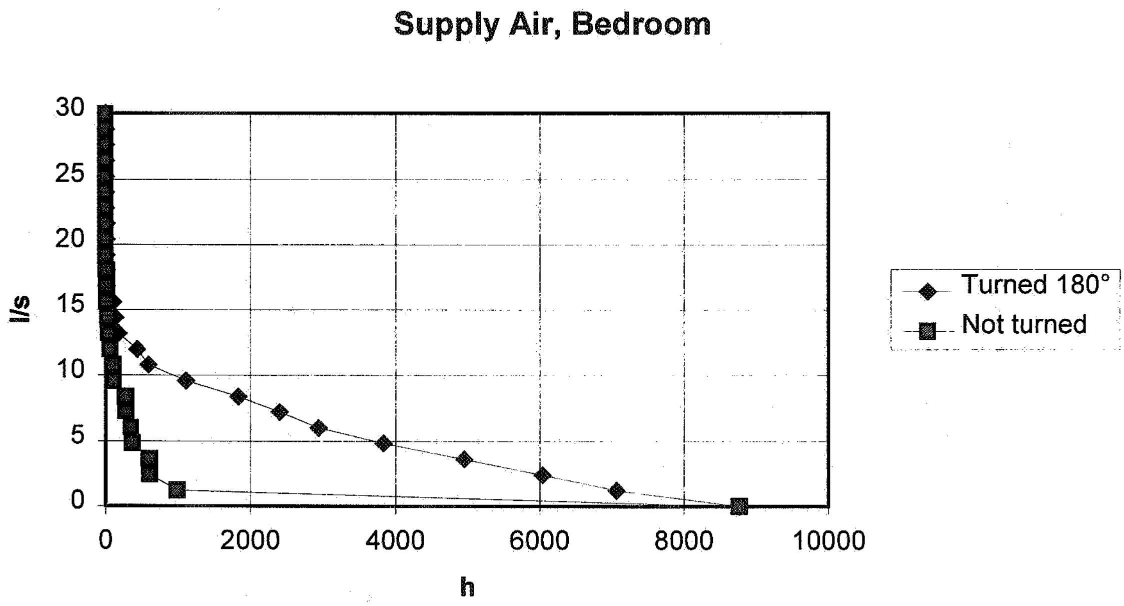

Figure 5. Outdoor air flow rate to the same bedroom with two different wind directions. Comparison performed for system type 1.

Table 4. The effect of wind direction on supply air flow rate to bedroom, system type 1

System type	Fresh air flow rate, l/h (mean value over the year)
Passive stack system (type 1)	1
Passive stack system with building rotated through 180°	4

It can be seen from Figure 5 and Table 4 that the wind direction has a considerable effect on the ventilation of individual rooms. Turning the building through 180° increases the fresh air flow rate from 1 l/s to 4 l/s, expressed as mean values over the year. Figure 5 shows that the fresh air flow rate through the bedroom is less than 1 l/s for over 6000 hours of the year. Any fresh air inlet device or fitting that is intended to compensate for the natural drive forces in the system must be able to compensate both for temperature and for wind.

The effect of height above ground

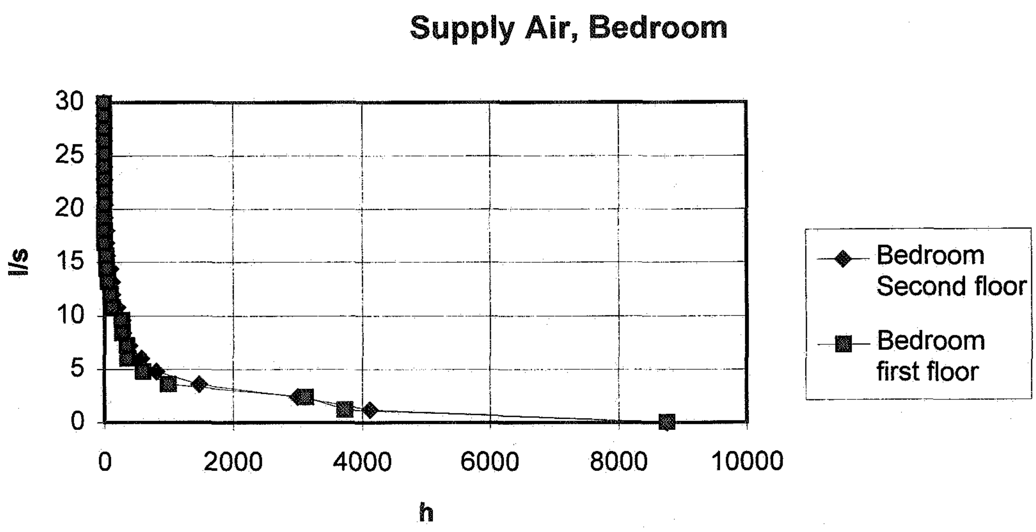

Figure 6. Supply air flow rate to a bedroom on the ground floor (level 1) and on the second floor (level 3), system type 2.

Table 5. Mean annual flow rate to a bedroom on the ground floor (level 1) and on the second floor (level 3), system type 2.

	Mean annual flow rate, l/s
Ground floor bedroom	2
Second floor bedroom	2

That there is virtually no difference between the outdoor air flow rates on different floors is explained by the pressure drop in the ventilation duct. At the ground floor, there is greater stack effect than on the second floor. However, as the ventilation duct is also longer, there is also a greater pressure drop, which more or less cancels out the effect of the greater stack effect so that the flow is approximately the same on both floors.

5. CONCLUSIONS AND RECOMMENDATIONS

System analyses with computer simulations have shown themselves to be a valuable aid in assessing total ventilation performance and how the systems operate. The quality of the input data is the prime determinant in deciding how closely the simulated results agree with real conditions. For more accurate investigations, it will be necessary to complement the computer simulations with actual measurements of the indoor climate conditions, airtightness of the building envelope, performance of the ventilation system and energy use.

The technology of fan assistance of passive stack ventilation systems needs to be further developed. As expected, the systems in this investigation with ancillary fans handled higher ventilation flow rates than unassisted passive stack systems, but are hardly more stable than them, as shown, for example, in Figure 3.

The temperature-controlled outdoor air inlets do not affect the outdoor air flow rate to make it more constant over the year in the apartment investigated here (compare curves 1 and 2 in Figure 3).

The effect of wind on the flow rate to individual rooms is considerable, as shown in Figure 5. Although the total fresh air flow rate to the apartment is acceptable, there is a problem in ensuring that the air enters each individual room.

The fact that the difference in air flow rate between the floors is only slight is explained by the pressure drop in the passive stack ventilation duct. The more powerful stack effect for the ground floor apartment is offset by the greater pressure drop in the duct due to the longer length of duct. This means that there is no greater difference in outdoor air flow rate between apartments on the ground floor and on the second floor.

If the systems are to be improved, it will be necessary to improve the performance of the components. If designs are to be improved, it will also be necessary to improve the technical specifications of the components. Finally, adjustments of the air flow rates in the systems must be improved in order to make the ventilation flow rate more uniform throughout the year.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

17th AIVC Conference, Gothenburg, Sweden, 17-20 September, 1996

The Development of an Occupancy-Controlled Exhaust Air Ventilation System

Per Levin

The Royal Institute of Technology, KTH
Department of Building Sciences
Brinellv. 34, S-100 44 Stockholm, Sweden

Synopsis

Many dwellings with natural or gravity ventilation systems suffer from poor airchange rates. In Sweden, especially houses built in the 1960-ies and 1970-ies heated with electric resistance heating and thus without chimneys, are at risk. Improving the airchange rate in these houses is to some extent performed to decrease Radon gas concentrations where appropriate. For comfort, most homeowners learn to live with low airchange rates, accepting e.g. odours or window condensation and trying to compensate this with increased airing. They are often reluctant to install mechanical ventilation systems.

This paper describes a new concept for an occupancy-controlled exhaust air ventilation system. The system features bedroom night-time ventilation, wet-room day-time ventilation and variable total air flow at a low installation cost.

The system has since autumn 1994 been successfully installed in four houses and evaluated by air flow measurements, tracer gas tests, multizone airchange calculations etc. The homeowners are very pleased with the performance of the systems and the improved airchange rates.

1. Introduction

During the 1950-ies and 60-ies electric resistance heating systems were introduced in a large scale in Sweden which meant that houses could be built without chimneys for combustion gasses. Separate ventilation ducts for gravity-driven air flows was drawn from "wet rooms". A separate cooker-hood with fan was mounted above the stove.

At the end of the 70-ies, more strict requirements on thermal insulation and building airtightness of building envelopes were introduced. In spite of that sealing measures were performed both in new production and in older houses, no requirements to ensure an adequate airchange rate was given. Natural gravity ventilation systems was for a period allowed for single-family houses even though the required air flow, 0,35 l/sm^2, not could be guaranteed for all seasons. In practice, almost all new permanently occupied houses in Sweden since the end of the 70-ies have mechanical ventilation systems installed.

More effective weather-stripping and other sealing measures in houses with natural gravity ventilation restricts the supply air flow but the large area of the ducts still remains. The relatively small pressure differences that drives the air flow becomes insufficient and the system becomes unstable. Starting the cooker-hood fan will then cause the air flow to change direction in one or more ducts. If these ducts are cooled down there is a risk that a more stable condition takes place in that e.g. the duct in the bathroom becomes supply instead of exhaust at the same time as the total air flow is greatly increased.

Unevenly distributed air leakage can cause parts of houses to be poorly ventilated, especially at exhaust or gravity systems. Air leakage also includes open vents and windows etc.

The present Building Code (BBR 94, in effect from 950101) stipulates a supply air flow to bedrooms of 4 l/s and person as supplement to the requirement on overall airchange rates. Also, recirculating air is not allowed to bedrooms anymore.

In a large investigation performed by KTH, Dept. of Built Environment, 737 single-family houses in 60 Swedish communities built before 1988-89 were investigated by interviews and measurements. The investigation involved, among other things, potential moisture problems and measurements of airchange rates. In the following, some relevant results of the investigation are reviewed. The reference list contains some of the reports (in Swedish) that are relevant for this paper's topic.

From the total number of single-family houses in Sweden built by 1990, 1,7 million, 1,35 million houses have natural gravity ventilation systems installed. Most of them are built before 1976. About 500 000 houses with natural gravity ventilation systems are built 1961-1988 of which about 1/3 are heated with electric resistance systems. Of these, only about 10 % have chimneys.

Window condensation occurs to a relatively large extent. More in the houses built before 1975 (30%) than in the newer (13%). Differences in glazing and airchange rates explains the difference. Moisture problems in wet rooms occurs in about 15 % of the houses, relatively independent of house age. This indicates that even in new houses with mechanical ventilation systems there are difficulties to manage the moisture load from showers, laundry etc. that exist today.

Performed measurements in the houses show low airchange rates compared to the present Swedish Building Code requirement, 0,35 $l/s \cdot m^2$. About 80 % of the single-family houses do not meet this requirement and the measured airchange rates was lowest for the houses ventilated with natural gravity system even though the measurements were performed in winter. Table 1 gives measured ventilation air flows for houses with different ventilation systems. Average values for houses with gravity ventilation systems built after 1961 show even lower values, about 0,17 $l/s \cdot m^2$, with some variation between different parts of the country.

Table 1. Measured ventilation air flows in Swedish single-family houses as result of the ELIB investigation which can be compared with the code requirement: 0,35 $l/s \cdot m^2$.

Type of ventilation system	Ventilation air flow, $l/s \cdot m^2$
Gravity	0,23
Exhaust	0,24
Supply and exhaust	0,29

Risk group

In conclusion, a potential risk group could be identified as:

- Houses with gravity ventilation systems built after about 1961
- House that don't have chimneys and own boiler or changed to direct electric or district heating.
- One-story houses with low roof pitch.
- House where supply air devices are missing or where air sealing measures have been performed.

The size of this risk group can roughly be estimated to between 200 000 and 250 000 houses in Sweden.

2. Scope

Draw-backs with installing ventilation systems constructed according to present codes are that they either are very expensive to install, or increase energy use requiring full continuous ventilation, or don't guarantee the airchange rates in all rooms, e.g. in bedrooms at night.

The scope of this project was to find a relatively cheap ventilation system that in a simple way guarantees proper airchange rates in single family houses. The primary target was houses with natural gravity ventilation systems as indicated in the risk group defined previously.

The following criteria was considered important in the development process:

- occupancy control,
- secure bedroom night-time ventilation,
- secure bathroom daytime ventilation,
- easy maintenance and service.

Kitchen ventilation obtained low priority because most houses have cooker-hood fans installed that are used intermittently. Living and dining rooms were also considered less important. The thought behind occupancy control is that the occupants can direct air flows to where they are most needed i.e. bring the ventilation with them. In this way, the total air flow from the house could be reduced without a reduction in comfort and the ventilation energy losses are decreased compared to Swedish standard systems. This idea was penetrated by Stig Jahnsson already in 1979 and has since 1988 been tested and developed in a series of full scale test houses. The earlier tests included the use of small, 15 W, fans that were installed in bedrooms, bathrooms and living rooms in different concepts. The fans were tested both in supply and exhaust modes. For several reasons, this concept was abandoned *(Levin and Isaksson 1991)*.

The occupancy-controlled exhaust air system

In principle, the system is an extended exhaust fan ventilation system. The differences from a regular system are that ducts are drawn to bedrooms and that no ventilation outlets in kitchen or closets are installed. For older houses, existing ductwork are used as much as possible. An exhaust fan is connected and placed for easy access for cleaning and service. A schematic drawing of the system principle is shown in Figure 1.

Occupancy control is achieved by the exhaust air devices that are opened when needed, e.g. in bedrooms at bedtime. The bathroom air flows then reduces. The speed control of the fan (5 speeds) allows the occupants both to reduce the total air flow when the house is empty and increase the air flow in the summer or when many people are present. The capacity is also enough to fulfil the Building Code requirement, 0,35 l/sm^2 of continuous air flow when that is desired.

The objective of easy cleaning and service has been met by using a very serviceable fan type with removable fan blade. The exhaust air outlet has partly been developed within this project. Adjustment and locking is simple and it contains a filter where cleaning does not spoil the adjustment. It could be used in bedrooms, where the air flow is maximised, or in bathrooms where the air flow is minimised. Thus could the bathroom air flow be increased when needed. Air flow measurements could be done with standard measurements. Special measurement lids has also been manufactured in order to facilitate air flow measurements, where pressure difference measurements are used together with a calibration diagram.

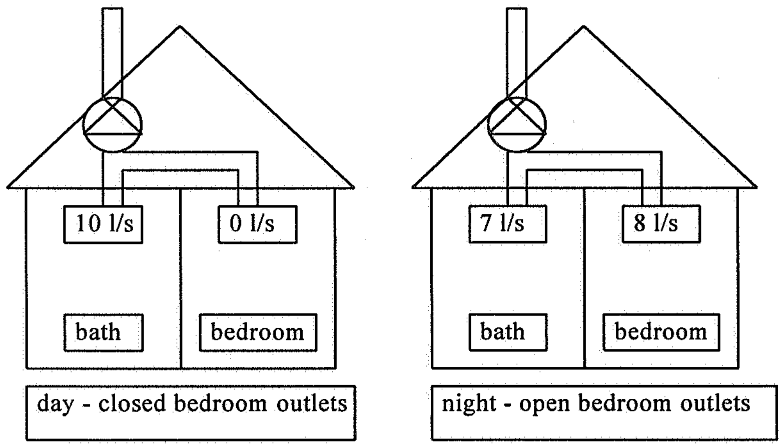

Figure 1. The principle for the occupancy-controlled exhaust air ventilation system with examples of obtained air flows for day and night operation in a bathroom and a master bedroom. The fan is placed to obtain easy access and low noise disturbance.

3. Full-scale test houses

Four houses in different Swedish cities were chosen as full-scale test objects. The main purpose was to see installation time, potential problems and cost, ease of adjustment and performance. The four houses all had natural gravity ventilation systems before measures. The reasons for needing improved ventilation were different, Radon gas from building material, unevenly distributed air leakage between floors or different parts of a house or just low airchange rates. An example of a floor plan for one of the test houses is given in Figure 2.

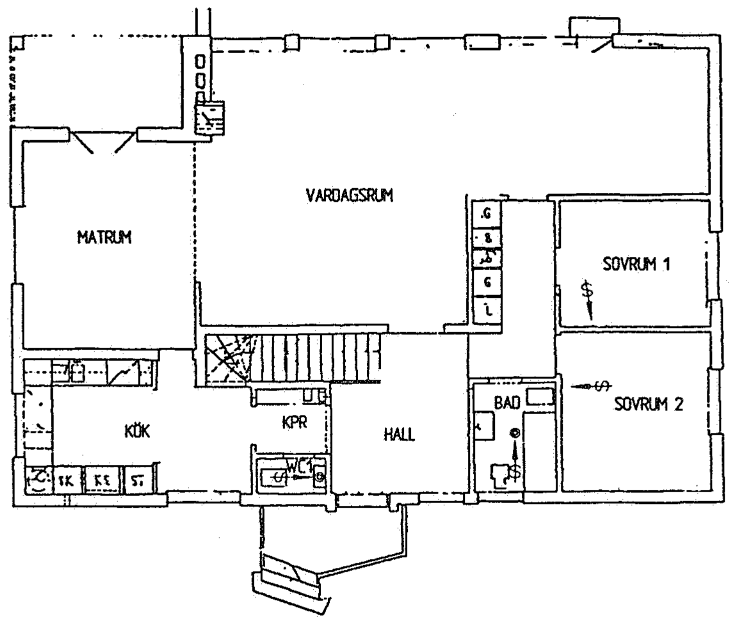

Figure 2. First floor plan with exhaust air points for one of the test houses. This 1 1/2-story house was built 1960 and the floor area is 133 m^2 for this floor.

Commissioning

Air flows were adjusted for day and night operating mode, where the target was to achieve air flows according to the building code in bathrooms at daytime and in bedrooms at night (4 l/s and person). In guest toilets that are not frequently used, air flows were adjusted lower than the Building Code requirement of 10 l/s. This means the total exhaust air flow from the house is much smaller than the air flow according to the Code. An example of adjusted air flows for the test house shown in Figure 2 at different operating modes is given in Table 2.

Table 2. Example of air flows at different operating modes for the 1 1/2-story house of about 180 m^2 at different fan speeds. The outlet devices have an adjustment scale marked 0 to 10. The adjusted flows for fan speed 2 were a little low, why normal operation is on fan speed 3.

Room	Fan speed 2				Fan speed 5 (all open max.)	
	"Day"		"Night"		"Summer"	
	Adj.	l/s	Adj.	l/s	Adj.	l/s
Bath	3,0	10		5,6	10	15
WC	1,0	3,6		1,9	10	12,5
WC (upstairs)	1,2	5,0		2,2	10	12,5
Bedroom 1	closed	0	5,0	4,2	5,0	5,8
Bedroom 2	closed	0	10	6,4	10	9,4
Bedroom 3 (upstairs)	closed	0	3,0	3,9	3,0	6,7
Bedroom 4 (upstairs)	closed	0	2,4	3,3	2,4	4,4
Total exhaust air flow		18,6		27,5		66,3

4. Evaluation

In addition to the measurements of exhaust air flows, air change rates have been measured in bedrooms with tracer gas decay and complete mixing. These measurements were made with closed doors at both night and day operating modes. Measurements of pressure differences, noise levels and Radon gas concentrations have complemented the evaluation when relevant. The occupants were also interviewed of how they perceived the changes from the conditions before measures. A sensitivity analysis was performed with a multizone air exchange calculation model.

Airchange rates

Measured airchange rates in bedrooms were in general very close to the exhaust air flows in the rooms. One concern was whether extra supply vents had to be installed in the bedrooms. In spite of that none of the houses had supply vents, no large pressure drops were measured across the building envelopes which would indicate the need for supply vents or to remove a part of the weather-stripping. What the tracer gas measurements did not show was whether the air to the bedrooms comes from the outside or from other parts of the building. In one case, the air change rate in a bedroom was higher because of air flow to a nearby bathroom. Additional air flow because of very strong winds was also seen in one case.

Energy costs

The energy use could be expected to increase by the measures, both from the increased airchange rate and electricity use for the fan. In continuous use at middle fan speed (135 V) an electric use of about 315 kWh per year could be expected for operating the fan.

A energy comparison between the occupancy-controlled system and a standard exhaust-supply system with an air-to-air heat exchanger have been made for the test houses. For the occupancy-controlled system, the lowest fan speed was assumed be used 40 hours per week and night ventilation mode was used 70 hours per week. At the exhaust-supply system, a temperature efficiency of 60 % was assumed on the heat exchanger. The weekly average effective air flow for the shown test house then becomes 21,9 l/s for the occupancy-controlled system and 25,3 l/s for the exhaust-supply system with heat recovery. In the Stockholm area, this roughly corresponds to an energy use of about 400 kWh/year more for the latter system. To this, electric use for the supply fan, maybe 350 kWh/year, should be added. Further, the effects of air infiltration and exfiltration has not been considered in this comparison.

Practical experiences

The man-hours for installing the systems have varied greatly because of ease to connect to existing ductwork and how the ducts to the bedrooms could be placed and insulated (outside the envelope). In total, up to 24 man hours of installation work was required for one house. The commissioning of the systems was very easy to perform, especially when using the flow-pressure diagrams and a simple manometer. The air flow adjustment procedure should not take more than an hour for a normal single-family house.

All the occupants in the four test houses are very satisfied with the system and claims better indoor air quality in bedrooms. For some, it is hard to remember to close the bedroom outlets in the morning. No complaints on noise levels have occurred and ventilation noise were in most cases hard to notice.

Computer calculations

For a simple sensitivity analysis of the system, the IDA Multizone Air Exchange simulation module was used. The computer program was developed from the MOVECOMP code *(Sahlin and Bring, 1995)*.

The previously shown test house was chosen to form the input data. The floor plan of the first floor was simplified into five zones for the analysis. WC, bath and bedroom 1 faced the windward side, the smaller bedroom 2 the gable side and the rest of the floor faced the leeward side. A block diagram over the zones and connecting air leakage paths is shown in Figure 3.

Two levels of airtightness were considered. One according to the present Swedish Building Code, that requires a maximum air leakage of 0,8 l/sm^2 at 50 Pa, where the area relates to exterior building envelope area. The air leakage level is about the same as was first introduced in 1978. This level might be considered as too airtight for older houses, but is relevant for the risk group as defined in the beginning of this paper. Also, sealing measures as replacing weather-strips, caulking, etc. have often been performed. The other air leakage level was chosen to 1,5 times the present code value, thus 1,2 l/sm^2 at 50 Pa.

Only effects of different air leakage paths and wind speed were considered in this study. Buoyancy-driven air flows were thus not modelled. Other simplifications included one-way air flow through open doors, constant exhaust air flows, no air leakage through floor and ceiling and interior air leakage paths dominated by door leakage. Closed doors were assumed to have a 1 mm gap around the perimeter.

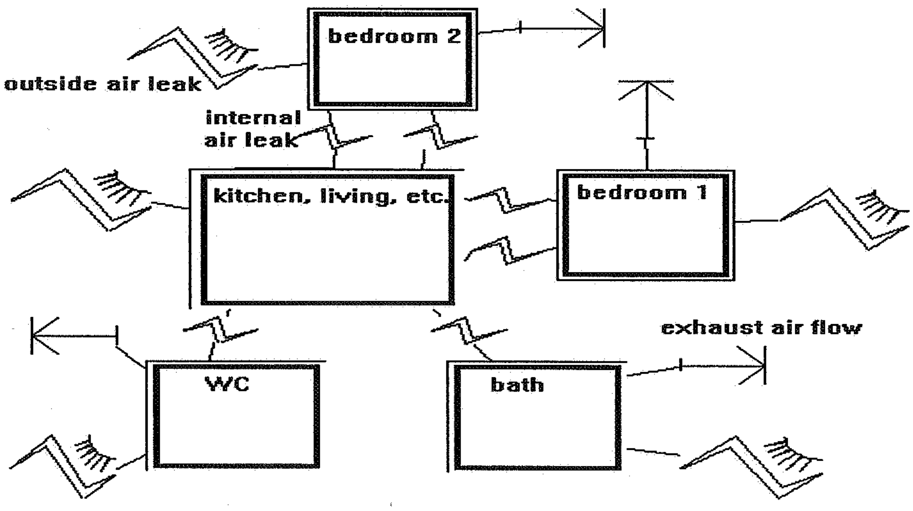

Figure 3. Simplified zone model with air leakage paths as used in the calculations.

The most important operating mode to study was night-time and air change rates in bedrooms with closed doors. Opening the bedroom exhaust outlets increased the air change rates to the setting of the mechanically controlled air flows, several times more than without exhaust. For the tighter case, about half of the air leakage came through the closed interior door and half through infiltration. Adding an outside vent in a bedroom caused the major part of the supply air to the room to enter through the vent.

With the doors closed, pressure differences up to 20 Pa occured between the small wet rooms and the house, which indicate that the air leakage between the rooms are underestimated or the need for slot vents in interior partitions to the wet rooms. When opening the interior doors, pressure differences between rooms reduces to very small numbers, less than 0,1 Pa. These two cases form the range of pressure differences, and the reality is likely to be in between. The calculations showed that door opening does not have a major effect on the air flows between rooms in the tighter test case except in combination with an open outside vent and 6 m/s wind speed.

The small bedroom 2 with side wind obtained more air from the house at increasing wind speed. The size of the air leakage in combination with the exhaust air flows made the air exfiltration through the envelope very small if no outside vents were installed.

5. Conclusions

Installing and commissioning the occupancy-controlled exhaust air ventilation systems in the four test houses were performed as intended. Sometimes, extra man-hours were needed e.g. to properly connect to the old ductwork, which could be of non-standard shapes and sizes. Adjustment of design air flow was a relatively reliable and easy procedure using the special exhaust air outlets.

Tracer gas measurements have shown substantial increases in airchange rates for bedrooms with exhaust air, which also was verified with multizone airchange calculations. The amount of air that comes directly from the outside depends on the air leakage paths and their distribution in the house. At air leakage levels according to the Swedish Building Code, an outside air vent should be installed in bedrooms to avoid excessive pressure drops across the building envelope. However, adding more outside vents (which are large air leaks) will increase the risk of air exfiltration through the building envelope because of wind pressures and thereby contribute to uncontrolled increase in airchange rates. The buoyancy-driven air flows should be considered for two-story buildings.

The increase in energy use for heating ventilation air could be regarded as small if the occupants use the features of the system as intended. A comparison with an exhaust and supply air system with an air-to-air heat exchanger as commonly used for new houses, show that the occupancy-controlled system in most cases will have lower ventilation energy losses. The installation cost in an older house is about one third for the occupancy-controlled system compared to the exhaust and supply air system with an air-to-air heat exchanger.

6. Acknowledgements

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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Air Dehumidification by Absorption

(A Model for Numerical Calculation)

Prof. Dr.-Ing. F. Steimle
Dipl.-Ing. M. Reckzügel
Dipl.-Ing. J. Röben
University of Essen, Germany

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Synopsis

Especially in modern buildings with small capacity of humidity storage it is necessary to reduce the humidity in the supply air. Normally this was done by using a refrigeration system mostly with CFC´s. There are some alternative fluids available, but mostly they show a high global warming potential. All these systems need electrical energy to be driven and therefore it is necessary to consider other possibilities with alternative systems.

The most promising systems are sorptive systems that are used now in open cycles. In these systems the air is dehumidified by a liquid sorbens and cooled indirectly by evaporating water in an open circuit. In order to calculate the process of absorption on several conditions, computer based calculations are required. A model that describes the dehumidification - process is introduced.

List of Symbols

A	transfer area	[m²]
c_p	specific heat capacity	[J/kg K]
C	water content (solution)	[kg water/kg salt]
dx	width of an element	[m]
dy	length of an element	[m]
h	specific enthalpy	[J/kg K]
k	coefficient of heat transmission	[W/m² K]
$\dot{m}$	mass flow	[kg/s]
p	partial pressure of vapour	[Pa]
p^*	partial pressure (saturation)	[Pa]
$\dot{Q}$	heat flux	[W]
r_0	specific heat of evaporation	[J/kg]
T	temperature	[°C]
$\bar{T}$	mean temperature	[°C]

V	variable of state	[-]
X	water content (air)	[kg water/kg dry air]
X*	water content (saturation)	[kg water/kg dry air]
y	concentration	[kg salt/kg solution]
	y - coordinate	[m]

Greek Symbols

α	heat transfer coefficient	[W/m²K]
β	mass transfer coefficient	[kg/m² s], [kg/m² s Pa]
η	dynamic viscosity	[Pa s]
ρ	density	[kg/m³]

Indices

CW	cooling water
G	gas (air)
i	counter of elements (x-direction)
j	counter of elements (y-direction)
L	liquid (solution)
LG	liquid-gas
S	salt
V	vapour
X	water content

1. Introduction

Heating, ventilating and air conditioning (HVAC) systems are build to process outdoor air to a special indoor air condition. The demand of the air quality depends on the kind of building. In an industrial building the quality of the products is very important, but in an office building the thermal comfort of the employees must be guaranteed. Basically the parameters temperature and humidity can be changed by special components of a HVAC-system.

Especially in modern office buildings with small capacity of humidity storage it is necessary to reduce the humidity in the supply air. The classical way to dehumidify the outdoor air is using a refrigeration system. Everybody knows the problems and the discussions about the refrigerants. For that it is very important to find alternative components to dehumidify the air.

An interesting technique to dehumidify air is using hygroscopical solid or liquid substances. The adsorption respectively absorption technology used in HVAC-systems is nowadays a good enlargement to the existing refrigeration systems but particulary by using liquid desiccants a high demand of research work is necessary. In this paper a model to calculate the dehumidification - process is described.

2. Dehumidification by Absorption

Several technologies in order to dehumidify moist air are imaginable. One possibility to remove water out of the air is to cool the air below the dew point. In this case water will condense at the surface of a chiller or water droplets. To reach the necessary dew point the condensation requires temperatures of nearly 6°C, these low temperatures can be realised by water cooled chillers. To guarantee comfort in the room the air often must be reheated after the dehumidification. Because cooling and heating of supply air consume a large amount of energy other methods of dehumidification are investigated.

By cooling the surface of a chiller below the dew point, the vapour-pressure at the surface is very low compared with the vapour pressure in the air flow. This pressure gradient will cause a movement of vapour in direction of the cold surface where condensation follows.

This mechanism is the basis to develop other methods of air dehumidification. There are materials, which reduce the vapour-pressure without being cooled and even when they are at a higher temperature than the air. Materials that reduce the vapour-pressure at their surface are called hygroscopic.

Desiccating by contact with hygroscopic materials is distinguished by the kind of the used materials, there are both solid and liquid desiccants. There is a great choice of solid materials for the use to dehumidify air. They can be used either for technical drying or in air conditioning plants. Silicagel and hygroscopic salts are privileged for the application in air dehumidifiers with solid desiccants.

This paper will introduce a model of an absorber in order to simulate the behaviour.

3. The open absorption cycle

The transfer of vapour between two phases causes a simultaneous transfer of heat in mostly the same direction. The difference between the temperatures of air and liquid causes a sensible heat transfer while the condensation of vapour initiates the occurrence of latent heat. So the total heat effect can be written as follows:

$$\dot{Q} = \alpha \cdot A \cdot (\overline{T}_G - \overline{T}_L) + \beta_p \cdot A \cdot (p - p^*) \cdot r_0 \qquad (1)$$

Moist air at the surface of the desiccant is assumed to be saturated. The knowledge of the dew point in dependence of the solution's conditions [4] allows the calculation of the water content of the saturated air. The difference of the partial pressures can now be replaced by the difference of the water contents of air.

$$\dot{Q} = \alpha \cdot A \cdot (\overline{T}_G - \overline{T}_L) + \beta_X \cdot A \cdot (X - X^*) \cdot r_0 \qquad (2)$$

Both the latent and the sensible heat lead to a higher temperature of the liquid desiccant. The appearing absorption heat is neglectible. Higher temperatures cause higher dew points and a higher water content by saturation, the hygroscopicity of the solution decreases. Therefore the plates of the absorber are cooled internally with water in order to prevent this effect. The temperature of the cooling water can be higher than the dew point, a refrigerating assembly is not necessary.

An experimental apparatus was constructed to obtain the quantity of the appearing heat and mass transfer coefficients. Fig. 1 shows the structure of one plate, the test-absorber consists of five plates.

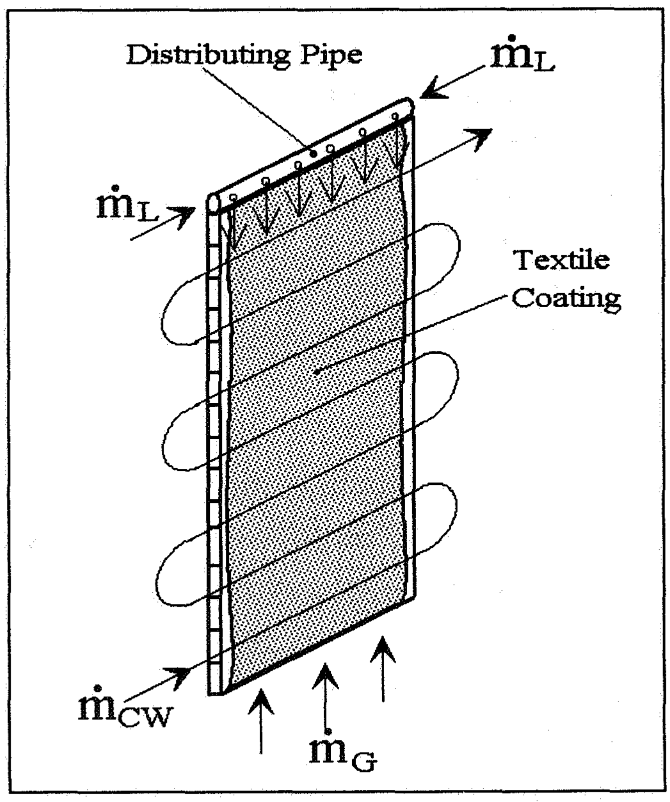

Fig. 1: Structure of an absorber plate

4. Development of the Model

4.1 Desiccants and Physical Properties

There's a great variety of hygroscopic materials for the use in the absorber. Desiccants for the purpose of technical drying are aqueous solutions of lithium-chloride (LiCl) or calcium-chloride ($CaCl_2$). The application in climate plants is also possible. In addition to this pure materials there's a mixture of $CaCl_2$, $Ca(NO_3)_2 \cdot 4H_20$ and an emulsifier „S“ called „Klimat 3930 S“ [2]. The behaviour of aqueous solutions of LiCl, $CaCl_2$ and Klimat 3930 S is investigated.

Some physical properties of the different solutions at varying temperatures and concentrations must be known. The most important are:
- density ρ,
- viscosity η,
- dew point at the surface of the desiccant respectively the water content at saturation,
- heat capacity of the aqueous solution.

4.2 The Model

To formulate the problem, the following assumptions are made:
- the flow is fully developed and laminar
- diffusion thermal effects are negligible
- the surface temperature of the liquid is equal to the bulk temperature
- the only gradient of variables of state occurs in the direction of flow
- the main resistance for the mass transport is in the air

The behaviour of the different substances are described by a system of differential-equations. These equations cannot be solved in an analytic way, the integration must procced numerically. The point-slope method developed by Euler and Cauchy allows an easy solution of the system [1]:

$$V_{j+1} = V_j + dy \cdot V_j'(y_j, V_j) = V_j + \Delta V_j \quad j = 0, \ldots, m \qquad (3)$$

y ... place, V ... variable of state

y_0, V_0 ... quantities at the beginning of the calculation (estimation)

(to estimate the properties of cooling water, j has to be substituted by i, y by x)

Because the gradient of the variables will change across the plate, the absorber must be devided into a number of elements. The gradients will be calculated in each element. Fig. 2 shows the model for the calculation of the absorber. Equation 3 shows that the variables and it's gradients must be calculated to estimate the quantities in the following element. Fig. 3 illustrates the method of calculation when the gradients in two adjacent elements are known.

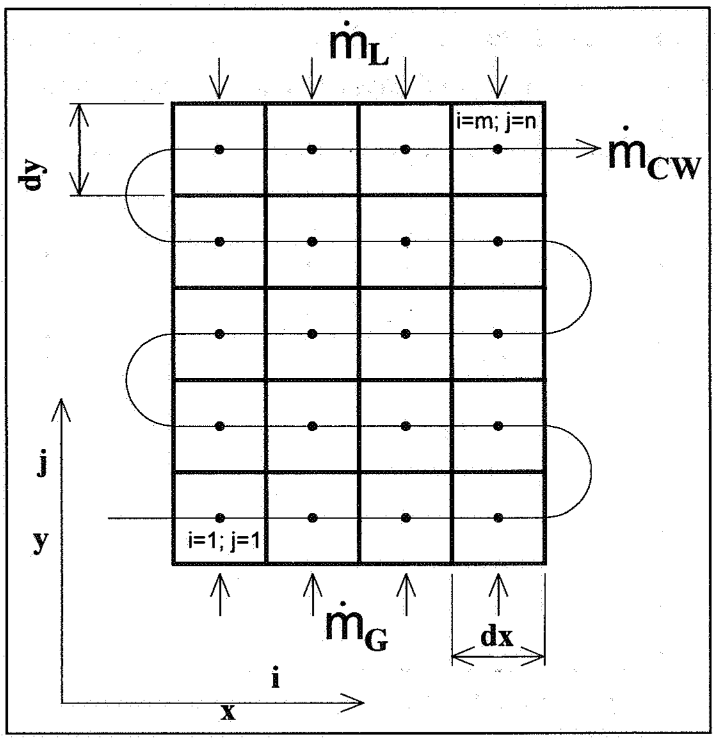

Fig 2 : Model of the absorber (one plate)

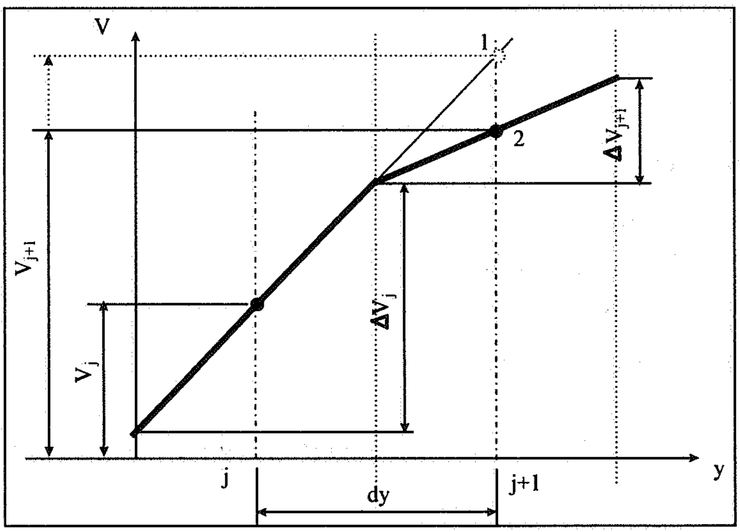

Fig. 3 : Calculation of variables of state in adjacent elements

Without knowing the value of ΔV_{j+1} point 1 in Fig. 3 is calculated:

$$V_{j+1} = V_j + dy \cdot V_j'(y_j, V_j) = V_j + \Delta V_j \quad (4)$$

Knowing the gradient of V_{j+1}, V_{i+1} can be calculated at point 2 in Fig. 3:

$$V_{j+1} = V_j + dy \cdot 0{,}5 \cdot \left(V_j'(y_j, V_j) + V_{j+1}'\left(y_{j+1}, V_{j+1}\right)\right) \quad (5)$$

The gradients of the variables of state are evaluated by solving the mass and energy balance and the basic equation for the heat- and mass transfer in each element. Fig. 4 shows one element and the mass flows.

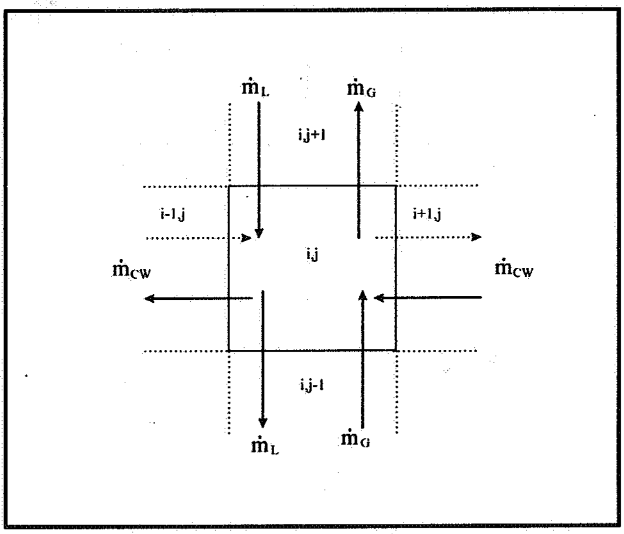

Fig. 4: Element and mass flows (mass flow of cooling water: dotted: j odd, continuos: j even)

The derivation of the equations to calculate the gradient of the air temperature and enthalpy is shown below exemplary.

Air:

IN: $T_{G\,in} = T_{G\,i,j-1} + ½\,\Delta T_{G\,i,j-1}$

$X_{in} = X_{i,j-1} + ½\,\Delta X_{G\,i,j-1}$

$h_{G\,in} = h_{G\,i,j-1} + ½\,dh_{G\,i,j-1}$

OUT: $T_{G\,out} = T_{G\,i,j-1} + ½\,\Delta T_{G\,i,j-1} + \Delta T_{G\,i,j}$

$X_{G\,out} = X_{G\,i,j-1} + ½\,\Delta X_{G\,i,j-1} + \Delta X_{G\,i,j}$

$h_{G\,out} = h_{G\,i,j-1} + ½\,\Delta h_{G\,i,j-1} + \Delta h_{G\,i,j}$

The modification of the specific enthalpy in element i,j can be calculated as follows:

$$\Delta h_{G\,i,j} = c_{pG} \cdot \left(T_{G\,i,j-1} + \frac{1}{2} \cdot \Delta T_{G\,i,j-1} + \Delta T_{G\,i,j} \right) + \left(X_{G\,i,j-1} + \frac{1}{2} \cdot \Delta X_{G\,i,j-1} + \Delta X_{G\,i,j} \right) \cdot \left(r_0 + c_{pV} \cdot \left(T_{G\,i,j-1} + \frac{1}{2} \cdot \Delta T_{G\,i,j-1} + \Delta T_{G\,i,j} \right) \right) - c_{pG} \cdot \left(T_{G\,i,j-1} + \frac{1}{2} \cdot \Delta T_{G\,i,j-1} \right) - \left(X_{G\,i,j-1} + \frac{1}{2} \cdot \Delta X_{G\,i,j-1} \right) \cdot \left(r_0 + c_{pV} \cdot \left(T_{G\,i,j-1} + \frac{1}{2} \cdot \Delta T_{G\,i,j-1} \right) \right) \quad (6)$$

Equation 6 is used to determine the change of temperature in element i,j. The variation of the enthalpy is given by change of temperature and humidity (equation 7).

$$\Delta h_{G\,i,j} = \frac{1}{\dot{m}_G} \cdot \alpha_{LG} \cdot dx \cdot dy \cdot \left(\bar{T}_{G\,i,j} - \bar{T}_{G\,i,j} \right) + r_0 \cdot \Delta X_{i,j} \quad (7)$$

y is known as the concentration (kg salt/kg solution) and C characterises the water content of the salt (kg water/kg salt). The relation between y and C is given in equation 8.

$$C = \frac{1-y}{y} \quad (8)$$

Because the mass flow of salt and dry air are constant, the specific enthalpy is related to them, equation 9 correlates the change of humidity and water content of the solution.

$$\Delta C_{i,j} = -\frac{\dot{m}_G}{\dot{m}_S} \cdot \Delta X_{i,j}; \quad \dot{m}_S = \dot{m}_L \cdot y \quad (9)$$

The physical properties and the gradients depend on temperatures and water contents. The solution of the system of differential equations requires an iterative calculation. The following equations show the complete system.

I. Water content of the air:

$$\Delta X_{i,j} = \frac{\beta_{i,j}}{\dot{m}_G} dx\, dy \left(X_{i,j} - X^*_{i,j} \right)$$

II. Water content of the solution

$$\Delta C_{i,j} = -\frac{\dot{m}_G}{\dot{m}_S} \Delta X_{i,j}$$

III. Specific enthalpy of the solution:

$$\Delta h_{L;i,j} = -\frac{2\,\dot{m}_G\, \Delta h_{G;i,j} + 2\,\dot{m}_S\, \Delta C_{i,j}\, h_{L;i,j+1} + 2\,\dot{m}_{CW}\, c_{p;CW}\, \Delta T_{CW;i,j} + \dot{m}_S\, \Delta C_{i,j}\, \Delta h_{L;i,j+1}}{\dot{m}_S \left(2 + 2\, C_{i,j+1} + 2\, \Delta C_{i,j} + \Delta C_{i,j+1} \right)}$$

IV. Temperature of the air:

$$\Delta T_{G;i,j} = -\frac{2\,\Delta X_{i,j}\left(r_0 + c_{p;V}\,T_{G;i,j-1}\right) - 2\,\Delta h_{G;i,j} + \Delta X_{i,j}\,c_{p;V}\,\Delta T_{G;i,j-1}}{2\,c_{p;G} + c_{p;V}\left(2\,X_{i,j-1} + 2\Delta X_{i,j} + \Delta X_{i,j-1}\right)}$$

V. Temperature of the cooling water:

$$\Delta T_{CW;i,j} = \frac{k_{i,j}\,dx\,dy\left(T_{L;i,j+1} + \frac{1}{2}\left(\Delta T_{L;i,j+1} + \Delta T_{L;i,j}\right) - T_{CW;i+1(-1)^j,j} + \frac{1}{2}\left(\Delta T_{CW;i+1(-1)^j,j} + \Delta T_{CW;i,j}\right)\right)}{\dot{m}_{CW}\,c_{p;CW}}$$

VI. Specific enthalpy of the air:

$$\Delta h_{G;i,j} = \frac{1}{\dot{m}_G}\,\alpha_{LG;i,j}\,dx\,dy\left(T_{G;i,j-1} + \frac{1}{2}\left(\Delta T_{G;i,j-1} + \Delta T_{G;i,j}\right) - T_{L;i,j+1} + \frac{1}{2}\left(\Delta T_{L;i,j+1} + \Delta T_{L;i,j}\right)\right) + r_0\,dX_{i,j}$$

VII. Temperature of the solution:

$$\Delta T_{L;i,j} = f(\Delta h_{L;i,j}) = \frac{\Delta h_{L;i,j}}{c_{pL}\left(y, T_{L;i,j}\right)}$$

A function that correlates enthalpy (or specific heat capacity), concentration and temperature of the solution must be known to calculate the change of the solution temperature. This interrelation depends on the composition of the solution.

While the properties of the different desiccants are well known, the calculation of the heat and mass transfer coefficients is difficult, investigations and measurements are essential for each type of absorber.

5. Conclusions

The introduced activity at the University of Essen is supported by the German *Bundesminister für Bildung, Wissenschaft, Forschung und Technologie (BMBF)*. The aim of this work is the investigation of an absorptive dehumidifier for the inset in air conditioning plants. The attention is turned on energetic aspects and the calculation of these systems. In order to calculate the complete cycle (with evaporative cooling of the air and regeneration of the solution) the model has to be expanded. Several results of measurements are made available from the Bavarian Center of Applied Energy Research (ZAE Bayern), Munich.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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A CONTROL SYSTEM THAT PREVENTS AIR FROM ENTERING AN AIR-HANDLING UNIT THROUGH THE EXHAUST AIR

John E Seem*, John M House**

* Johnson Controls Inc, Milwaukee, WI, USA

** National Institute of Standards and Technology, Gaithersburg, MD, USA

A Control System that Prevents Air from Entering an Air-Handling Unit through the Exhaust Air Damper

John E. Seem, Ph.D. John M. House, Ph.D.

Synopsis

Traditional air-handling unit (AHU) control systems link the position of the exhaust air damper, recirculation air damper, and outdoor air damper. Tests at the National Institute of Standards and Technology (NIST) on a variable-air-volume (VAV) AHU have shown that air can enter the AHU through the exhaust air damper. This can negatively impact indoor air quality if the exhaust air duct is located near a pollution source.

This paper presents a new control system for variable air volume AHU's that use volume matching to control the return fan. The new control system links only the position of the exhaust air damper and recirculation air damper. During occupied times, the outdoor air damper is in the fully open position.

Simulation and laboratory results are presented to compare the new control system and a traditional control system. Several cases are simulated to examine the effect of damper sizing and system load on airflow in AHU's. The simulations demonstrate that the new control system can prevent air from entering the AHU through the exhaust air damper for conditions that the traditional control system cannot. A case demonstrating the limits of the new control system to prevent this phenomenon is included in the simulation results. The laboratory results provide further evidence that the new control system prevents air from entering the AHU through the exhaust air damper for conditions that cause the phenomenon with the traditional control system.

Introduction

The primary function of an AHU is to provide conditioned air to various rooms in a building. An AHU typically has dampers that are used to control the amount of outdoor air that enters the system, the amount of air exhausted from the system, and the amount of return air from the rooms that is recirculated through the system. Damper control is generally influenced by two main factors, namely, the need to provide ample outdoor air to meet indoor air quality (IAQ) standards, and a desire to conserve energy by limiting heating and cooling in the AHU coils. AHU's are designed with the intent that outdoor air enter the system only through the outdoor air duct and dampers. However, tests at NIST have shown that air can also enter an AHU through the exhaust air damper. Air entering the AHU through the exhaust air damper can have a negative impact on IAQ if, for example, the exhaust air damper is located near a pollution source such as a truck loading dock. Also, in some AHU's, prefilters and/or preheat coils are placed in the outdoor air duct. Air entering the AHU through the exhaust air damper bypasses these components. If the prefilter is bypassed, IAQ may suffer. If the preheat coil is bypassed and the air is very cold, the coils in the AHU could be damaged or the freeze protection thermostat may cause the entire AHU to shut down.

The objective of this paper is to describe a new AHU control system that helps prevent air from entering a VAV AHU through the exhaust air damper. The new control system may also save energy by reducing fan power under certain operating conditions.

J.E. Seem is with Johnson Controls, Inc., Milwaukee, Wisconsin, U.S.A. and J.M. House is with the National Institute of Standards and Technology, Gaithersburg, Maryland, U.S.A.

The basic components and traditional control strategies that influence airflow in a VAV AHU are described first. The new control system is then introduced. Equations governing airflow in the AHU are then presented, followed by results comparing the traditional and new control systems. This is followed by experimental results comparing the two control systems. Conclusions of this study are then presented.

Description of Traditional Air-Handling Unit

Figure 1 is a schematic of a VAV AHU. The primary function of an AHU is to provide conditioned air to various rooms. Also, AHU's are usually controlled to maintain a positive air pressure in the building relative to the outdoors. This helps prevent unconditioned air from entering the building through cracks in the building structure.

The supply fan is controlled to maintain a static pressure in the supply duct at a setpoint value. A common strategy for controlling the return fan is to maintain a constant difference between the supply airflow rate and return airflow rate. This strategy is called volume matching. With a volume matching strategy, the difference in the volume of air entering and exiting the AHU through the exhaust and outdoor air ducts must equal the difference in the supply and return airflow rates.

VAV AHU's are usually controlled to maintain a constant discharge air temperature. This is accomplished by controlling the cooling coil, heating coil, and dampers to provide the desired discharge air temperature in the supply duct.

Today's AHU's link the position of the exhaust air damper, recirculation air damper, and

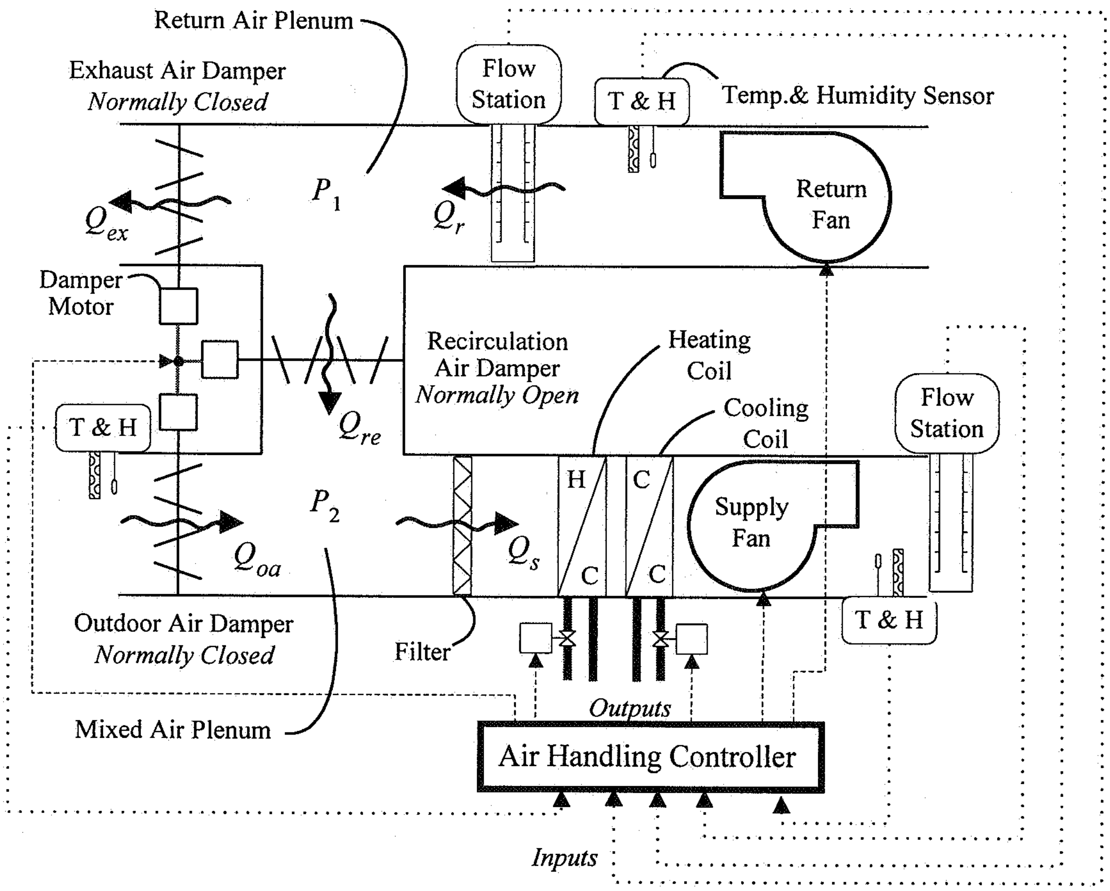

Figure 1 Schematic of air-handling unit.

outdoor air damper. The exhaust air damper and outdoor air damper are normally closed, and the recirculation air damper is normally open. As the exhaust and outdoor air damper begin to open, the recirculation air damper begins to close. Either mechanical linkage or the control system maintains the relationship between the three dampers. For traditional AHU's, the following equations describe the relationship between damper positions:

$$\theta_{re} = 1 - \theta_{ex} \tag{1}$$

$$\theta_{out} = 1 - \theta_{re} \tag{2}$$

where θ_{re} is the fraction of fully open position of the recirculation air damper, θ_{ex} is the fraction of fully open position of the exhaust air damper, and θ_{out} is the fraction of fully open position of the outdoor air damper.

New Air-Handling Unit Control System

The new AHU control system links the position of only the exhaust air damper and the recirculation air damper using the relationship in Equation 1. During occupied times, the outdoor air damper remains 100 percent open, i.e., $\theta_{out} = 1$.

Modeling of Airflow in Air-Handling Unit

This section reviews equations for modeling airflow in AHU's. The equations are based on the conservation of mass and energy. Assuming the density of air is constant throughout the system, conservation of mass applied at the mixed air plenum and return air plenum yields

$$Q_r = Q_{ex} + Q_{re} \tag{3}$$

$$Q_s = Q_{oa} + Q_{re} \tag{4}$$

where Q_r is the return airflow rate, Q_{ex} is the flow rate of air exiting the AHU through the exhaust air damper, Q_{re} is the recirculation airflow rate, Q_s is the supply airflow rate, and Q_{oa} is the flow rate of air entering the AHU through the outdoor air damper. Airflow rates just defined are shown in Figure 1. The airflow rates are related to air velocities and damper areas by

$$Q_{ex} = V_{ex} A_{ex} \tag{5}$$

$$Q_{oa} = V_{oa} A_{oa} \tag{6}$$

$$Q_{re} = V_{re} A_{re} \tag{7}$$

where A_{ex} is the area of the exhaust air damper, A_{oa} is the area of the outdoor air damper, A_{re} is the area of the recirculation air damper, V_{ex} is the velocity of air leaving the AHU through the exhaust air damper, V_{oa} is the velocity of air entering the AHU through the outdoor air damper, and V_{re} is the velocity of air going from the return air plenum to the mixed air plenum through the recirculation air damper.

If the total pressure in the return air plenum (P_1) is greater than atmospheric pressure (P_a), then return air will exit the AHU through the exhaust air damper. The energy equation for return air exiting the AHU through the exhaust air damper is

$$\frac{P_1}{\rho} + \frac{V_{ex}^2}{2} = \frac{P_a}{\rho} + \left(C_{exd} + C_{exit} + C_{screen}\right)\frac{V_{ex}^2}{2} \tag{8}$$

where P_1 is the static pressure in the return air plenum, ρ is the density of air, P_a is the atmospheric pressure, C_{exd} is the loss coefficient for the exhaust air damper, C_{exit} is the exit loss coefficient, and C_{screen} is the loss coefficient of a screen that is placed in the exhaust duct to prevent birds and other small animals from accessing the AHU ductwork. Other dynamic losses due to the ductwork are neglected. A damper loss coefficient is a function of damper position. For opposed and parallel blade dampers, the loss coefficient can be estimated from

$$C_{damper} = a_0 \, e^{a_1\theta + a_2\theta^2} \tag{9}$$

where a_0, a_1, and a_2 are constants that are determined from nonlinear regression, and θ is the fraction that the damper is fully open, e.g., if a damper is half-way open, then θ is 0.5.

If the atmospheric pressure (P_a) is greater than the total pressure in the return air plenum (P_1), then air will enter the AHU through the exhaust air damper. The energy equation for outside air entering the AHU through the exhaust air damper is

$$\frac{P_a}{\rho} = \frac{P_1}{\rho} + \frac{V_{ex}^2}{2} + \left(C_{exd} + C_{en} + C_{screen}\right)\frac{V_{ex}^2}{2} \tag{10}$$

where C_{en} is the entrance loss coefficient.

The energy equation for air entering the AHU through the outdoor air damper is

$$\frac{P_a}{\rho} = \frac{P_2}{\rho} + \frac{V_{oa}^2}{2} + \left(C_{oad} + C_{en} + C_{screen}\right)\frac{V_{oa}^2}{2} \tag{11}$$

where P_2 is the static pressure in the mixed air plenum and C_{oad} is the loss coefficient for the outdoor air damper. The energy equation for airflow from the return air plenum to the mixed air plenum is

$$\frac{P_1}{\rho} = \frac{P_2}{\rho} + C_{red}\frac{V_{re}^2}{2} \tag{12}$$

where C_{red} is the loss coefficient for the recirculation air damper.

The equations used in the AHU airflow analysis are idealized. However, it is believed that this level of analysis does capture the major physical characteristics of interest in this study.

Simulations

Simulations were performed to compare the traditional and new control systems. A simulation program [3] was used to solve the system of equations defined in the previous section. Four cases are compared. The base case simulation has the following characteristics:

Damper Geometry:	$A_{oa} = 2.5$ m^2	$A_{ex} = 2.0$ m^2	$A_{re} = 2.0$ m^2
Minor Losses:	$C_{en} = 0.5$	$C_{exit} = 1.0$	$C_{screen} = 0.32$
Airflow Rates:	$Q_s = 5.0$ m^3/s	$Q_r = Q_s$ - 1.0 m^3/s	
Air Properties:	$P_a = 101.4$ kPa	$\rho = 1.202$ kg/m^3	

Additionally, the constants a_0, a_1, and a_2 were obtained by performing a nonlinear regression to data for opposed blade dampers having a ratio of the sum of the damper blade lengths to duct perimeter equal to 0.5 [1,2]. Damper sizes and minor losses were selected based on published guidelines [2].

Figure 2 shows the flow rate of air through the exhaust air damper for the base case simulation for both the traditional and new control systems. Negative values of the exhaust airflow rate indicate that air is entering the AHU through the exhaust air damper. For the traditional control system, outdoor air enters the AHU through the exhaust air damper when it is less than 30 percent open. For the new control system, outside air does not enter the AHU through the exhaust air damper. By opening the outdoor air damper, the flow resistance through the outdoor air duct is reduced and the pressure in the mixing plenum is raised. This raises the pressure in the return plenum to the point where it is greater than atmospheric pressure, thus preventing the reverse airflow phenomenon. Note that as the exhaust air damper approaches the fully open position, the curves in Figure 2 converge. This desirable (albeit expected) result is seen in all ensuing comparisons of the two control systems.

Figure 3 shows the fraction of outdoor air to supply air versus exhaust air damper position for the two control systems. The outdoor air quantity includes air entering the AHU through the exhaust air damper. When the exhaust air damper position is less than 30 percent open, the fraction of outdoor air for the traditional control system remains constant. For this range of damper positions, the flow rate of outdoor air entering AHU through both the exhaust and outdoor air dampers equals the difference between the supply and return airflow rates, i.e., 1.0 m^3/s. Because $Q_s = 5.0$ m^3/s, the fraction of outdoor air to supply air is 0.2.

Figure 4 shows the exhaust airflow rates for the control systems when the area of the recirculating air damper is changed from 2.0 m^2 to 0.8 m^2. Notice that when the exhaust air damper is near the fully closed position, the traditional control system allows air to enter the AHU through the exhaust air damper whereas the new control system does not.

By comparing the results for the traditional control system in Figures 2 and 4, it is seen that the transition point for reverse airflow in Figure 4 occurs when the exhaust air damper is more fully closed. The smaller recirculation damper area corresponding to the results in Figure 4 causes the air velocity between the return air plenum and mixed air plenum to increase for a given airflow rate. Equation 12 shows that increasing the recirculation air velocity increases the pressure difference between the return and mixed air plenums. Thus, increasing this pressure difference helps prevent reverse airflow through the exhaust air damper. However, there is an energy penalty associated with increasing this pressure difference: the fan power will increase.

Figure 5 shows the exhaust airflow rates for the control systems when the supply airflow rate is reduced from 5.0 m^3/s to 3.0 m^3/s. Again, the traditional control system allows outdoor air to enter the AHU through the exhaust air damper and the new control system does not.

Comparing the results for the traditional control system in Figures 2 and 5, it is seen that the transition point for reverse airflow in Figure 5 occurs when the exhaust air damper is more fully open than in Figure 2. Because the airflow difference between the supply and return air ducts is constant, for a given damper position, a higher percentage of the supply air is outdoor air when the supply airflow rate is reduced. This means there is a larger driving potential in the system to draw air into the AHU through the exhaust air damper. Thus, it is expected that the reverse airflow problem occurs more often at low load conditions.

Figure 6 demonstrates a case in which the new control system fails to prevent the reverse airflow problem. This is a more extreme case of the low load example shown in Figure 5. In this case, the supply airflow rate was reduced to 1.5 m^3/s and all other parameters were set to their base case values. Both curves in Figure 6 represent results obtained using the new control system. The curve labeled $A_{re} = 2.0$ m^2 (base case value) shows that the new control system is unable to prevent the reverse airflow phenomenon when the exhaust air damper is nearly fully closed. The second curve shows that by reducing the area of the recirculation air damper (by disabling and closing one or more blades), air is prevented from entering the AHU through the exhaust air damper.

An alternative to reducing the area of the recirculation air damper is to limit its maximum open position. The failed case corresponding to the curve labeled $A_{re} = 2.0$ m^2 in Figure 6 is shown as the curve labeled $\theta_{re} = 1 - \theta_{ex}$ in Figure 7. The curve labeled $\theta_{re} = 0.5\ (1 - \theta_{ex})$ corresponds to the new control system with maximum open position of the recirculation air damper limited to 50 percent. Thus, by limiting the maximum open position of the recirculation air damper, air is prevented from entering the AHU through the exhaust air damper.

The results in Figures 6 and 7 are presented to demonstrate that the new control system does not eliminate the possibility of outdoor air entering the AHU through the exhaust air

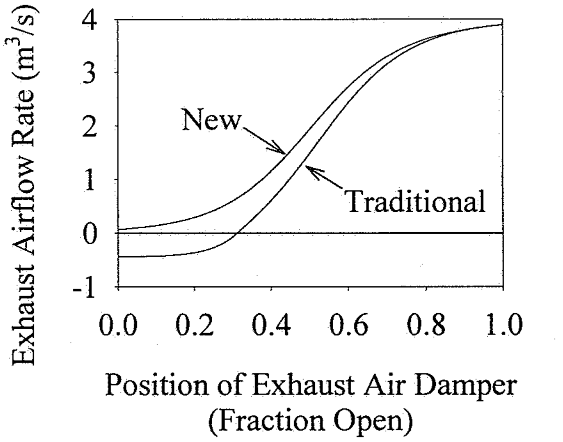

Figure 2 Exhaust airflow rate for base case.

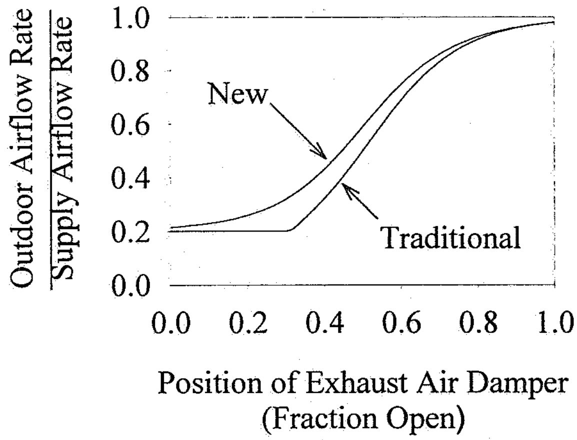

Figure 3 Fraction outdoor air for base case.

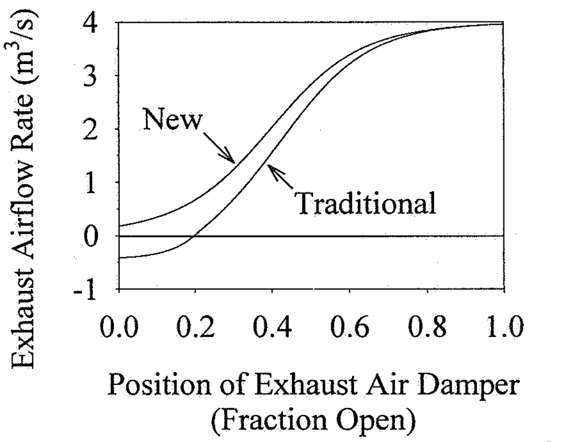

Figure 4 Modified base case: $A_{re} = 0.8$ m^2.

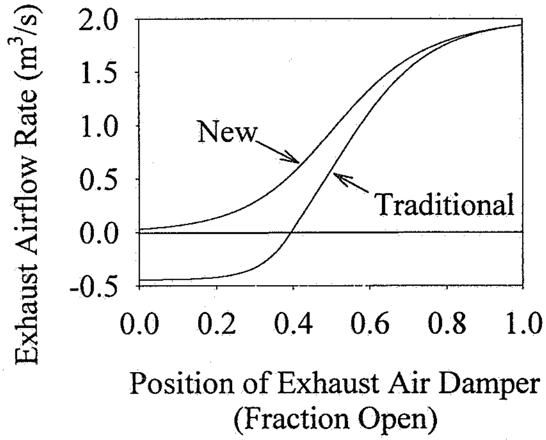

Figure 5 Modified base case: $Q_s = 3.0$ m^3/s.

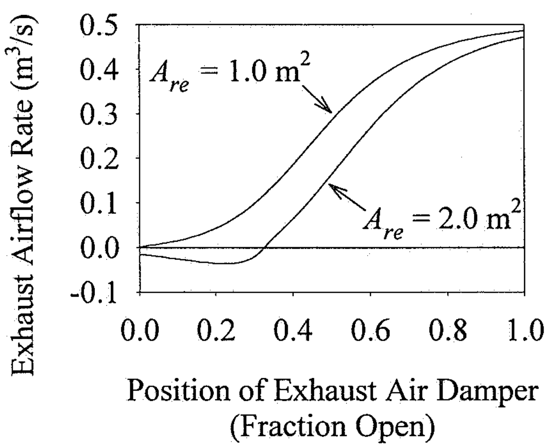

Figure 6 New control system when
$Q_s = 1.5$ m^3/s and $Q_r = 0.5$ m^3/s.

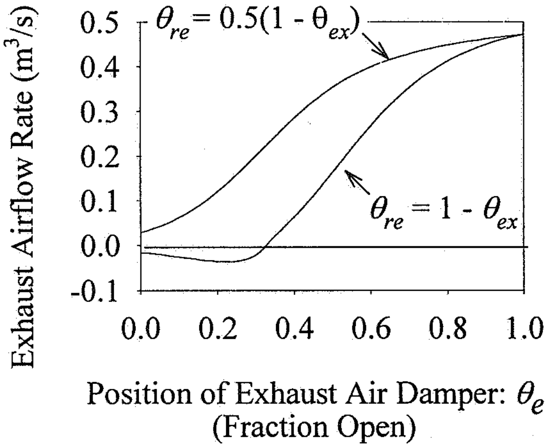

Figure 7 New control system when
$Q_s = 1.5$ m^3/s and $Q_r = 0.5$ m^3/s.

damper, it simply reduces the likelihood of occurrence of the phenomenon. Under extreme conditions, it may be necessary to modify the flow resistance through the recirculation air damper to alleviate the problem.

Laboratory Tests

To validate the simulation results, the control systems were implemented in the NIST laboratory AHU. The outdoor and recirculation air dampers of the laboratory AHU have a parallel blade geometry and the exhaust air damper has an opposed blade geometry. Each damper has a separate actuator. Airflow measurements were obtained using airflow stations and differential pressure transducers. The airflow stations yield an average velocity pressure for a cross section of the duct. The differential pressure transducers are accurate to ± 0.05 percent of the reading + 0.001 percent of full scale [4].

To evaluate the error associated with the airflow measurements, the differential pressure measurements obtained with the airflow stations were compared with those obtained using a Pitot tube traverse. The recirculation airflow station error ranges from 6 to 7 percent. The error associated with the return airflow station is difficult to determine because the Pitot tube traverse measurements are not reliable. This is probably due to the presence of abrupt changes in duct geometry directly upstream and downstream of the airflow station. Thus, an alternative comparison involving the recirculation and return airflow station measurements was undertaken. This comparison consisted of sealing the exhaust air damper and measuring the recirculation and return airflow rates over a range of damper positions and fan speeds. The mean value of the disagreement in the measurements was 6.9 percent. For all conditions, the recirculation airflow rate was higher than that of the return airflow rate. Thus, in the results that follow, return airflow rates are corrected by multiplying the airflow rate by 1.069. Details of the assessment of measurement errors are available [5].

The exhaust airflow rates for the traditional and new control systems are plotted in Figure 8 as a function of the normalized control signal to the exhaust air damper, where normalized control signals of 0 and 1 indicate that the damper is commanded to the fully closed and fully open positions, respectively. The dashed (traditional) and solid (new) curves in Figure 8 are fourth order polynomials that were fit to the data using least-squares regression. For the experimental results, the supply airflow rate was nearly constant and approximately equal to Q_S = 1.38 m^3/s. The return fan was controlled to maintain the flow difference between the supply and return air ducts at 0.57 m^3/s.

The curves in Figure 8 are very similar to those presented in the simulation results. Although the data for the new control system indicates that airflow through the exhaust air damper becomes negative as the exhaust air damper is closed, no evidence of reverse airflow through the exhaust air damper was observed when smoke tests were performed. Smoke tests did confirm that air is drawn into the AHU when the traditional control system is used and the exhaust air damper approaches the fully closed position.

The experimental results lend credence to the simulation results obtained for the two control systems. The new control system is easy to implement and will reduce the range of conditions for which air can be drawn into the AHU through the exhaust air damper. However, it does not eliminate the possibility of this phenomenon. Thus, it may be necessary to take additional measures if the problem persists.

Conclusions

The objective of this paper was to describe a new VAV AHU control system that helps prevent air from entering an AHU through the exhaust air damper. Simulation and experimental results presented in this paper demonstrate that air can enter an AHU through the exhaust air damper for a traditional control system that links the positions of the outdoor air damper, exhaust air damper, and recirculation air damper.

The new control system links only the positions of the exhaust and recirculation air dampers and was developed for AHU's that use a volume matching control strategy for the return fan. For the new control system, the outdoor air damper is fully open during occupied

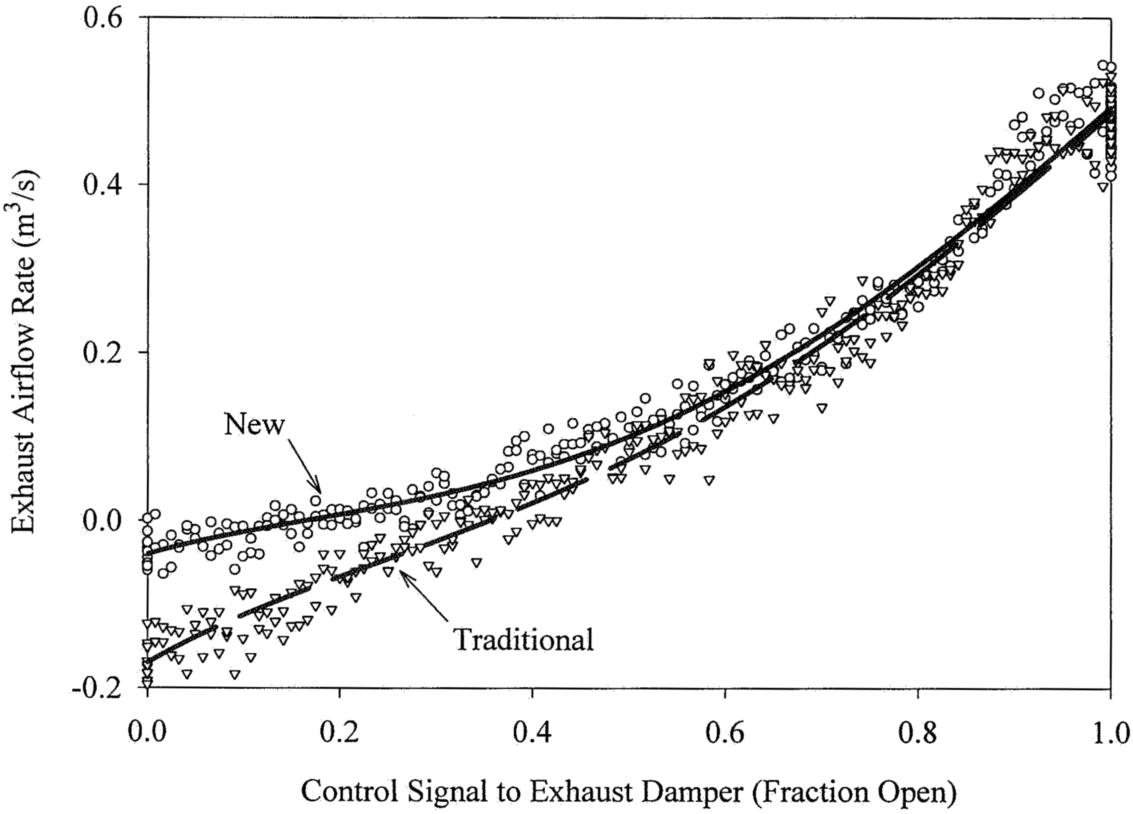

Figure 8 Results from NIST Air-Handling Unit Laboratory.

times. Simulation and experimental results demonstrated that the new control system helps prevent the reverse flow phenomenon, although the potential for the problem is not eliminated. The new control system can fail if the recirculation air damper does not provide a sufficiently large flow resistance. In this situation it is necessary to disable one or more of the recirculation air damper blades or to limit the maximum open position of the damper. Finally, the new control system is easy to implement and will save energy by reducing the fan power.

Acknowledgments

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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The Effects of Building Form on the Natural Ventilation of Commercial Buildings

D.K. Alexander, H.G.Jenkins, P.J.Jones

Welsh School of Architecture
University of Wales, Cardiff
Bute Building,
Cardiff CF1 3AP, U.K.

1 Synopsis

Wind pressures can significantly affect ventilation performance. However often they are overlooked in the design of a naturally ventilated building, with buoyancy forces presumed to offer the worst case scenario for design. The result is that airflow patterns and the ventilation performance of the building is often different from the design intent.

Successful natural ventilation design requires careful consideration of the building form, and so must involve the architect at the early stages of fabric development. Whilst devices and features may be used to enhance the performance of ventilation outlets, the wind flows at inlets are more likely to be affected by the form of the building, and less amenable to improvement by small devices. Performance may be improved by larger features that in effect become part of the building form itself. Therefore the designer requires a greater awareness of the effect of wind on natural ventilation. They may require access to targeted design information and design tools that are as yet generally unavailable.

2 Introduction

As part of a current EPSRC funded investigation, a number of current building designs were studied using wind tunnel and CFD techniques. Each design case study had an explicit intent to naturally ventilate all or part of the interior. The studies focused on the likely performance of the ventilation design with regards to wind driven ventilation. Example building models are shown in figures 1 and 2.

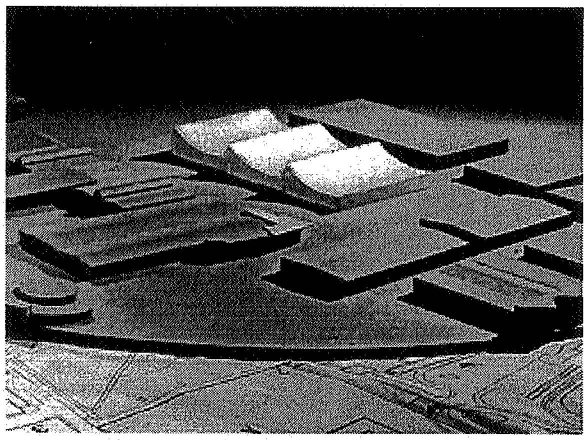

Figure 1 Wind Tunnel Model

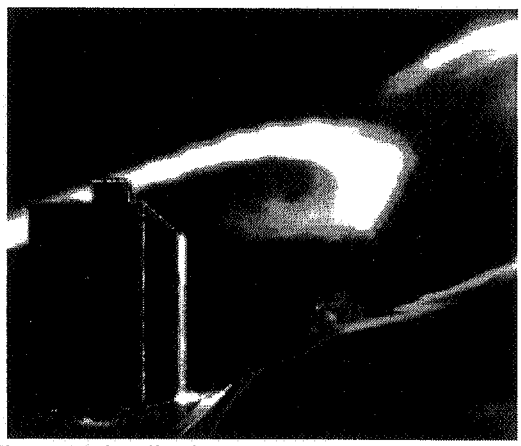

Figure 2 Visualisation of Air flow over a Building

3.1 Wind vs. Stack Effects

It is apparent that many natural ventilation design strategies have been developed with only buoyancy in mind, that is, the design has focused on the provision of low level inlets and high level outlets. This is understandable for two reasons. Firstly, the design intent is often developed using summer and winter design conditions, when temperature difference is robust and can easily be estimated. Secondly, thermal buoyancy and stack effects are well understood, documented, and accepted into design theory.

In contrast, there is often little design intent involving wind conditions. Wind pressures are not felt to be mysterious, but they are considered unpredictable and as a result erroneously ignored. The "no wind" scenario is often considered to be the worst-case for design development. If wind has been considered in the design then it is generally presumed to be beneficial, that is that wind induced ventilation will always be fortuitous, assisting the buoyancy flows. Further detailed design of wind flows, when the subject is taken more seriously, can be crippled by the overall complexity of dealing with the buildings and its surroundings and the lack of appropriate design information and suitable tools.

In most cases wind conditions are important to the performance of the ventilation design. In mid-season conditions, for instance, temperature differences (and therefore stack pressures) may be low. Wind speeds are rarely calm in most locations, and wind pressures generated in summer and winter conditions may rival buoyancy pressures. The example in figure 3 is a graph of wind speed recorded at Cardiff through a warm week in August, presented as an average day. This example shows a clear diurnal pattern in wind speed. During midday, the times of high temperature and highest solar loads (and hence the greatest need for ventilation cooling), the wind speeds are significant and would produce significant ventilation forces.

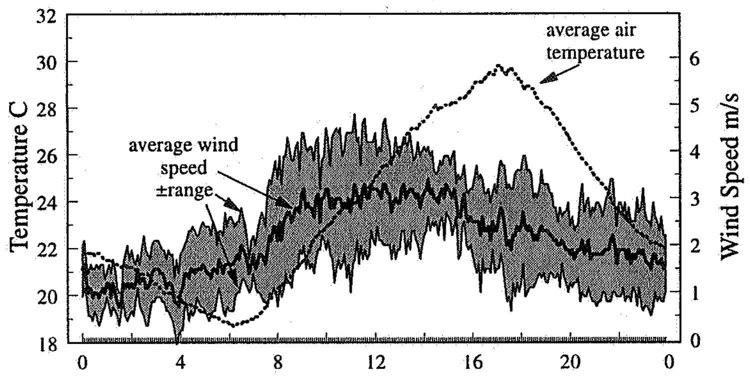

Figure 3, Wind conditions during a warm period in Cardiff

Wind induced forces may not always be beneficial; depending on the nature of the building form and on the location of inlets and outlets, wind pressure differences may oppose the buoyancy pressure gradient. For a range of wind speeds, this may result in a reversal of flows from the design intent. This can lead to comfort, odour or pollution problems. Further, particular wind directions and speeds may lead to the "capping" of outlets, that is, the wind and buoyancy forces may balance, resulting in the general decrease in ventilation. Ultimately this would leave only ventilation driven by turbulence and short term variations in wind velocity, or adventitious ventilation through other openings. This then would constitute the "worst case scenario" for ventilation, as opposed to the "no wind" case.

Buoyancy pressures can be calculated from

$$Ps = H \cdot 3462 \cdot \left[\frac{1}{Ti} - \frac{1}{To}\right] \qquad 1$$

where

Ps is Buoyancy pressure (Pa);

H is height difference between openings (m);

Ti, To are inside and outside temperature respectively (K).

The buoyancy pressure can be estimated as approximately 1 Pa for a moderate temperature difference of 10K and a 3m stack height.

Wind pressures are normally determined from pressure coefficient (Cp) values, where

$$Pw = Cp * 0.5\rho V^2 \qquad 2$$

where

Pw is Wind pressure on surface (Pa);

Cp is surface pressure coefficient,

ρ is air density (kg/m3);

V is air speed (m/s).

Therefore for moderate winds (3 m/s) wind pressures can commonly be in the range -3 to +3 Pa, depending on the orientation of the opening.

Thus wind pressures can easily equalise or dominate buoyancy pressures at quite reasonable wind speeds. The term "Cp gradient" can be used to describe the difference in Cp between proposed inlet and outlet areas, and can also be used in the above equation to estimate the wind driven pressure across two openings. A negative Cp gradient implies the wind pressures oppose the buoyancy pressures. For the example moderate conditions as used above (10K temperature difference, 3m stack height, 3 m/s wind), a Cp gradient of only -0.2 would cancel the buoyancy pressures, leaving minimal ventilation. Such adverse Cp gradients have been found in practical designs.

3.2 Inlets and Outlets

The successful natural ventilation design requires the identification of suitable areas for inlets and outlets, robust against different wind speeds and directions. It may be straightforward to develop a strategy for one wind direction, allowing the inlets on the upwind face and the outlets on the downwind face or roof, but rarely will such a simple site be found; the wind comes from different directions, though the course of a year, season or even a day. The wind flows will be influenced by the building as well as by its' surroundings.

Vertical or near-vertical facades are particularly problematic. Identified as inlets or outlets for one direction they can easily reverse flows for even small shifts in wind direction. In addition, high level openings on a windward facade naturally have higher wind pressures than the equivalent low level openings. This is an adverse gradient as described above. An example of such a system is provided in figure 4a.

Openings on vertical faces, when they are to act as outlets, are amenable to improvement by the addition of shields or deflectors, modifying the local flow so that direct impact is avoided. This can substantially alter the pressure regime, as shown figure 4b, wherein a simple plate baffle has been placed in front of the upper opening. This sort of feature can be used to "tune" the airflow patterns generated by the building form.

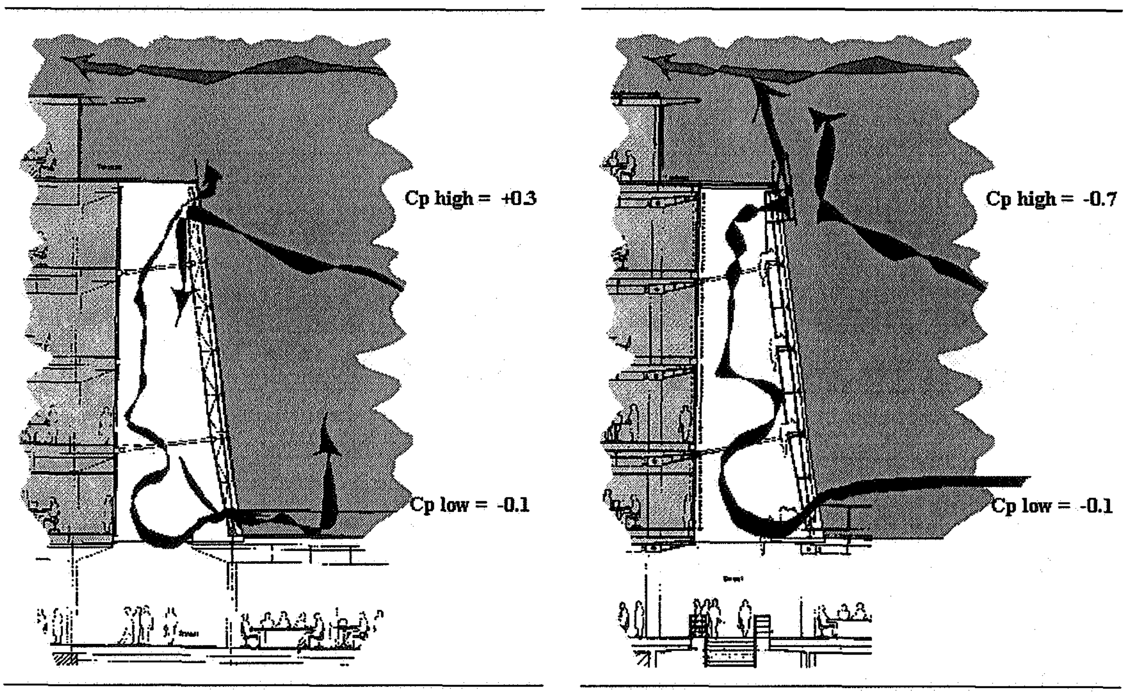

Figure 4, Ventilation of a simple atrium, before and after a wind baffle is placed over the outlet positions

In the above case the scheme was fortunate in having a relatively consistent prevailing wind direction. However the shielding concept can be used to reduces directionality for a feature, such as an extract tower; an example is the "H-pot" stack termination device which can successfully increase suction pressure for a wide range of wind conditions, as illustrated in figure 5 .

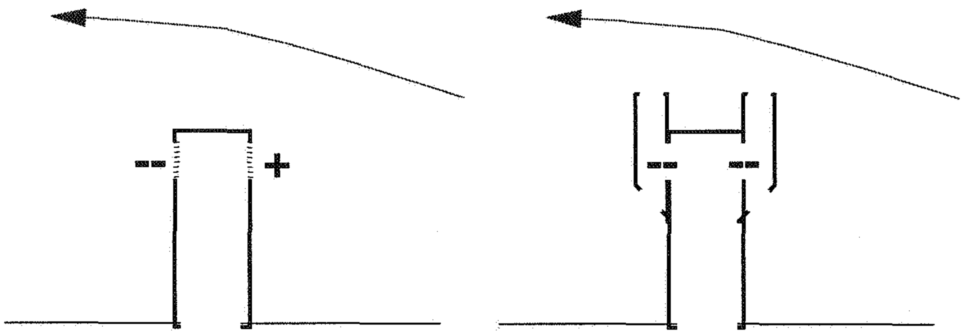

Figure 5 "H-pot" termination increases suction pressure, pressure at outlet denoted by - (suction) or + (positive pressure)

Shielding devices are generally appropriate only for outlet areas. Inlet areas appear to be less susceptible to improvement (that is in relatively increasing local air pressure) by the addition of simple devices. The air flow near inlets is largely determined by the bulk of building and its' surroundings. Often inlets are positioned at low levels (perhaps as dictated by buoyancy considerations) and may be in the wake of nearby structures for some wind directions. In general the small pressures involved in natural ventilation constrain inlets and outlets to be as close as possible to the spaces they will serve, so as to avoid high duct pressure losses. As illustrated in figure 6, the wake can disrupt the design flows, defeating even a well chosen extract device.

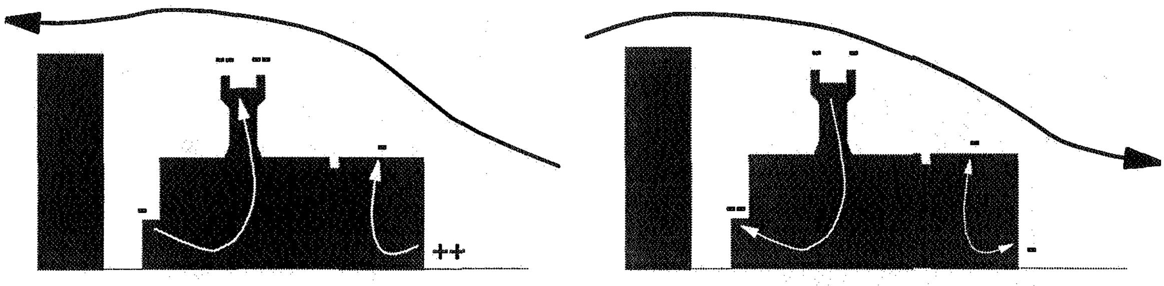

Figure 6, The effect of building wake on ventilation flows

The disrupting wake need not be caused by nearby buildings, but may be caused by the building itself. A common natural ventilation design feature is the repetition of a device as a building module. However a device, developed in isolation and perhaps ideal for greenfield site, may have its' potential degraded when used as a module, with each device causing disruption and wind shadowing for each downstream neighbour, as illustrated in figure 7.

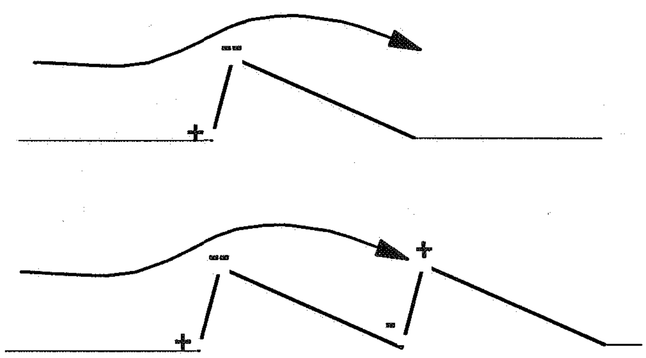

Figure 7 Effect of upstream device on downstream neighbour.

4 The Building Designer as Wind Engineer

It has become apparent that the form of the building (and its' immediate surroundings) can have a greater effect on the likely performance of the ventilation system than what may be termed devices (that is stacks, wind catchers, fins, wings, and cowlings). Outlet devices may be to some extent "tuned" to improve their performance under varying wind conditions. However due to the difficulty in affecting the pressure near inlet areas, ventilation problems posed by the building form may not be correctable by "bolt-on" devices selected from a natural ventilation "catalogue".

The form of the building largely determines the wind flow characteristics around the building and consequently the patterns of surface pressures. Small features cannot disturb these gross trends until they themselves become of significant size and so effectively become part of the building form itself.

Devices intending to grossly modify the pressures and flows near the building need to be large compared to the building before a substantial effect may be seen. These large devices effectively become part of the building form and become architectural features. Figure 8 shows a student design project lead by such principles, and illustrates the scale of the devices under consideration.

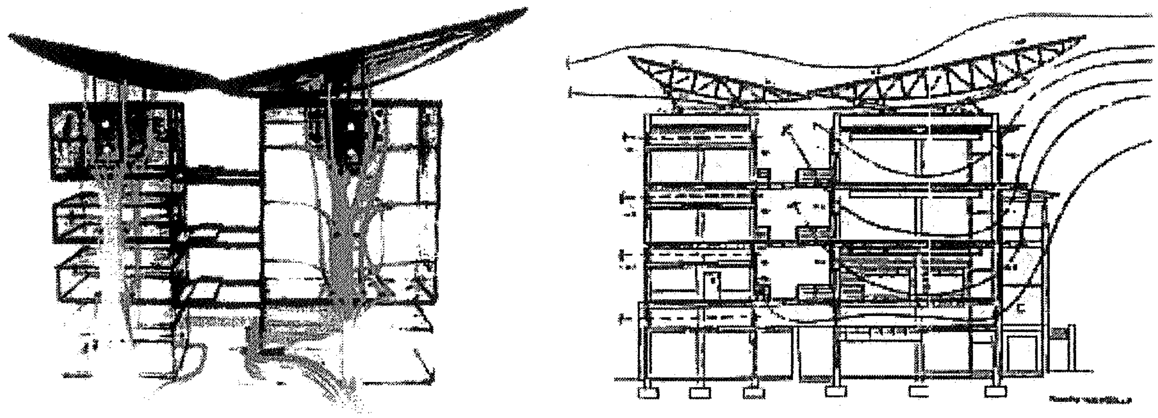

Figure 8, Student design project for wind enhanced ventilation of an office block

The control of the building form lies in the hands of the architect, and as such the architect becomes the prime designer of the ventilation system. Where the form of the building may be tempered by other criteria, such as cost or noise control, or a desire to capture solar energy, the problem lies outside the competence of most architects.

There is currently little information available aimed at the architect; idealised flow patterns contained in handbooks and primers offer little useful information and often serve to reinforce a concept of a rigid stack flow or wind direction. The lack of design information often means ventilation design is passed on to engineers or consultants, by which time the form may have been set.

Numerical information and calculation tools developed for natural ventilation exist, but are still in a format suited for use by engineers. As such they are often "unfriendly" to the architect, even to the ever emerging generations of increasingly computer literate and numerate designers. Appropriate design and information tools are needed to allow early design choices to be made.

5 Summary

Wind tunnel scale modelling can provide an important input into the early design process, however as yet they are inaccessible to many designers.

Wind pressures can significantly affect the ventilation performance of a building, altering flow paths in relation to the design intent and potentially detracting from the overall ventilation performance when in conflict with a buoyancy driven design.

Successful natural ventilation design requires the consideration of the building form, and so must involve the architect at the early stages of fabric development. While devices and features may be used to enhance the performance of outlets, the wind flows at inlets are more likely to be affected by the form of the building, and less amenable to improvement by small devices. Performance may be improved by larger features that in effect become part of the building form.

Therefore the designer requires a greater awareness of natural ventilation principles, and requires access to design information and design tools targeted to their professions, that are as yet generally unavailable.

Acknowledgements

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

17th AIVC Conference, Gothenburg, Sweden, 17-20 September, 1996

NOVEL METHODS OF INDUCING AIR FLOWS WITHIN BUILDINGS

D P Campbell, R S Webb

Department of Building Engineering & Surveying, Heriot-Watt University, Edinburgh, EH14 4AS, UK

Novel methods of inducing air flows within buildings

by: Dr. D. P. Campbell
R.S. Webb,
Dept of Building Engineering & Surveying,
Heriot-Watt University,
Edinburgh, EH14 4AS, U.K.

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Synopsis

Water use is distributed throughout building structures. Energy used to pump the water to higher levels in the building is not currently recovered, and is dissipated by performing work on air in the ventilation system which is vented to the atmosphere, when the water is discharged into the drainage stack. This energy can be utilised productively, however, by strategically placing the air inlet for the drainage stack inside the building, thereby utilising the potential energy stored in the water to draw air through the building. Airflow induction by falling water films is a well known problem in the drainage industry, and airflow rates of 10-20 times the water flow rate are common. Basic analysis of the ventilation requirements for typical large buildings suggest that this may contribute between 1 and 7.5% of the total ventilation requirement. Air admittance valves would be effective at isolating the habitable space from the waste water.

1 Introduction

The increasing cost of energy and raw materials, and the desire to maximise investment value, are exacting an increasing demand on services engineers to produce energy efficient buildings. In addition to this, the increased public awareness of energy management and so-called 'green issues' are generating the driving forces for designers and engineers alike to produce new or alternative solutions to traditional servicing problems. These forces are also encouraging the development of new technologies or the utilisation of existing technologies and techniques which, up until now, have been ignored, unseen or simply considered uneconomical, and to use them in new ways.

This paper considers the application of existing technology in a new way to assist in the movement of air for ventilation in large buildings. The technique itself relies on the potential energy of water raised to the top of large buildings. When the water is discharged into the drainage stack after use, the potential energy is presently dissipated unproductively by allowing it to induce an air flow within the stack. Drainage systems are designed to optimize the resulting air flows generated, but it is possible to isolate certain attributes of drainage systems which can be exploited to maximise air flow induction rather than optimize it. If properly integrated within a building, such a system may be able to generate sufficient air flow within certain building designs, and with certain usage patterns, that it becomes significant in terms of the contribution made to passive ventilation.

2 Theoretical Background

The prevention of excessive air flow induction by water flowing within building drainage systems has been successfully achieved following intensive research over many decades. The mechanisms governing the induction process are well understood and predictable within the scope of standard drainage fittings. The mechanisms of air movement will be examined within this context, and then applied to the role of ventilation by natural air movement in buildings.

2.1 Water flows in drainage systems

Within a drainage network the ability to induce air flows belongs to the applied water flow, and the efficiency of air movement is heavily influenced by the water film thickness and velocity. In a circular pipe, water quickly forms an annulus as illustrated in Figure 1. Other workers have investigated the development of the annular flow pattern (Pink, 1973[1,2]) and the velocity and thickness of the annulus has been accurately measured (Wise, 1986)(Swaffield and Thancanamootoo, 1989). When the free surface flow in the branch reaches the stack junction, the horizontal velocity component enables the flow to reach the opposite wall of the stack quickly, where the flow separates and flows round the vertical stack with a high degree of swirl. Annular flow is quickly established (typically within 1 metre) but, until the flow becomes annular, elliptical air spaces will intermittently form on either side of and below the branch jet. These constitute a curving path, joining the air in the stack above the branch to the central air core below the branch, the area of which is influenced by the geometry of the junction. The area of the elliptical spaces available for air transport is obviously also dependant on discharge rate, which in turn is dependant on the nature of the appliance(s) discharging and the

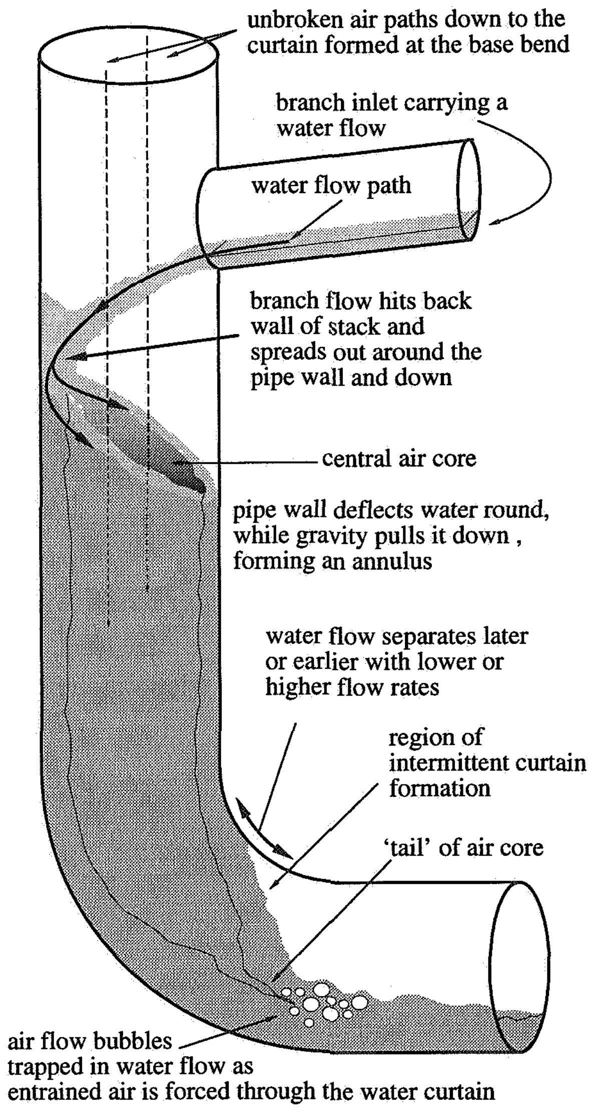

Figure 1 Formation of annular water flow with an entrained air flow at the core, when a branch discharges into a vertical stack

diameter ratio of the branch to the stack. Narrow, fast flows with high slopes give rise to larger spaces than wide, slow and horizontal discharges. In any event, the flow quickly forms as a descending annulus around a central air core, with the motive force that the water film exerts on the air core being applied along the wetted perimeter of the water/air interface.

As the water in the stack moves down, the velocity changes under the influence of gravity and friction with the stack walls. Eventually the friction force equals the gravity force and the water velocity becomes constant. Figure 2 illustrates this condition for uniform and steady conditions, in which it is assumed that the initial lateral swirl velocity components have decayed to zero. From Figure 2, in which the shear stress between the water and the air is neglected, and from Newtons Second Law of Motion,

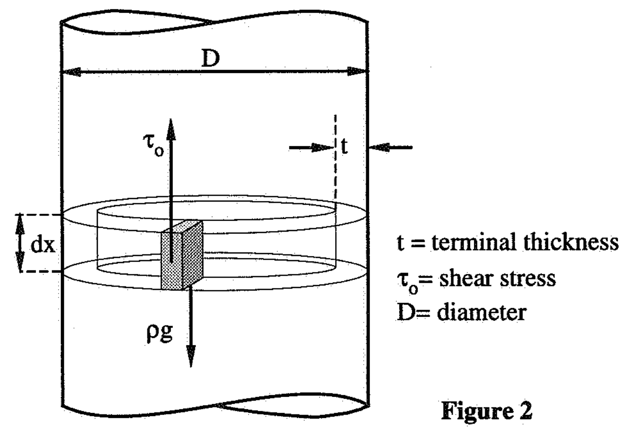

Figure 2

Forces acting on a toroidal flow element in a developed annular stack water flow. Shear stress acts at the pipe/water interface and negates the gravity force when terminal velocity is reached. Forces shown acting on a segment of the torrus for clarity.

$$\rho g(\pi D\, t\, dx) - \tau_0 \pi D\, dx = \rho\, \pi D\, t\, dx \frac{dv_w}{dt} \qquad (1)$$

dividing through by $\pi D dx$ gives

$$\rho\, g\, t - \tau_0 = \rho\, t \frac{dv_w}{dt}.$$

At the terminal velocity of the water flow, the acceleration term becomes zero, thus

$$\tau_o = \rho g t \qquad (2)$$

The shear stress of the water on the wall is given by

$$\tau_o = \tfrac{1}{2}\rho f v_w^2 \qquad (3)$$

where f is an appropriate friction factor. Substituting equation (3) into equation (2) and rearranging for t gives:

$$t = \frac{f}{2g} v_w^2 \qquad (4)$$

Rearranging equation (2.4) again, this time for v_w gives

$$v_w = \sqrt{\frac{2gt}{f}} \qquad (5)$$

Equations (4) and (5) demonstrate that a relationship exists between water film terminal thickness and the terminal water flow velocity. This relationship can be developed to yield an expression relating terminal water velocity to stack diameter and volumetric water flow rate (Swaffield and Galowin, 1992):

$$v_t = \left(\frac{Q_w}{D}\right)^{0.4} \qquad (6)$$

which in turn demonstrates that the terminal water velocity and film thickness are related to both stack diameter and water flow rate. Figure 3 shows the variation in water film thickness (at the terminal water velocity) for flow rates up to 4 l/s for 75, 100 and 150mm diameter smooth stacks (Swaffield & Galowin, 1992). Figure 4 shows the percentage change in central air core wetted perimeter, i.e. after the thicknesses of the respective annular water films have been subtracted, for the three pipes. Figure 5 shows the variation in terminal water velocity for the three pipes under the same conditions, and Figure 6 shows the percentage change in the water velocities. From Figures 4 and 6, the variation in air core wetted perimeter is very small for water flow rates of between 1 and 4.5 l/s, whereas the variation in water velocity is very large over the same range of flow rates.

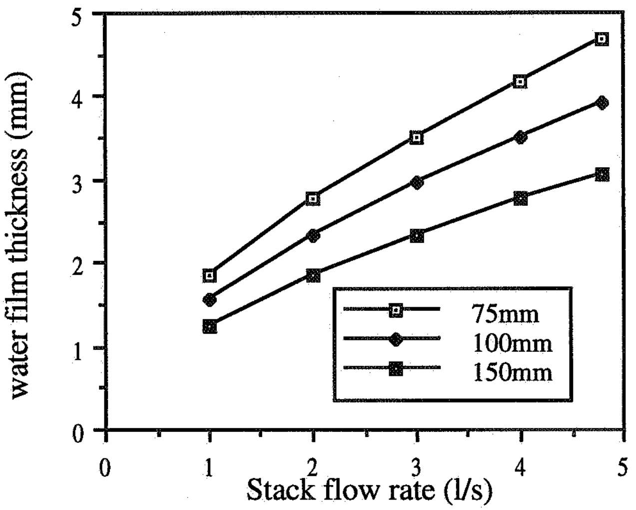

Figure 3. Water film thickness and flow rate

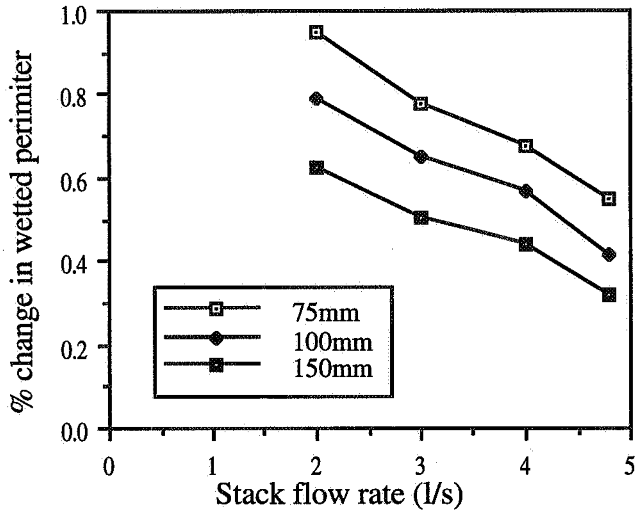

Figure 4. % change in wetted perimiter

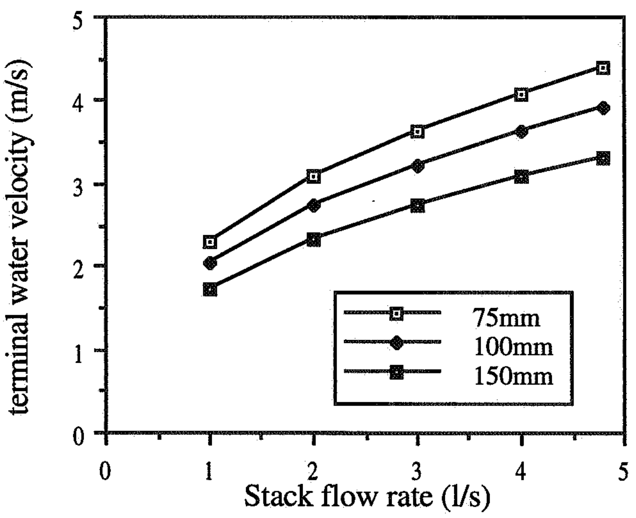

Figure 5. Terminal water velocity and flow rates

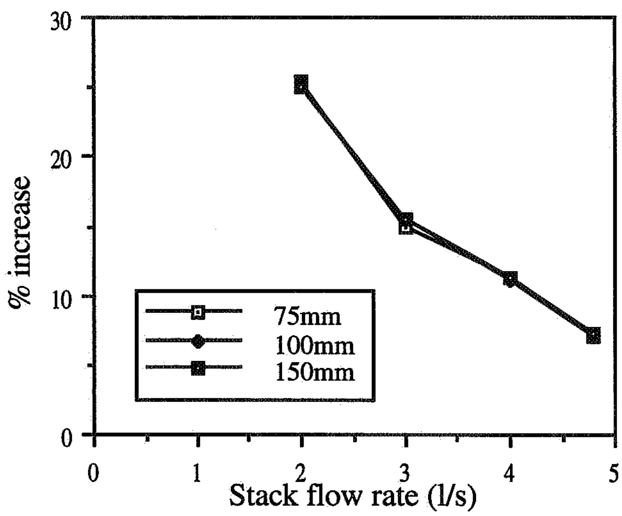

Figure 6. % change in water velocity with air flow

For typical smooth drainage pipes, such as UPVC plastic or glass, stack limiting flows are defined so that the water film thickness of an annular flow which has attained its terminal velocity is never greater than $\frac{1}{16}$D, i.e. $\frac{1}{4}$ of the stack area. The vertical distance H required for the water film to reach 99% of the terminal velocity v_t is $H = 0.159v_t^2$ (Swaffield & Gallowin, 1992). For vertical stack water flow velocities of about 4 m/s (about 4.5 l/s), this suggests that v_t occurs at about 2.5m from branch entry. It should be noted that the equations in this Section for determining water velocity are based on flow rate and pipe area and, in the absence of any correction factors, will give mean velocities for the water film. In fact, a gradient across a section of the water film will exist, from zero at the water/pipe interface, to the mean velocity somewhere inside the flow element, through to some higher value at the inside (air column) surface of the water annulus. Although the inside surface of the water annulus is not smooth, very little water, in the form of spray or droplets, appears in the central air column (Pink, 1973[1]). It is to be expected that air velocities will be higher than those calculated based on mean water film velocities as the applied shear, based on the air/water interface velocity, is underestimated.

This Section has illustrated the relationship between water film thickness and water velocity in the stack, and the dependance of those flow parameters on stack diameter. Clearly, the objective of this study will be to determine the best geometry of duct or pipe for a given water flow rate for achieving maximum air flow rates.

2.2 Cause of air movement in the stack

In the 'wet' stack length, air movement is induced by interface shear between the inside surface of the descending water annulus and the air core. This force can only be exerted along the wetted perimeter of the air core and is therefore proportional to its length as well as the velocity of the water surface (not the water mean velocity). At the air/water interface, the velocity of air and water are equal. In the 'dry' upper stack, above any active discharging branches, air is drawn in from outside to sustain the air movement and a pressure drop will develop as the air encounters entry and frictional losses.

It has been pointed out that a velocity profile existed across a water flow element, and that calculated velocities were mean values. Assuming that the velocity profile is linear, the velocity of the water at the air/water interface will be double the calculated mean velocity. The velocity profile is unlikely to be linear, however. The motive force applied to the air by the water may be expressed as:

$$F = \tau_i P_{AC} \Delta L \text{ (Swaffield and Galowin, 1992)}$$

where the wetted perimeter of the central air core P_{AC} is $\pi(D\text{-}2t)$ and τ_i is the shear force, which is dependant on the square of the air velocity. From Figures 4 and 6, the shear force F is influenced by two parameters:

(a) wetted perimeter of the air core, whose rate of decrease in length is decreasing by fractions of % as flow rate increases, Figure 4.

(b) water velocity, whose rate of increase is decreasing by tens of % as water flow rate increases, Figure 6

Thus the contribution of the water velocity is easily the dominant term in determining the motive force that the air core experiences for any given diameter of stack.

The overall effect on air transport in the stack is that as the flow increases towards conditions that promote full bore flow, the rate of air transport will begin to decrease and this is supported by experimental observations. This can be explained by appreciating that an increase in water flow rates will bring about a reduction in the area of the central air core, as the water film thickens. This will tend to reduce the air carrying capacity of the central air column, and cause the motive force due to the increasing water flow rate to peak and then decrease. Maximum air flow rates currently obtained with circular cross section pipes are of the order of 10-15 times the water flow rate (Campbell, 1993).

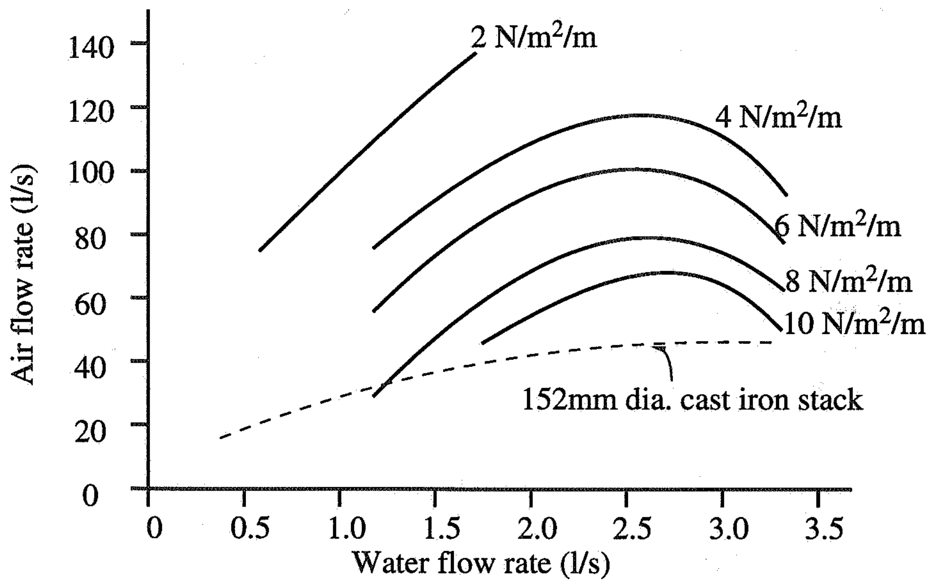

Figure 7 Form of the relationship and typical values for entrained air flow rate variation with water flowrate and pressure gradient in the 'wet' stack length,

3 Analysis

3.1 Water usage in Buildings

Water usage rates for large buildings are easily measured by water meters. Predictions of water consumption rates are more problematic and depend on statistical methods for estimating water demand based on probability and the discharge unit method, in which appliances are assigned arbitrary discharge unit values. Standardised d.u./Q curves allow engineers to convert from estimated demand units back to a normalised, steady state flow rate, which approximates to the averaged long-term demands of the system. These demand estimations are gross over-simplifications because they provide only steady-state estimates of what the continuous flow rate would be, when averaged over a suitably long period. As a guide, CIBSE (1986) guidelines on sizing new installations suggest that water demand will lie somewhere between 45 (office) and 140 (hotel) litres per person per day (l.p.d.), although many usage patterns can be higher. Assuming a flow rate of 100 l.p.d., and a population of 500 people, occupying the building during office hours, gives $500 \times 100/7/60/60 = 2 l s^{-1}$.

It has been shown that water velocity is the most important factor in determining the volume of air transported. Maximum water velocities achievable under gravity for a flow of 2l/s are in the region of 2.2 m/s and, under these maximum velocity conditions, Equation 4 predicts the water film thickness to be around 2mm. At 2.5 m/s and 2l/s, a water film of 2mm thickness would have a length of 0.002/(2.5*0.002) = 40cm. In order to maximise the water velocity, friction with the pipe or duct wall must be minimised, suggesting a circular cross section. A circular cross section would also maximise the volume of the air core, further increasing air movement efficiency. The fact that the wetted area of the air core is minimised by this design is not important as the lengths involved will maintain an adequate frictional driving force on the air. From Figure 3, the airflow expected from these conditions will be around 30 l/s.

3.2 Ventilation Requirements

Ventilation requirements vary according to building occupancy and usage profiles. Typical estimates from CIBSE guides give 5l/s per person or 0.8l/s per square meter of floor area. Using these values, and assuming a floor area of 500 square meters, suggests that 30l/s from waste water sources would provide somewhere between 1 and 7.5% of the total ventilation requirement for the hypothetical building, depending on usage patterns.

3.3 Design & Performance of Waste Water System

The data used in this analysis have included water from all sources, i.e. both grey and black. Clearly, there will be a requirement to isolate the habitable space from the effect of this waste and the current state of the drainage industry will set a more than adequate standard. The use of air admittance valves as a means of allowing air into a drainage system, and not out, has been accepted throughout Europe (with the exception of the U.K.) and the U.S.. The simple provision of one of these devices at the head of the stack, strategically placed or vented to a suitable space, would therefore be expected to draw 30 l/s of air out of the space and down the stack, while preventing the possibility of odour or pathogen escape. Air admittance valves have been found to present little extra entry loss to air movement, and are silent and dependable in operation.

If incorporated into the design of a new build, there will be very little extra cost incurred, if any. Diverting the stack head to a suitable location in existing buildings will cost very little, even if a wall has to be drilled.

Conclusions

Airflow induction caused by water falling in circular pipes can be understood and predicted by the application of basic fluid mechanics principles. The rate of airflow induction depends principally on the water velocity but also on the pipe diameter. The optimum shape for maximum airflow induction is circular. It may be possible to supplement the ventilation in large buildings by utilising the airflows set up within building drainage ventilation networks.

Typical large building forms, with either 500 occupants or 500 square meters of floor area, produce airflow rates in the drainage ventilation networks which amount to 1 and 7.5% of the recommended ventilation requirements, respectively. A usage pattern study is recommended to find the optimum building plan for use in this way.

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AIR LOCKS. EFFECTIVENESS AND DESIGN RULES

Hans Phaff

TNO Bouw BBI, P O Box 49, NL 2600 AA Delft, The Netherlands

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Optimum Ventilation and Air Flow Control in Buildings

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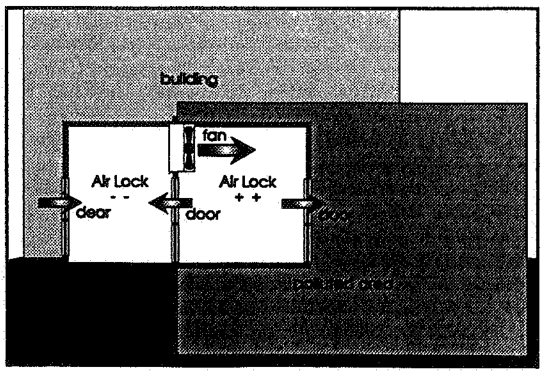
Title	Air Locks. Effectiveness and design rules.
Author	Hans Phaff
Address	TNO Bouw BBI ; PO Box 49; NL 2600 AA; Delft; The Netherlands.
Tel	+31 15 2608 462
Fax	+31 15 2608 432

Abstract.

To guard one space in a building from the air borne pollutants in an adjacent zone Air Locks are widely used. A comparison of different approaches has been made with a ventilation model that predicts the dynamical coarse of concentration.
Dependency of the frequency of persons passing the doors, the number of doors, the ventilation conditions, etc. is shown.
From the wide number of possible variants of air locks the best are selected.
Selection criteria and Design rules are proposed.



OPTIMUM VENTILATION AND AIR FLOW
CONTROL IN BUILDINGS

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SUMMER COOLING FOR OFFICE-TYPE BUILDINGS BY NIGHT VENTILATION

M Kolokotroni, B C Webb, S D Hayes

Building Research Establishment, Garston, Watford, Herts, WD2 7JR, UK
Tel: +44 1923 894040 Fax: +44 1923 664095

SYNOPSIS

The suitability of night ventilation for cooling for the UK is first assessed by presenting plots of summer weather data on the bioclimatic chart for three locations within the country. These indicate that most of the external weather conditions lie within the thermal mass and ventilation effectiveness areas of the charts. To confirm this, thermal simulations of a typical office module under a variety of internal conditions and summer weather data were performed. Predictions have shown that internal temperatures can be maintained below the external values for solar and internal gains of up to about 50W/m². Field measurements in a refurbished natural ventilated office have confirmed that temperatures in night ventilated spaces are generally lower during the following day, especially during the early hours of the working day. Finally, the development of a pre-design tool in the context of IEA Annex 28 on 'Low Energy Cooling Systems' is discussed. The main aim of the tool is to increase the awareness of designers for the energy benefits and the range of parameters for the application of night ventilation as the first means of cooling so that the need for artificial cooling is minimised or avoided altogether.

1. INTRODUCTION

Of the many techniques available to cool buildings, Night ventilation is an appropriate strategy for office buildings in the UK. This is mainly due to the relatively low peak air temperatures which occur during the day during the cooling season and the coincident medium to large diurnal temperature range. Such a combination allows the thermal mass of the building to use the cool night air to discard the heat absorbed during the day.

An initial examination of the weather conditions experienced during the summer months of June to September in the UK indicates that most peak conditions of external weather fall within the ventilation and thermal mass edge of the bioclimatic chart [1, 2]. To illustrate this, the percentage frequency of combination of hourly dry bulb and wet bulb temperatures [3] for London, Manchester (north England) and Plymouth (south west England) have been plotted on the bioclimatic chart. Figure 1 shows that the summer (June to September) climatic envelope is within the heating, comfort, thermal mass and ventilation effectiveness areas of the chart. Although some hours fall within the humid area, these occur at night and the increase in temperature during the day brings the conditions within the comfort zone.

2. MODELLING NIGHT COOLING

To reconfirm the suitability of night ventilation cooling for UK office-type buildings, thermal simulations were carried out using the computer model APACHE [4] taking into account typical construction of office modules, occupancy patterns and other internal heat gains. A parametric analysis was carried out to examine the effect of a number of variables such as orientation and solar gains, internal gains and occupancy patterns, hourly external temperatures and infiltration, day and night ventilation rates as well as simple control strategies to avoid overcooling. Figure 2 presents the result of four simulations during the cooling period of June to September. Internal gains of 20 and 40 W/m² were assumed with and without night cooling with an external maximum temperature of 28.5°C. The example in Figure 2 shows that for day and night ventilation of 4 air changes per hour (ach) internal temperatures are reduced by about 1° to 2°C compared to the cases with no night ventilation.

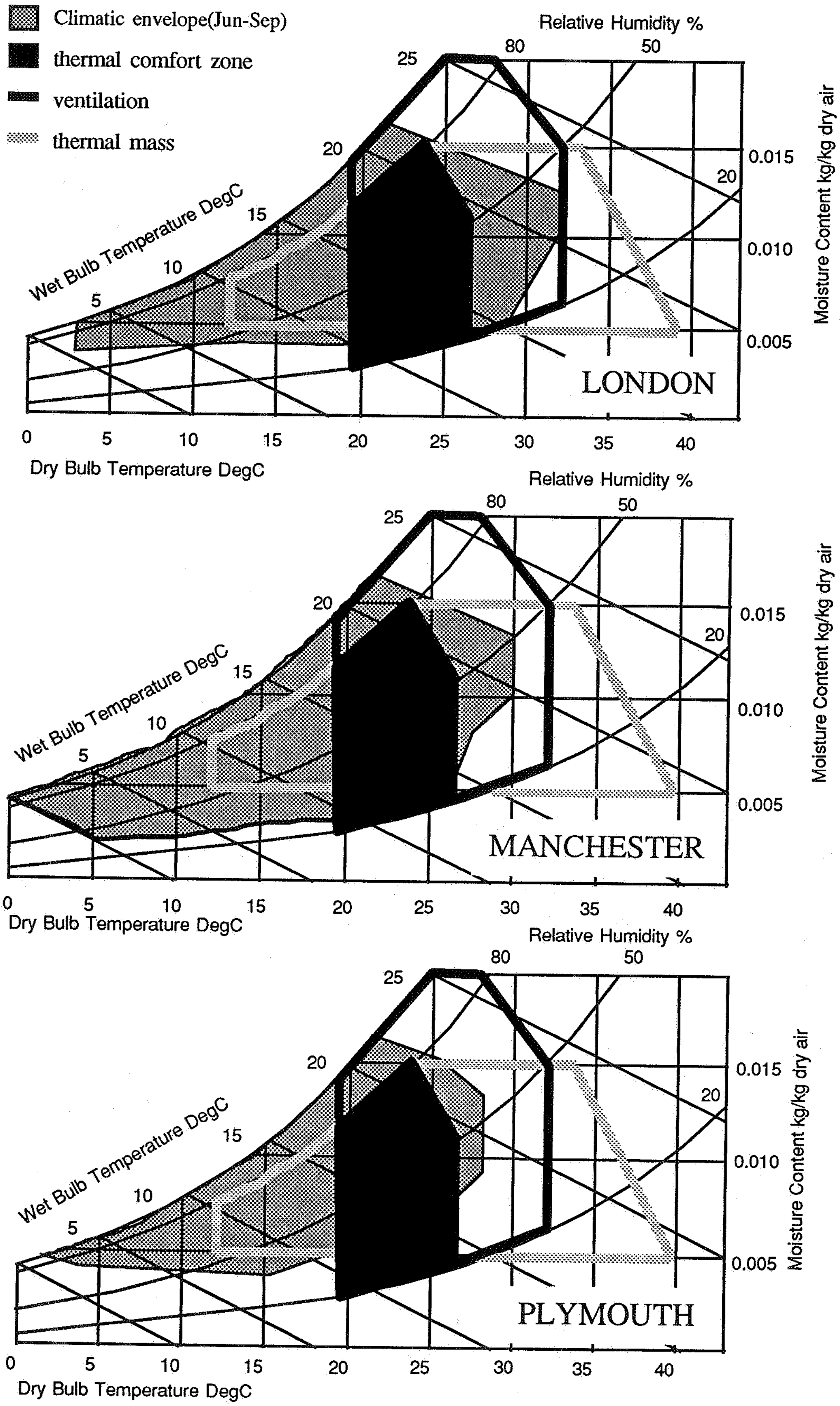

Figure 1: Bioclimatic chart with summer climatic envelopes.

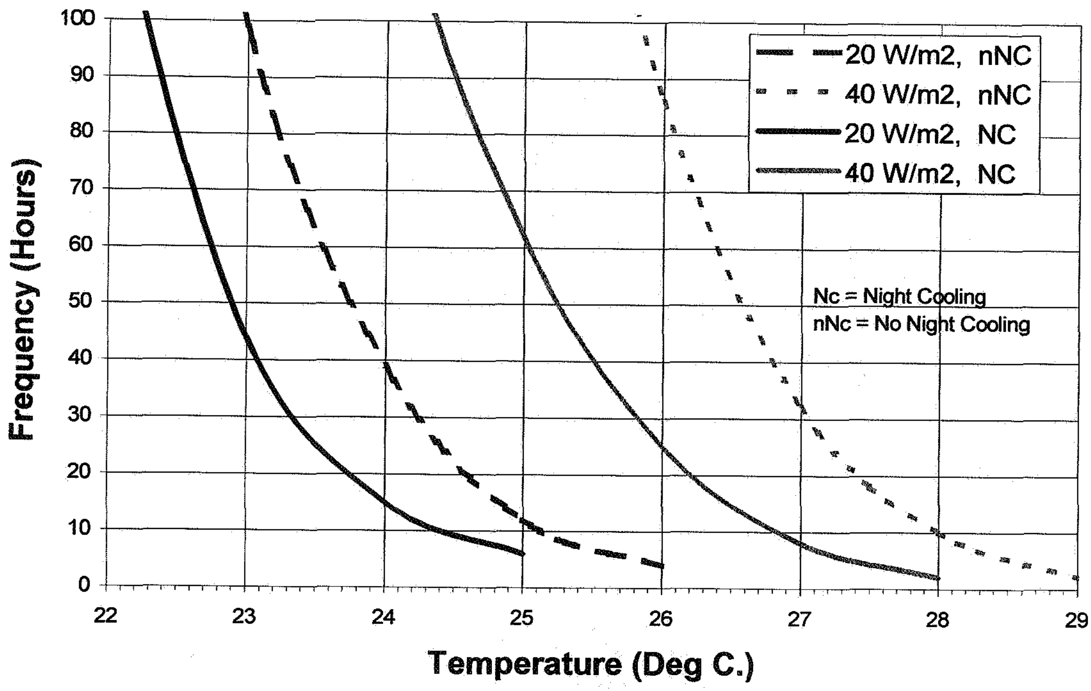

Figure 2: Internal temperature frequency distribution of a typical office module with two levels of internal + solar gains. Day ventilation and night ventilation were 4 ach.

3. MONITORING NIGHT COOLING

A number of recently built innovative buildings in the UK utilising night ventilation for cooling have given encouraging results [5]. However, the potential also exists in older buildings due for refurbishment. Such a building was monitored during the summer of 1995, over a four week period, and the results reported [6]. It follows a summary of the monitoring results in this building for completeness.

The monitored building is a typical 1950's, four-storey office building in west England which had its curtain walling replaced. The project team wanted it to remain a naturally ventilated building, with the users given the option of night cooling. It was assumed that the 'coolth' would be stored in the building mass, i.e. exposed concrete columns, outside wall structure and furnishings. Ventilation was provided by 850 x '600mm bottom-hung ventilators installed around the perimeter of the building. In front of the ventilators was a mesh screen and perforated external louvres, providing 24 hour security and weather protection.

A large open plan office, spanning the width of the building, on floors 1 and 3 was chosen for the study. The offices varied slightly in shape, number of ventilators and amount of glazing. Both offices were cross-ventilated. During each week, Monday to Friday, between 18.00 and 08.00, the ventilators were kept closed on one floor and open on the other.

Floor 3 vents open, Floor 1 vents closed.

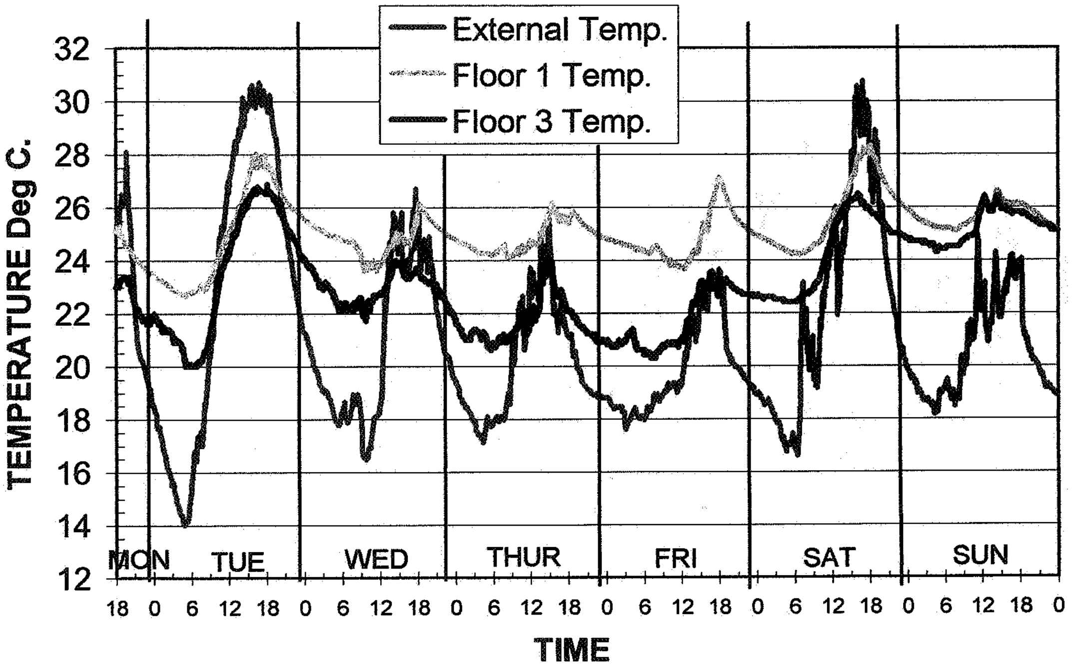

Figure 3: Measured external and internal temperatures in a night cooled and a control office.

At weekends, the ventilators on both floors were either closed or open. For subsequent weeks, and weekends, the conditions were reversed. During the monitoring period the weather was very hot with the external temperature exceeding 30°C on seven days. The maximum external temperature occurred quite often during late afternoon (i.e. 17.00 to 18.00), near to the time when the ventilators would usually be opened (or closed) for the night. If automatic controls were available the ventilators would be opened at an earlier or later time depending on the external temperature so as to maximise the cooling effect.

Figure 3 shows the results of a week, when overnight, from Monday to Friday, the ventilators were kept open on the third floor and closed on the first. The graph compares the mean dry resultant temperatures on both floors with the external temperature. Night cooling is clearly evident on floor 3 with its lower overnight temperatures, and more importantly, the daytime temperatures which remained below those on floor 1 for the majority of the following working day. The ventilators were closed on the weekend days. Figure 3 shows that internal temperatures had reached similar values on both floors by Sunday.

It is easier to compare the night cooling effect between the two floors by determining the difference in the internal mean dry resultant temperatures. Figure 4 shows a plot of the results of two weeks, when the ventilators were closed on floor 1, and open on floor 3. This graph shows that opening the ventilators can reduce the internal temperature overnight, and at the start of the next working day by up to 4°C. However, on some days, towards midday, the temperature on both floors were similar.

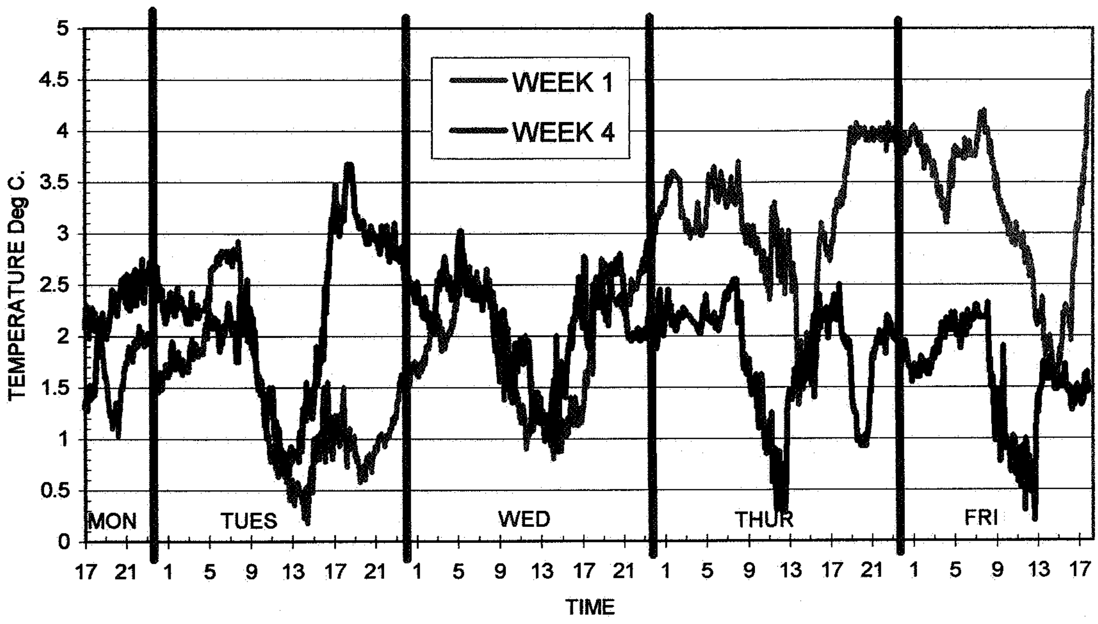

Figure 4: Temperature excess on floor 1 for two weeks. Vents were closed overnight on floor 1 and open on floor 3.

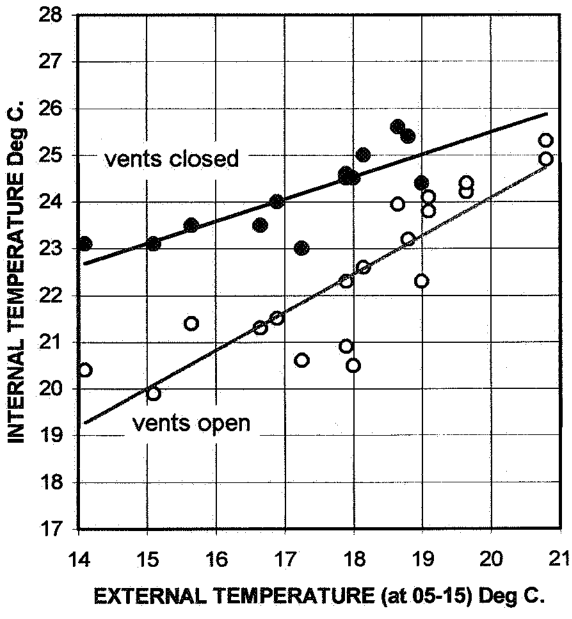

Figure 5: Internal temperatures at 08-00 against external temperature at 5-15 in a night cooled and a control office.

The internal temperature at the start of the working day, 08.00, will give an indication of how effective night cooling was the previous night. It will also have a bearing on peoples comfort level at the start of the day. Figure 5 shows a plot of the internal temperatures, at 08.00, against the external temperature at 05.15 (sunrise, usually the coolest time of the night). The graph shows a linear trend of the internal temperature with external temperature on both floors. The vertical difference between the two lines indicates the night cooling benefit and the degree to which this diminishes at higher external temperatures. At 15°C night temperature, the difference in internal temperature in the morning is 3°C, while at 19°C it reduces to about 1.5°C.

4. DEVELOPING A PRE-DESIGN TOOL

All the interrelated parameters affecting the effectiveness of night ventilation makes it a complex prediction process. Therefore it is necessary for designers to have access to a simple user-friendly and yet accurate model when they are assessing the viability of night ventilation during the initial design stage. We have identified that for such a model, the key predicted output parameters of interest to designers would be:

- maximum dry resultant temperature during the occupied period
- dry resultant temperature at the start of the occupied period
- potential energy savings

Such a tool in the form of design curves has been developed as part of a programme of work within IEA Annex 28 on Low Energy Cooling Systems. The tool is structured as a step-by-step procedure:

- heat gains (Solar + Internal + Day fan pick-up)
- internal / ambient temperature differential for day ventilation only
- night fan pick-up (mechanical ventilation)
- internal / ambient temperature reduction with night ventilation
- peak temperature and frequency distribution
- night cooling
- fan energy consumption (mechanical ventilation)

For night cooling of commercial buildings in the UK, data for the design curves has been generated using a weather year (based on Heathrow weather data) characterised by a peak dry bulb temperature of 28.5°C and suitable for the south-east of the UK . A thermal simulation computer model was used to obtain the data to calculate the curves.

The building model is based on a typical cellular office and positioned in the middle of a row of offices on the middle floor of a typical office block. A thermally lightweight and thermally heavy weight construction were simulated as extremes for creating the curves. Simulation runs included the summer months of June to September. Occupancy is assumed between 8.00 and 18.00 hrs during weekdays only; during this time, day ventilation is operated. Night ventilation is operated between 24.00 and 7.00 hrs. The controls are as follows [7]:

- the time is between midnight and 7.00 hrs
- inside air temperature > cooling setpoint 18°C
- the outside temperature > 12°C
- the outside air temperature < inside air temperature

Figure 6 shows one set of the generated curves where the case with no night cooling and day ventilation rate of 2 ach is plotted for comparison. The curves with night ventilation of 2 and 8 ach demonstrate the reduction in peak day temperatures possible during the cooling season. In addition curves of the free cooling provided in kWh/m^2/annum and the number of hours that the fan would run during the night for mechanical systems are shown. The advantages of using a heavy weight construction with exposed thermal mass is clearly demonstrated in these graphs through the achieved temperature reductions, and the higher amount of free cooling available. It should, however, be noted that the curves demonstrate that night ventilation for cooling is a worthwhile strategy to follow even in lightweight buildings.

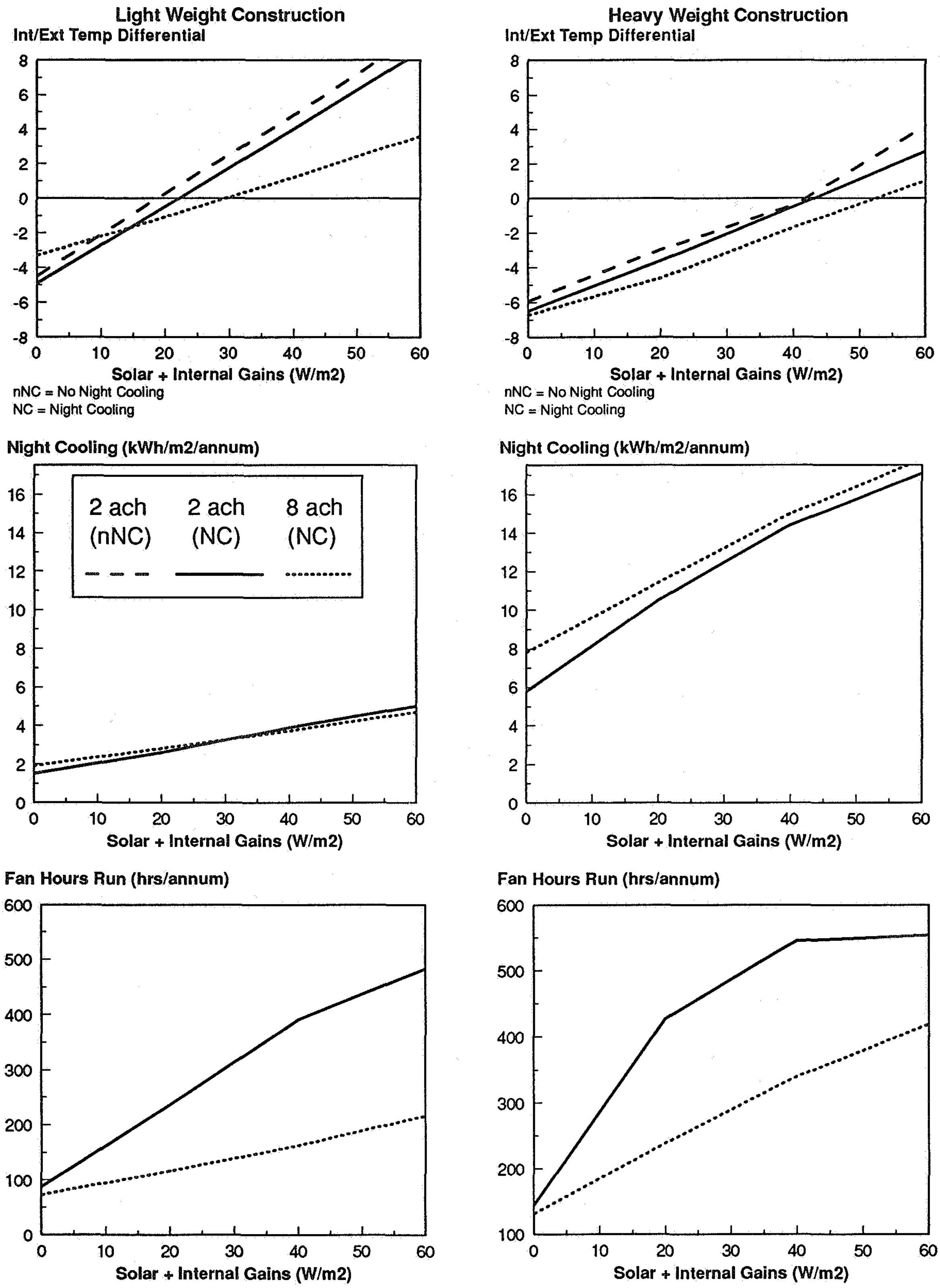

Figure 6: Night ventilation design curves derived using London Heathrow weather data and typical lightweight and heavyweight UK offices. They show the temperature reductions, free cooling provided and the hours a fan would run in mechanical systems.

The design curves so far obtained do provide an indication of maximum expected temperatures during the day and the potential energy savings. However, they do not address the issue of temperatures at the start of the occupied period (when overcooling may be a problem), nor do they give the flexibility of investigating various external weather scenarios and their effect on the internal conditions. To remedy this, a spreadsheet-based pre-design tool which is currently been developed can provide the degree of flexibility required in the initial design stage [8,9]. It is anticipated that the tool will be available before the end of 1996.

5. CONCLUSIONS

The work reported here has assessed the effectiveness of night cooling UK buildings. The study has included desk assessments using bioclimatic charts, thermal prediction procedures using typical office construction and occupancy profiles and field measurements in the office buildings with night-cooling strategies. Results from these studies show that night-cooling is a viable method for addressing the issue of summer overheating in many UK office type buildings.

It is recognised that simplified and pre-design tools either in the form of hard copy design curves or simple computer based calculations will allow the wider uptake of this low energy cooling technique by design and building facilities managers. This paper reports the development of such tools within the current activities of IEA Annex 28 on Low-Energy Cooling Systems.

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NATURAL VENTILATION STUDIES WITHIN THE FRAME OF PASCOOL PROJECT

M Santamouris*, E Dascalaki*, F Allard**

* GR-BES University of Athens, Dept of Applied Physics, 33 Ippokratous Str., Athens, 10680 Greece

** University of La Rochelle, LEPTAB, Av. Marrillac, La Rochelle, 17000 France

NATURAL VENTILATION STUDIES WITHIN THE FRAME OF PASCOOL PROJECT

M. Santamouris[(1)], E. Dascalaki[(1)], F. Allard[(2)]

(1): GR-BES, University of Athens, Dept. of Applied Physics, 33 Ippokratous str., Athens, 106 80, Greece.
(2): University of La Rochelle, LEPTAB, Av. Marrillac, La Rochelle, 17000, France.

ABSTRACT

Natural ventilation studies were carried out within the frame of PASCOOL EC Research Project. Research on this topic included experimental and modeling work aiming to fill the existing gaps in our knowledge of indoor air conditions in naturally ventilated buildings. Experiments were carried out in full scale and test cell facilities during the summer period. Single sided and cross ventilation as well as air flow through large internal openings were the basic topics that were studied. Existing models were validated and new ones were developed. A new computational tool for ventilation prediction was developed, based on the airflow network modeling. An intermediate approach, between network and CFD was proposed to take into account the impact of non-homogeneity on the indoor air motion.

1. INTRODUCTION

Natural ventilation is a very important strategy for the reduction of the cooling load and the improvement of indoor comfort, especially in the Mediterranean countries, where air conditioning systems do not represent a realistic alternative. Prediction of the ventilation rates for design assessment purposes is a complex problem and various methods and models have been proposed. However, very few experimental studies are available and validation of the existing models is limited. Natural ventilation studies were carried out in the framework of PASCOOL programme, which was partly financed by the European Commission, DG XII for Research.

Extensive experimental activities were carried out in full scale buildings and test cell facilities during summertime. The purpose of these experiments was to study three different aspects of natural ventilation:

- single sided ventilation
- cross ventilation and
- air flow through large internal openings

Tracer gas experimental techniques were used in order to derive natural ventilation airflow rates and some limitations in their use were identified. Alternative methods to measure air flows in buildings were proposed. A new algorithm was developed for the computation of experimental airflow rates through large internal openings, when the tracer gas decay technique is used.

Data from the experimental programme were used in order to validate existing models and develop new ones. Further research was undertaken in order to fill the gaps that were identified. Knowledge acquired from experimental and modeling activities was incorporated in a new computational tool for the prediction of natural ventilation rates. Finally, an intermediate approach, between network and CFD, was proposed in order to take into account the impact of non-homogeneity on the indoor air motion.

This paper presents the natural ventilation research activities undertaken within PASCOOL as well as a brief presentation of their outcome.

2. SINGLE SIDED VENTILATION

Single sided ventilation was studied experimentally in full scale buildings and in test cell facilities. Experiments in full scale buildings were carried out in Athens (Greece) and in Madrid (Spain). The test cells are fully equipped, two room outdoor facilities for thermal and solar monitoring. The ones located in Athens and BBRI (Belgium) had been developed within the frame of PASSYS, EC Research Project [1]. Test cell experiments were also carried out in Porto (Portugal). Data from a total of 76 different configurations were gathered to compose the biggest and most important existing database on single sided ventilation. All the experiments were carried out using the tracer gas decay technique, with the exception of those in BBRI, where the constant injection technique was followed. N_2O, SF_6 and CO_2 were used as tracer gases. Two different approaches were used in order to derive the air flow rates in the case of the Belgian experiments: the tracer gas and the heat balance approach. The latter was found to provide a less fluctuating and more accurate flow [2].

Simplified methods were used in order to calculate the air flow rate for some of the experiments. The following simplified methods that were used:

- de Gids and Phaff [3]
- BS [4]
- University of Athens method [5]

Comparison of predicted and measured air flow rates has shown that, in general, simplified methods do not predict the air flow accurately, especially in the cases where the wind forces are dominant. The main reason for this is the fact that the impact of the wind is not correctly taken into account in combination with the flow due to temperature difference.

Five air flow network models were used in order to simulate the experiments that were carried out in Athens:

- COMIS [6]
- BREEZE [7]
- AIRNET [8]
- ESP [9]
- PASSPORT-AIR [10]

All tools gave similar predictions for the air flow rates. Further analysis using Warren plots has proved that these experiments were inertia dominated. As network modeling practically disregards the wind effect in the case of single sided ventilation simulation, a disagreement was observed between predicted and measured air flow rates, the correlation coefficient between the two sets of data never exceeding 0.4. To mend for this inaccuracy, a new model was developed and validated: the CF model.

According to the model, the network model predictions are multiplied by a "correction factor", CF, to yield more accurate air flow rates:

$$Q_{predicted} = CF x Q_{network(Cd=1)} \tag{1}$$

$$CF = 0.08(Gr / Re_D^2)^{-0.38} \tag{2}$$

where:

$Q_{network(Cd=1)}$	: air flow rate predicted by a network model, taking the discharge coefficient equal to unity (Cd=1)
Re_D	: Reynolds number (=UD/ν)
Gr	: Grashof number ($=g\ \Delta T\ H^3 / T\ \nu^2$)
U	: wind speed at 10 m (m/s)
ΔT	: indoor-outdoor temperature difference (^oK)
T	: mean indoor-outdoor temperature (^oK)
ν	: air viscosity (m^2/s)
H	: vertical size of the opening (m)
D	: room "depth" (m), defined as the distance from the wall where the opening(s) is (are) to the wall opposite to it in the studied zone.

If the resulting value from eqn. (2) is less than 0.6, the CF=0.6.

The above described model has been validated with success using data from other single sided ventilation experiments. It was found to improve the accuracy of network models, especially in the cases of important wind speed and small temperature difference. Validation has been done for wind speed ranging from 2 to 10 m/s, temperature difference from 0.5-8 ^oC and room depth varying from 3-7 m.

Based on the BBRI PASSYS Test Cell experiments a "multi-term" model was developed:

$$Q = \sqrt{Q^2_{network}(Cd, \Delta T ...) + (\alpha A V)^2 + (\beta A)^2}\ ,\ m^3/s \tag{3}$$

where:

$Q_{network}$	: air flow rate calculated by network models using Cd=0.66
A	: opening surface (m2)
V	: wind speed at 10m (m/s)
α, β	: wind and turbulent term parameters related to the investigated environment

The multi-term model was derived from a particular experimental condition and this fact restricts the limits of its applicability.

3. CROSS VENTILATION

An extensive literature survey on reduced order models predicting air flow in naturally ventilated buildings revealed that the limits of applicability of most of the existing models are not well defined; therefore, it is very difficult to apply them with confidence.

Three different kinds of experiments have been carried out in Greece, Spain, Belgium and Switzerland, following the decay and the constant injection technique. The Aynsley[11] model was used in order to derive the discharge coefficient for a single zone cross ventilation configuration in Athens. The discharge coefficient was found equal to 0.2 and did not seem to be influenced by the indoor-outdoor temperature difference. In a two-zone cross ventilation experiment held in Athens, the mixing of the tracer gas in the room space was found to be very good and no stratification was observed. This was not in agreement with Murakami's report [12] that an inertial flow would be expected in the room. This was verified in the case of the Mendillori apartment in Spain, where cross ventilation experiments were carried out using constant injection and decay techniques. As the air tightness of the studied space was not good, the decay technique proved to be more appropriate. Analysis of the results showed the existence of a virtual stream tube in the center of the room, which prevented the homogenous mixing of the gas. Cross ventilation experiments were also carried out in a test cell facility, using two tracer gases, namely: SF_6 and N_2O. Air velocities measured near the exterior opening of the test cell were correlated to wind speed data at 18m according to Givonni's model [13]:

$$V_i = 0.45\,(1-\exp(-3.84X))V_z, \quad X=4 \qquad (4)$$

The main result of the study in this topic was that the hypothesis of homogeneity which is adopted by all existing models is no longer valid. The absence of homogeneity imposes the necessity of concentration field measurement, which, at the moment, is restrained by the characteristics of existing gas analyzers. Further research in this topic of ventilation will require the design of experiments and the development of new measurement techniques for specific experimental facilities permitting a rigorous control of the boundary conditions.

4. LARGE INTERNAL OPENINGS

Research activities in this topic focused on two goals:

- to check whether the global value of discharge coefficient equal to 0.4 is valid in the case of a real scale building. This value had been derived from experiments in Liege and in Minibat [14], characterized by a strong stratification and an important boundary layer flow.
- to check whether the air flow resistances formed by open doors in series can be predicted by a simple Bernoulli model and usual discharge coefficients.

For the investigation of the discharge coefficient value, the air flow through a doorway was experimentally studied in a controlled climatic environment, the Optibat test cell in Lyon as well as in a PASSYS test cell in Athens. Two new software tools developed within PASCOOL were used: EXAC 1.0 [15] for the calculation of the experimental and PASSPORT-AIR [10] for the calculation of the theoretical air flow rates. The discharge coefficient was calculated by the ratio:

$$Cd = \frac{\text{Experimental air flow rate}}{\text{Theoretical air flow rate (computed by PASSPORT-AIR for Cd=1)}} \qquad (5)$$

The discharge coefficient was found to vary between 0.47±0.06 and 0.67±0.09 (mean: 0.59±0.08), when the temperature difference varied from 1 to 3.7 ^{o}C. The following correlation was derived:

$$C_d = 0.563 + 4.11\frac{\Delta T}{T} \qquad (6)$$

Thus the derived value of the discharge coefficient was different from the one proposed by IEA/ECB Annex 20. However, it was in very good agreement with the global Cd value of 0.6 proposed by the Canadian Group of Wilson [16].

Air flow through internal openings in series was studied in the LESO building in Lausanne, using the constant injection technique under conditions of buoyancy dominated flow. The Bernoulli model was found to predict the air flow rates with sufficient accuracy. The discharge coefficient is defined as the product of the velocity and the contraction coefficients, C_v and C_c respectively. In this series of experiments a value of C_v=0.7 was derived from velocity measurements. Smoke visualization has shown that the contraction coefficient, C_c, has a value greater than 0.6 for low velocities in large openings. In the case of cross ventilation with several openings in series, C_v approaches unity, especially in the case where the distance between the openings is smaller than 10 times the opening size. During these experiments, the flow was underestimated when a value of C_d=0.6 was used for each of the openings in series.

5. AIR FLOW PREDICTION

Two approaches have been studied within the frame of PASCOOL regarding air flow modeling : Network and Zonal modeling. A new air flow computational tool was developed based on the network concept: PASSPORT-AIR. Research activities on the second approach lead to the developments of a primary zonal model: a Stratification Predictive Model.

5.1 PASSPORT-AIR : A multizone air flow model

This computational tool, based on the principles of network modeling, calculates the air flow rates through building openings, such as windows, doors and cracks, based on a power-law formula. According to the concept of the network approach each zone is represented by a pressure node. Boundary nodes of known pressure are also used in order to represent the outdoor environment. The unknown pressures at the interior nodes of the network are derived from the solution of a non-linear system of equations, which is formed from the requirement of mass balance at every node:

$$\sum_{j=1}^{m} \rho Q_{ij} = 0 \tag{7}$$

where m is the number of zones that are adjacent in the flow direction. An iterative Newton-Raphson method is used in order to solve the system.

PASSPORT-AIR was validated using data from the experiments carried out in PASCOOL. Table 1 summarizes the predicted and measured air flow rates for the experiments carried out in LESO under conditions of buoyancy driven flow. As shown, there is a good agreement between the two sets of data.

LESO building	MEASURED (kg/s)		PREDICTED (kg/s)	
Experiment No	INFLOW	OUTFLOW	INFLOW	OUTFLOW
1	2.03	1.90	1.91	1.92
2	1.10	1.16	1.07	1.07
3	1.39	1.41	1.09	1.09
4	1.00	1.16	0.91	0.92
5	1.01	1.13	0.82	0.83
6	1.64	1.76	1.66	1.68

Table 1: Measured and predicted air flow rates during cross ventilation buoyancy dominated experiments.

The CF model which was developed for single sided ventilation (section 2) was incorporated in PASSPORT-AIR to improve its accuracy in predicting the air flow rate in this mode of ventilation.

5.2 Zonal Modeling for natural ventilation

This modeling approach is an intermediate step between network and CFD modeling. Pressure network models calculate inter-zone air flow rates through cracks and large openings but they do not predict indoor air patterns. Such a prediction is very important when investigating the impact of ventilation on thermal comfort and indoor air quality.

Like CFD modeling, zonal modeling is based on a macroscopic division of the enclosures in subzones. The difference between the two approaches lies in the size of the control volumes, which are larger in the case of zonal modeling, using a typical size in the order of one meter. In each of the subvolumes that are created the conservation equations are formulated, namely:

- The total mass balance
- The enthalpy balance
- The momentum balance along each of the three space coordinates

The unknowns of the problems are:

- The normal heat fluxes (or velocities) on each boundary between the suibvolumes
- The static pressure and temperature in each subvolume

This modeling approach is a very promising one, yet further research effort is required in its computational translation for the model to reach a safe convergence to a unique solution.

6. CONCLUSIONS

A large research effort has covered a wide range of aspects in the field of natural ventilation. Practical answers have been given, following the goal of producing new design methods or empirical knowledge suitable to design purposes. Results and experience gained from the PASCOOL research activities lead to perspectives for future work in both the experimental and the theoretical level. There is an important lack of knowledge, equipment and facilities in order to characterize a non homogenous indoor environment and especially the air flow patterns in naturally ventilated buildings. The intermediate approach of zonal modeling proves to be a promising way to fill the gap between the well mixed hypothesis and the CFD approaches. However, much more effort is required in order to transform the research models developed so far into efficient tools for designers or architects.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

17th AIVC Conference, Gothenburg, Sweden, 17-20 September, 1996

PASSIVE COOLING, SIMULATIONS AND EXPERIENCES FROM REALIZED PROJECTS IN SWEDEN

Engelbrekt Isfält

Royal Institute of Technology, Building Sciences Engineering, S 100 44 Stockholm, Sweden

(Full paper unavailable at time of print)

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Title Passive Cooling, simulations and experiences from realized projects in Sweden

Author Engelbrekt Isfält

Address (including country) Royal Institute of Technology
Building Services Engineering
S-100 44 Stockholm, SWEDEN

Tel: 468-790 85 85 Fax: 468-41 84 32

Abstract

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Passive cooling, simulations and experiences from realized projects in Sweden.

The use of computers for simulating building thermal behavior started early at the Royal Institute of Technology in Stockholm, Sweden. The first example of such use dates from a 1957 study of an exterior wall exposed to solar radiation.

The simulation program, later named BRIS, has gradually evolved with regard to the users and growing computer capacity. It has been used since the early sixties for research projects, design work and development of new systems, among others the ventilated hollow-core slab (Thermodeck) system.

In 1990 the originators of BRIS recieved Swedish Great Energy Award for "distinguised contributions in the field of energy conservation". The jury stated that the knowlege we got from the simulations has lead to an annual saving of energy worth 100 millions Swedish Crowns.

BRIS contains different installation- and control components representing generic models rather than specific implementations. The components can be combined freely to correspond to the principal operation of any HVAC system.

The control strategy is based on a sequence of restrictions on the possible sources for heating, cooling or heat recovery. The restrictions are relaxed successively within each time step in the building model until a solution is found. The order in which the restrictions are to be relaxed may be varied, the capacity intervals can be open ended on one side, etc.

By combining loads and systems minimum energy strategies can be defined and found by the program. When limiting the installed capacities the building dynamics will be more active in the control process which has shown to give a surprisingly high potential to reduce peak power problems and energy use. We now have experience from over 300 buildings using the Thermodeck system for passive cooling. Some of thecse experiences are reported and commented in this paper.

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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WIND TOWERS - OLD TECHNOLOGY TO SOLVE A NEW PROBLEM

D J Harris, R S Webb

Dept of Building Engineering and Surveying
Heriot-Watt University, Riccarton, Edinburgh, UK

Wind Towers - Old Technology to Solve a new Problem

D.J. Harris, R. S. Webb.
Dept of Building Engineering and Surveying, Heriot-Watt University, Riccarton, Edinburgh.

Abstract

Wind towers (scoops situated on the roofs of buildings to catch the wind) have been in use for centuries in the Middle east and Pakistan, to provide ventilation and cooling with minimal mechanical plant. In Europe, the problem of cooling buildings has generally not been significant, but in recent years there has been a trend towards substantial increases in internal heat gains from IT equipment etc., and overheating in summer has become one of our major concerns. This has been dealt with by the use of air conditioning, but in many instances this could be avoided by making better use of natural ventilation through wind towers. This paper reviews the use of wind towers for cooling spaces, and reports on work currently being carried out, using wind tunnel tests on scale models, to examine the adaptation of these principles for use in modern office buildings, in order to avoid expensive air conditioning.

1. Introduction

Wind towers have been in use for centuries in the Middle east and Pakistan, to provide ventilation and cooling without using energy-consuming machinery. In Britain, the problem of cooling our buildings has never been a great one, until recently. The advent of IT and offices using ever larger quantities of electricity has made overheating in summer a serious problem. Last summer was one of the hottest on record in Britain, and emphasised the need for better cooling in offices. Air conditioning incurs both capital and running costs, and in many instances it could be omitted by improved building design. The use of opening windows is not always convenient, particularly in high-rise buildings where wind pressures may be high, and in cities, where traffic noise is excessive. Here, we review an alternative method of providing natural ventilation, an old technique used to supply cooling by means of wind towers, and examine how it can be exploited in today's modern office.

2. Historical background

The earliest examples of wind catchers are from Egypt where they are called *Malqaf*, and are known to us from drawings on papyrus dating from circa 1500B.C. These depict towers with only one windward opening, facing the prevailing North-westerly winds. The roof of the shaft is inclined at about 30° to force the air into the building, and a high central hall provides an outlet for the ventilation air.

The principle of a wind tower is simple. The tower is built on top of the building, has an opening to allow wind-driven air to enter, and is connected to the living space via an opening in the ceiling or a duct reaching to a lower level. Wind pressure forces air down into the building, providing ventilation air. The air may leave the building through open windows or gaps in the fabric. (Figure 1).

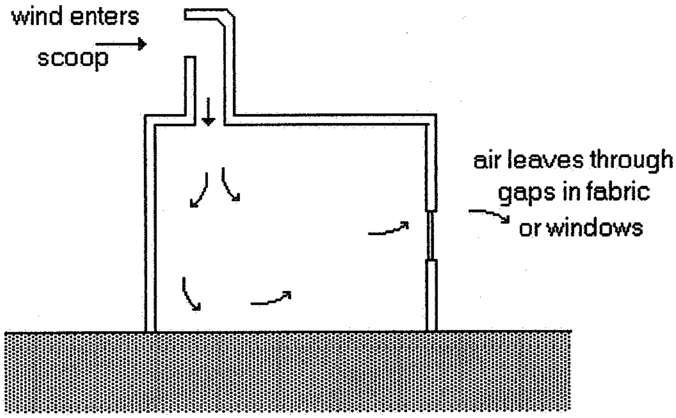

Figure 1. The Basic Wind Tower.

In areas where the wind is always in the same direction a simple uni-directional scoop may suffice, but in most regions the wind may come from any direction, and a scoop or head which can accommodate winds from all directions is required. An alternative which could be developed is a scoop which can be rotated to face the wind (Figure 2).

Variations on the basic wind tower principle include the use of tubes buried in the earth to provide additional cooling (Figure 3), or porous jars containing water to achieve evaporative cooling (Figure 4), and in some cases a combination of both.

Wind towers on houses are widespread in the Middle East, notably in Bastakia, Iran, where there are many such houses dating from the middle ages to as late as 1900, which represent a traditional style of building which developed over the centuries, and is appropriate to the local social and environmental conditions. These are large houses, designed to accommodate extended families of wealthy merchants. They are preferred by the older generation, who consider air conditioning to be draughty and noisy. The wind towers, known as *Badgeer*, are open on all four sides to catch the breeze from any direction and funnel it down to the occupants. In cool weather the vents can be closed off at the bottom with wooden traps [1]. In some Badgeer in Iraq, the duct is connected only to the room at the lower floor level and the cooled air is allowed to circulate to the upper rooms by natural buoyancy.

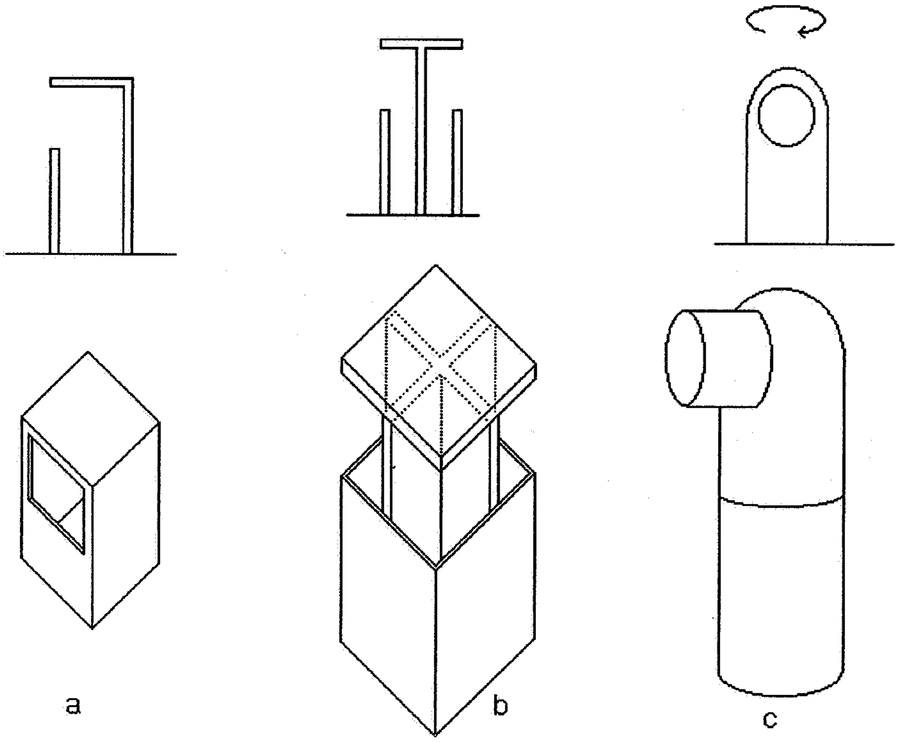

Figure 2. Range of Scoop Types. a. Uni-directional. b. Multi-directional. c. Rotating.

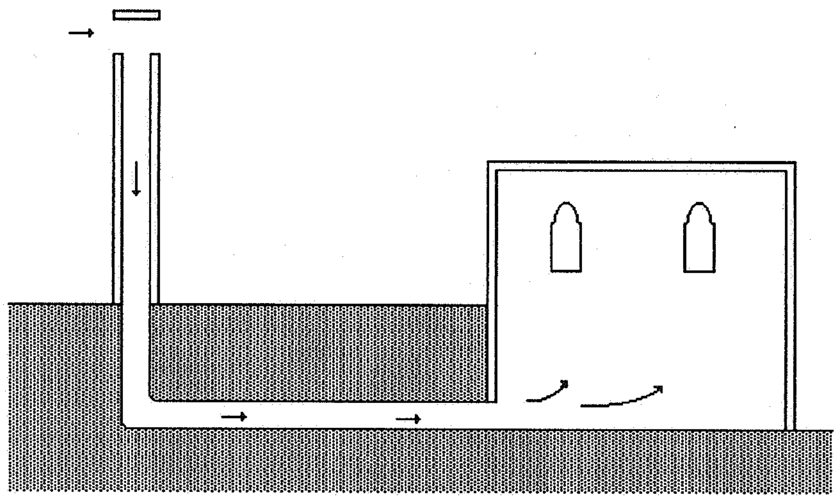

Figure 3. Additional Cooling Through Earth Tubes.

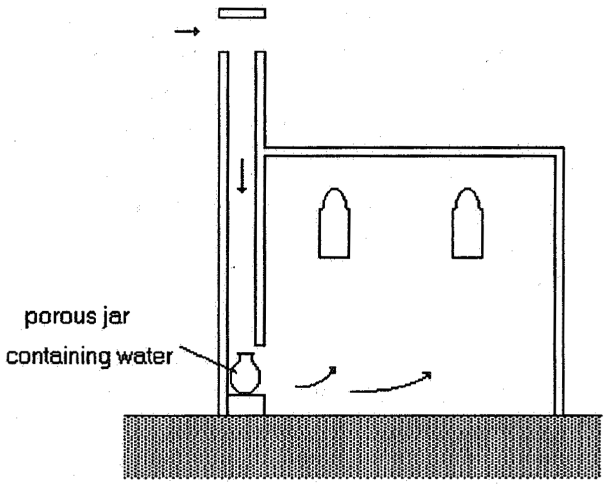

Figure 4. Evaporative Cooling.

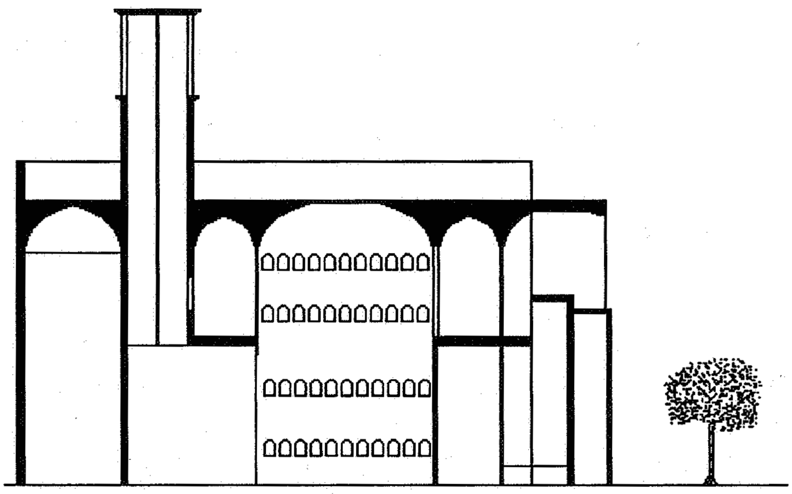

Figure 5. A Typical Badgeer on a House in Yazd, Iran.

In the town of Thatta in the Sind province of Pakistan, many houses have wind towers. The design here is different, and because the wind in the hot season is almost always from the south west, they are built to catch the wind only from that direction. These wind catchers in general have a square or rectangular plan, and are slightly enlarged at the top. The 'serviced' rooms (bathroom, kitchen) are all on the lee side of the building and wind tower. The wind tower is equipped with a metal grating to prevent entry into the house, and a trap door to close off the wind when not required. In this region, even some modern houses have wind towers - made from concrete rather than the traditional materials of wood and sun-dried mud-bricks. [2]. In Yazd in Iran, tall wind towers adorn some houses, with eight shafts to catch wind from all directions, and longer sides towards the prevailing north west and south east winds. (Figure 5) Felt hanging from the walls of the shafts can be watered for evaporative cooling, or pitchers of water made of porous stoneware. [3].

Each area using wind towers has evolved its own designs according to the requirements, the local climate conditions and local availability of building materials. Most wind towers today are on older buildings, although a number have been built in recent years, interest in wind towers having increased due to the mounting cost of energy, and greater concern for environmental considerations.

3. Mode of Operation

Wind catchers scoop the wind and drive it down through the building. Resistance to the flow of air through the building is created by friction against the walls of the tower or duct, with additional resistances at bends and from internal partitions. The mechanisms by which the air cools the building and its occupants are manifold.

- Air movement against the body creates a fresher feeling, even though the air may not be cooler.
- If the air is cooler than that inside, it may directly cool the occupants and the building fabric.
- At night, the air will cool the building fabric and make it pleasanter the next morning.
- The air may also be drawn over jars of water and produce evaporative cooling.

4. Existing Performance Data

The design of wind towers in Sind and the Middle east developed over the centuries to suit the local climate, social conditions, and available materials, as the best building practice should. The traditional approach to design through rule of thumb and experience unfortunately leaves us with little record of the process by which the shape and size of the towers were reached. Proper, quantitative design processes and guidelines need to be developed for today's environment, where the users' demands on the building are greater. Today's urban environment is much more crowded, space is limited, and there are limits to the height of towers we can erect determined by structural considerations, aesthetics, overshadowing of nearby buildings, and so on.

There is relatively little quantitative data on the performance of existing wind towers. Such data as there is suggests that the air speed at the outlet of the scoop is between 8 and 15% of the wind speed at the inlet. In a room 5m by 3m by 3m with a 1m x 1m duct, at a wind speed of 10m/s the speed of the air in the room was 1m/s [4], but no record of the overall ventilation rate was made. This air movement increases the acceptable maximum temperature by approximately 2.0°C at 45% relative humidity.

5. Variables Affecting the Performance of Wind Towers.

The chief environmental variables are the wind speed and direction, and for a given design and orientation there should be a fixed relationship between the wind speed and ventilation rate. The nature of this relationship is affected by:

- Tower dimensions in relation to room below.
- Form of tower and scoop head.
- Presence of internal obstructions which create a resistance to air flow.
- Effect on local wind patterns of nearby structures.

Further, the cooling effect that ventilation has on the fabric of the building and the internal air is a function of;

- The ventilation rate.
- The air temperature.
- The relative humidity.

6. Experimental work.

A 1:100 perspex model was constructed as shown in figure 6, representing the top storey of an office building, and was tested in a wind tunnel at a range of simulated wind speeds, with the wind direction parallel to the long axis of the model. The opening in the wind tower and the roof of the building represented 0.5% of the floor area, equivalent to a 0.075m^2 duct (27cm x 27cm) 1ft by 1ft in a 5m x 3m room .

Three sets of tests were carried out:-

a. Air movement tests using anemometers.
These measurements enabled the air velocity within the room to be measured. The air velocity at the outlet was also measured and correlated with the wind speed (Figure 7). This gives an overall mean ventilation rate throughout the space, but does not indicate the actual ventilation rate at particular locations.

b. Flume tests using coloured dyes.
These were carried out to evaluate the ventilation distribution in the model. They showed that the turbulence induced as the air entered the "room" created a wide spread of ventilation air throughout the room, and that ventilation air reached every corner of the room, using the vent position shown.

c. Vent rate tests using SF_6.
These were carried out using standard tracer gas decay technique. The detector location was centrally in the room. The ventilation rates achieved in the centre of the room are shown in Figure 8.

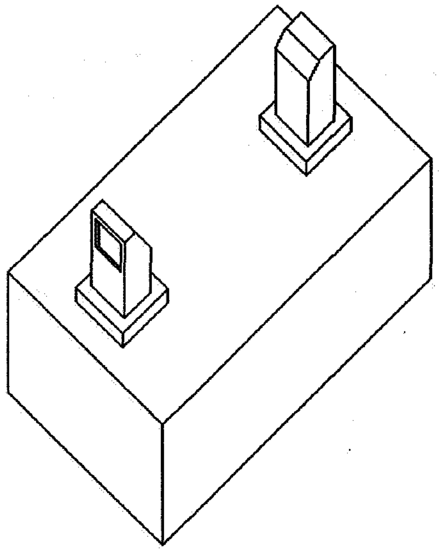

Figure 6. The model building used in the wind tunnel tests.

8. Results

The mean ventilation rate within the model should clearly be some function of the wind speed, for a given set-up. However, due to turbulence, the ventilation rate at particular locations may vary from this, due to changes in air flow mechanisms at different wind speeds from laminar flow to turbulent flow. The overall ventilation rate is roughly given by the air speed at the "exit" duct and shows good correlation.

Cleary the relationship between the internal ventilation rate and the wind speed is affected by a number of factors, including the vent size and shape. The results presented thus refer only to this particular combination., and a mathematical model will be developed in due course.

9. Conclusions.

Natural ventilation is becoming increasingly attractive to users wanting improved air quality and lower energy bills. Wind-driven ventilation from roof-mounted towers has been used successfully for centuries, with little attempt to date to quantify the effects or develop scientifically-based design guidelines. This work represents the initial stages in developing a fuller understanding of this form of ventilation. There was some correlation between wind speed and internal ventilation rate, and ventilation air was well distributed.

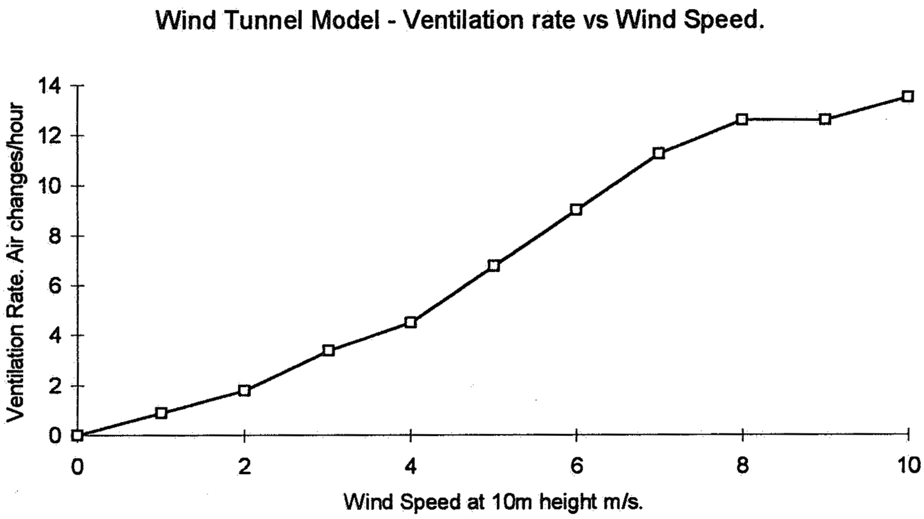

Figure 7. Results of Anemometer Measurements.

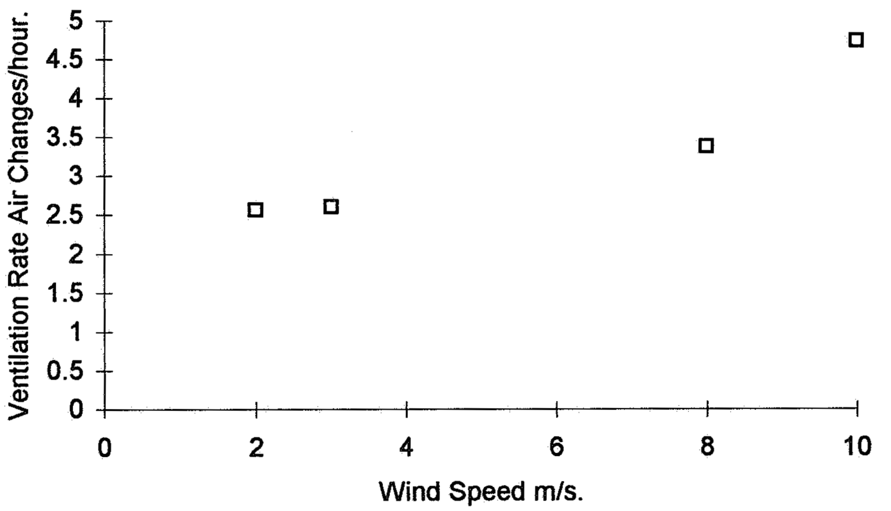

Figure 8. Results of Tracer Gas Ventilation Rate Measurements.

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ENERGY EFFICIENT AIR DISTRIBUTION SYSTEMS

Lennart Jagemar

Dept of Building Services Engineering, Chalmers University of Technology, S 412 96 Gothenburg, Sweden

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Title: **Energy Efficient Air Distribution Systems**

Author: Lennart Jagemar, Ph.D.
Address (including country): Dept. of Building Services Engineering
Chalmers University of Technology
S-412 96 GOTHENBURG
Sweden

Tel: +46-31 772 1147 Fax: +46-31 772 1152

Abstract

Fans account for a substantial part of the electrical energy end-uses of HVAC systems in commercial buildings. Monitored case studies in temperate climates give figures between 30% to 90%, with an average of 60% to 80%. This paper gives guidelines on how to design energy efficient air distribution system. Here energy efficiency is defined as:

Energy Efficiency
The ultimate goal of the total building design process is a building that, in every detail (e.g. the air distribution system), has the lowest possible energy requirements that can be achieved without jeopardising the fulfilment of any basic demands, and by sacrificing only a reasonable amount of resources.
Consequently, when selecting technical solutions and taking measures for achieving an energy efficient building, the following two main criteria must always be observed:
- The energy-related solutions must, not in any way, jeopardise the fulfilment of any of the basic demands, e.g. indoor air quality, thermal climate or lighting quality.
- The energy-related solutions and measures must be cost-efficient;

The method consists of three stages:
1. Select suitable **energy performance ratios**;
2. Use **marginal profitability analyses** and **data from monitored case studies** (here mainly European office buildings) as well as from **building codes, standards**, etc. to;
3. Obtain **energy targets**, i.e. numerical values of the energy performance ratios.

The energy performance ratios of current interest is:
- Specific Fan Power, *SFP*: Fan power over air flow rate at design conditions [kW/(m³/s)];
- Specific Fan Energy, *SFE*: Annual fan electricity use over design air flow rate [(kWh/year)/(m³/s)];
- Utilisation factor: Annual fan electricty use when VAV controlled over maximal fan electricty use without controll [1].

Marginal profitability analyses give that an optimum Specific Fan Power, *SFP*, for Swedish conditions is between 1.8 to 2.7 kW/(m³/s) depending on the annual run time of the air handling unit; for a typical office building with a run time of 2,500 to 3,000 hours/year the optimal *SFP* is 2.3 to 2.5 kW/(m³/s). Typical values from monitored office building are between 2 to 4 kW/(m³/s).