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BDAI

17th AIVC Conference Optimum Ventilation and Air Flow Control in Buildings

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Preface

International Energy Agency

The International Energy Agency (IEA) was established in 1974 within the framework of the Organisation for Economic Co-operation and Development (OECD) to implement an International Energy Programme. A basic aim of the IEA is to foster co-operation among the twenty-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*
- Annex 22 Energy Efficient Communities*
- Annex 23 Multizone Air Flow Modelling (COMIS)
- Annex 24 Heat Air and Moisture Transfer in Envelopes*
- Annex 25 Real Time HEVAC Simulation
- Annex 26 Energy Efficient Ventilation of Large Enclosures
- Annex 27 Evaluation and Demonstration of Domestic Ventilation Systems
- Annex 28 Low Energy Cooling Systems
- Annex 29 Daylighting in Buildings
- 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

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

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS?

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Synopsis

Air quality, air flows in buildings, and ventilation are most important topics. Good air quality is however critically dependent also on other things than ventilation, e. g. source strengths, as is well known. The theme of our conference could also lead to the assumption that there is a simple connection between ventilation and air quality. As the public tend to see ventilation systems as responsible for bad air quality, it is important to state that there is no general criterion for good air quality possible to use in practice to control ventilation processes. Automatic ventilation control is still possible in specific cases only.

The ventilation process depends on the total air exchange in the building, which is difficult to control. A related question is: what is acceptable as ventilation air? Is air leaking in through the outer wall acceptable? Is air supplied through the staircase acceptable? To answer such questions air quality criteria are needed too.

1. Introduction

The original meaning of "ventilation" is: "expose to the wind." The original purpose was:

- to cool when necessary by letting warm air out and fresh air in
- to clean room air when necessary by letting polluted air out and fresh air in

This original meaning of "ventilation" related naturally to outdoor air, which was "fresh." Ventilation efficiency, for clean supply air the quotient between the concentration of pollutant in the exhaust air and the mean concentration in the room, thus is defined for pollution sources in the room. Today, it is of course no longer obvious that outdoor air is "fresh." Exchanging the air in a building is no longer a certain method to improve indoor air quality.

Indoor air quality depends on

- the strength of pollution sources in the room and, if applicable, the effectiveness of local exhausts
- the quality of supply air and other air entering the room
- the rate of air exchange
- the quality of air exchange: ventilation and air exchange effectiveness

To improve air quality in a room, source control should be the first priority. Potential sources are processes, tobacco smoking, furniture and carpets, building materials, chemicals and detergents, and more. Then the quality of the air entering the room should be checked. Do supply air or leak air transport pollutants into the room? Also here there are many possibilities: exhaust gases from traffic and environmental pollutants in the outdoor air, emissions from neighbouring buildings, radon from the ground, mould from the ground or the building itself, pollutants from neighbouring rooms, pollutants from the air handling system, and more. The quality and rate of air exchange should then be considered. Is there any short-circuiting of air from supply to exhaust? Are there badly ventilated stagnation zones creating air quality problems? Is there unnecessary recirculation of pollutants in the room? Should a controllable air flow rate be changed?

Indoor air quality is of course no stationary matter. Room conditions as air exchange efficiencies depend much on air convection, changing with the thermal loads during the day. Outdoor conditions, building status and use, building and mechanical system maintenance are examples of important factors subject to change. Proper means for building users to change, control (or at least influence) the ventilation process and system should be provided. Proper maintenance is most important, and to achieve this is the responsibility of the building management but also of the designers. Energy, flexibility, and economic aspects provide together with comfort and noise criterion's boundary conditions for the engineers. The goal is healthy building users, satisfied with their indoor climate.

The aim of this paper is to discuss the possibilities to control air quality with ventilation, to control ventilation with air flows, and to control the air flows in a building.

2. Air flow control

The resulting air flow in a building depends on

- building size and design
- temperature differences between the air in the building and outdoor air
- wind forces around the building (which forces also depend on the surroundings)
- the distribution and size of openings and leaks at the building perimeter
- the distribution and size of openings and leaks within the building
- ventilation openings, ductwork, and fans; ventilation strategy and control

Within a single room the flow depends on

- convection currents (temperature differences)
- air flows from air terminal supply devices and other openings letting air in
- the location of air terminal exhaust devices and other openings letting the air out
- the room users

Most of these factors vary, rapidly or slowly: outdoor temperature and wind, the distribution of open doors and windows, leak areas, temperature differences within a room or between rooms, and the use of the building and its rooms. To make control of the resulting air flow easier one can try to minimize these influences: entrance lobbies, windows that cannot be opened, very tight buildings, mechanical ventilation systems with big pressure drops or constant flow devices in order to minimize weather dependence, and more. However, many such measures tend to decrease the freedom of the building users and make them experience ventilation as a burden, not a benefit. Poorly designed, constructed, or maintained systems enhance of course this impression, creating a lack of trust. Many ventilation engineers work in "wakes" of such experiences. Another strategy is to use the factors that can be controlled to compensate other influences. This is of course the same strategy as most building users have when they open windows. Consequently, also this strategy must be used with care, not to decrease the freedom of the user. It is also only too easy to make such systems complicated, or at least appear complicated to the user.

The air flow in a building depends on the ventilation system. A mechanical supply-exhaust system tends to control the flow rates, the indoor pressure has to adjust. Mechanical exhaust

systems decrease indoor pressure. "Natural" ventilation systems tend to induce pressure differences, flow rates depend on opening sizes. Only systems with mechanical supply air allow a direct control of distribution of ventilation air to rooms, at least part of it, so minimum flow rates can be reached at "all times." Exhaust and "natural" ventilation systems depend on the distribution of openings, that is ventilation openings, open doors and windows, and leaks. The area relations between the openings decide the flow rates. An open window, creating a big flow area compared with other openings, can "steal" all the ventilation air, leaving other rooms very poorly ventilated.

This short discussion illustrates the difficulties to control air flows in buildings. Evidently only part of the total air flow can be controlled, how big depend on the system design, on building tightness, on window openings and other building use, and on the efficiency of the maintenance organization.

3. Ventilation control

What is "ventilation air?" Supply air can be defined as air intentionally let into a space or a room, through terminal devices connected to ductwork or through openings intended to allow inflow of outdoor air. Exhaust air can be understood as air leaving a room or space, through terminal devices connected to ductwork or through openings intended to allow outflow of room air. But air entering or leaving a room otherwise also takes part in the air exchange, as measured with conventional tracer gas tests. Then, what is acceptable ventilation air? Is air that has leaked into a building through an outer wall acceptable? Obviously not always but when? Is air supplied through the staircase of an apartment house acceptable or is it only contaminated leakage air? Studies show (Herrlin, 1992) that in some cases such air represents a big part of the total air exchange in a flat.

The answer to such questions is: it depends on the air quality. Many excellent studies have been made to define criteria for the air quality (see proceedings from the conference series "Indoor Air" and for instance Cain et al 1995), but criteria feasible for ventilation control seem to exist only for specific cases, as CO₂ concentration as a measure of body odour. The difficulties regarding the quality of supply or other "ventilation air" are similar: only criteria for specific contaminants can be established and continuously controlled, allowing no general guarantees for good air quality. This is also true for sensory olfactory measures, although these have the benefit of direct aiming at user satisfaction. During later years the quality of mechanically supplied air has been questioned, which of course emphasizes the problem.

A traditional strategy is to prescribe minimum air flow rates for health (which hopefully not should be taken as maximum flow rates when energy auditing). Such flow rates have an empirical background and cannot be expected to work when new technologies, materials, processes and practices are introduced. The ventilation engineer cannot guarantee the air quality.

5. Discussion

Air quality depends on many factors, ventilation is only one. Ventilation is a process combining influences of the efficiency of air exchange, transport of air contaminants, and the quality of air entering the room or space. Air flows and air exchange in the building depend on many other factors than the ventilation system, as have been previously discussed, and are difficult to control. Ventilation control is not possible without an air quality criterion. Therefore, automatic ventilation control is still feasible in specific cases only.

Often ventilation is described or mentioned in such a way that it appears to the public as the key to good air quality. Then ventilation also gets the blame when air quality is bad, although the reason can be new sources of pollution, changed traffic conditions outdoors, or similar. The ventilation community must make the limits and possibilities of their technology clear. In practice, a ventilation system transports air flows at specified rates (within some limits). Noise criteria should be met. For supply systems, supply air quality should be only marginally lower than outdoor air quality, and with fewer particles. Supply and exhaust terminal devices should be chosen so criteria for draught and comfort not are violated, and so air exchange and ventilation efficiencies are acceptable. This is what should be expected of the ventilation system. In the design stage, the ventilation engineer should of course take active part in decisions about system air flow rates, and has also an obligation to work for flexibility and preparations for control and maintenance.

The aim of ventilation is good air quality for the building users. The designer of ventilation systems therefore is critically dependent on good indoor air quality criteria. The interaction between the building users and the ventilation system is also a key issue: the system must allow user influence on performance and control. The system should also be flexible, to meet varying demands. Studies about the relation between ventilation and air flows in buildings, and about the controllability of such air flows thus are very important to investigate the limits for this freedom.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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COMPUTERISED METHODS FOR BALANCING VENTILATION SYSTEMS

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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COMPUTERISED METHODS FOR BALANCING VENTILATION SYSTEMS

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SYNOPSIS

Adequate ventilation with the "right amount of air, to the right place and at the right time" are important factors for achieving a good indoor climate. Thus it is of prime importance that the ventilation system is working properly. Using traditional methods, it is a very time consuming, and some times even impossible, task to balance ventilation systems to achieve correct air distribution. In most countries the growing concern about Indoor Air Quality has resulted in Building Codes demanding increased ventilation rates. This makes it even more critical that ventilation systems are correctly balanced. Otherwise the effort of solving one problem: "inadequate ventilation", may create two new problems: "draught and noise".

This paper describes the "DPM method" which is a new method for balancing ventilation systems. The balancing "starts at the designer's desk" by doing a comprehensive computerised pressure loss analysis. The results from this analysis are then input to a PC connected with an electronic instrument. The PC/instrument may be used for balancing when the air terminals are already installed in the building. However, there is a much larger potential for time and cost savings when using the DPM method for pre-setting air terminals at the factory. For this purpose a "DPM machine" has been designed for computerised pre-setting and bar code labelling of air terminals.

With impressive results the ventilation systems in a number of buildings have been balanced in the last two years using the described innovative technology.

1. BACKGROUND

Sponsored by the Norwegian Research Council (NFR), three years ago the VEKST Foundation started developing a new method for balancing ventilation systems. It was based on the philosophy that "balancing starts at the designer's desk". I.e. from the very start the designer should have in mind how to balance the system. A necessary tool in this phase of work is a good computer program for duct design and pressure loss analysis. Such a program will give a complete picture of necessary pressure drop in dampers and air terminals in order to give design air distribution.

Even today, with the computer being a common inexpensive tool, many engineering consultants and contractors still consider making computerised pressure loss calculations of the ductwork a more or less "academic exercise" of little practical value. Being the supplier of a well known Norwegian duct calculation program we certainly do not agree with this opinion. But we felt that we had to do something in order to make duct analysis more attractive, e.g. by making the calculated result directly applicable for the final balancing of the system. We came up with a solution based on recent technological development of computers, electronics and ventilation equipment:

- * The Personal Computer has become an inexpensive common tool for most engineers. High computer costs are no longer an excuse for not making pressure loss calculations.
- * There are many computer programs for doing pressure loss calculations available on the market. Tests have proved that results from well known programs differ very little from actual measurements.
- * Reliable electronic pressure transducers are available that makes it very simple to measure the low air pressures that occur in ventilation systems. The readings are easily presented on a display or as an analogue electrical signal.
- * Dampers and air terminals are frequently equipped with a differential pressure device for measuring the air flow.

As can be seen, the prerequisites for designing and balancing duct systems in a more "engineering way" is now a reality. In the following a new method for balancing ventilation systems, the "DPM method", will be presented.

2. THE DPM METHOD

Any air terminal or damper has, for a given adjustment, a unique pressure loss characteristic. In ventilation systems the flow is usually turbulent, and in that case the relationship between air flow and pressure loss can be expressed as:

Average exponent in $AVAC = 1,9$
 Smooth duct = 1,7

$$P_t = k_1 Q^2 \quad \text{or} \quad Q = k_2 \sqrt{P_t} \quad (1)$$

Where: P_t = total pressure loss Q = air flow k_1 and k_2 = constants

Figure 1 shows Eq. 1 for a damper in three different positions. Imagine that a pressure loss calculation has shown that the damper must produce a pressure loss P_{nom} when the required air flow is Q_{nom} (point 1). The constant k is found by using P_{nom} and Q_{nom} in Eq. 1 and solving for k . If the damper is fixed in the correct position, air flow and corresponding pressure loss will follow the parabola through point 1. This parabola is the pressure loss characteristic of the damper in that position.

If you have equipment that "tells you" when you are on the desired pressure loss characteristic, air terminals and dampers can be given the correct adjustment even when the air flow is far off from the desired, or nominal value, for instance point 2 in Figure 1. This is what the DPM method is all about: "finding the correct pressure loss characteristic", or k -value, for air terminals and dampers.

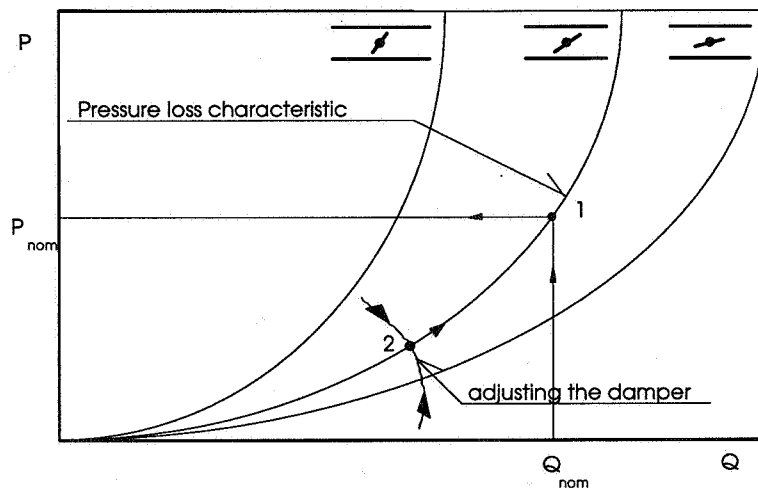


Figure 1 Pressure loss characteristics of a damper in three different positions

3. ELECTRONIC EQUIPMENT

By using two electronic pressure transducers and an "A/D-converter" connected to a PC with a standard RS-232C cable, it is possible to find the pressure loss characteristic of air terminals and dampers. Figure 2 shows one possible way of doing this. The differential pressures for both the air flow measuring device and the flow resistance are measured simultaneously. The fact that two differential pressures are measured is the reason for the name "DPM Method" (Dual Pressure Measurements). Certainly the optimum situation for using the DPM method is when the air flow measuring device is an integral part of the equipment, but that is not a requirement.

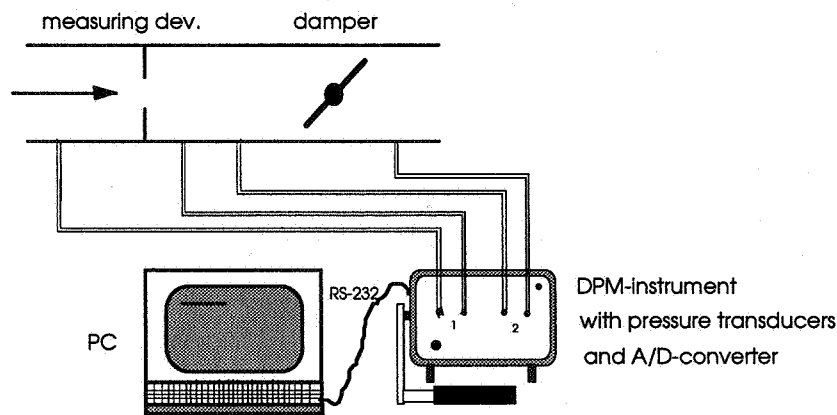


Figure 2 The principle of the DPM Method

When adjusting the air flow through the air terminal/damper the two pressure differences are measured continuously (1 - 2 times per second) by the program running in the PC. The program calculates the k-value of the momentary pressure loss characteristic and compares it with the desired k-value. When the correct k-value is reached, the PC informs you of that by giving a high pitch beep.

The DPM method allows you to adjust the damper or air terminal without having to bother to much about the magnitude of the air flow. Tests have shown that the adjustments may be done with the air flow as low as approx. 30 % of the design air flow. The lower limit is due to the danger of getting laminar air flow.

4. APPLICATIONS

Originally it was assumed that the main application of the DPM method would be for adjusting air terminals and dampers after they have been installed in the building. I.e. using the DPM Method as an alternative to other methods of balancing. Because the air flow at the time of balancing is of little importance, it is possible to do the balancing *before* the ventilation system is completed. For instance it is possible to balance a floor by hooking up a transportable fan to the ducts system of that floor.

The next logical step is to pre-set the air terminals or dampers in advance, i.e. at the factory before they are shipped. The goal of a present project is to develop the necessary procedures and tools for doing this is . The project is a development contract with the Norwegian Defence Construction Services (FBT), The Norwegian Industrial and Regional Development Fund (SND) in co-operation with Auranor A/S, a major Norwegian manufacturer of ventilation equipment. Figure 3 shows a computerised machine for pre-setting air terminals which will be installed at the factory. The machine consists of a small fan, a device for measuring air flow, a DPM-instrument, a PC, two printers and some pieces of ductwork. The ducts which are connected to the air terminals being pre-set are easily exchangeable. Tests made with a

prototype of this machine shows that it is quite easy to adjust an air terminal to the required pressure loss characteristic in less than 30 seconds.

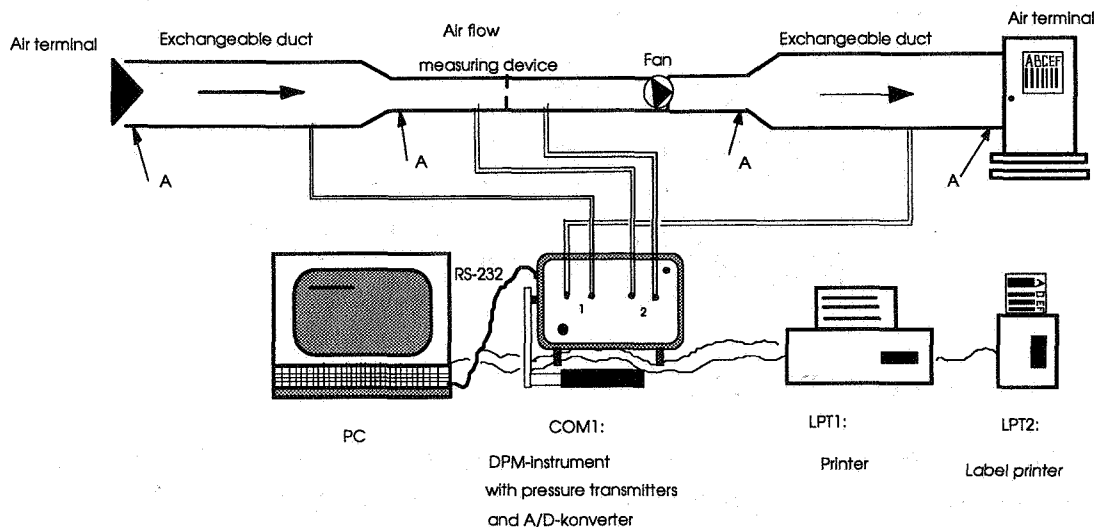


Figure 3 A "DPM machine" for pre-setting air terminals

At the same time as the setting is done, labels are printed to identify the correct location of the air terminal in the building. Two labels are attached to each terminal. One of the labels is human readable telling the installation people where to install the air terminal. It is put on a place that makes it non-visible after installation. The other label, which is quite small and has a bar code, is visible after installation. Thus, using a bar code reader connected to a portable PC, it is quite easy to check that an air terminal is installed in it's right place when doing the final control.

With the air terminals pre-set as described above, the physical balancing process will be reduced to making control readings of the air flows and doing some minor adjustments for a few terminals. In order to speed up this process, as well as giving a good documentation, a new PC-based instrument has been developed. It consists of 8 pressure transducers and an A/D converter connected to the serial port of the PC. The instrument makes simultaneous pressure readings of the air flow measuring devices of 8 air terminals.

The 8 readings are shown on the PC screen as "living columns" showing the relative air flows, i.e. actual air flow divided by design air flow (%). When the relative air flows are equal, the air terminals are balanced with respect to each other. By adjusting the damper for this branch of the ductwork and/or adjusting the speed of the fan, the correct absolute air flow is established.

Usually there are more than 8 terminals in a branch. In this case one starts with the 8 terminals farthest away from the fan. When they are balanced with respect to each other, the next 7 air terminals are connected to the instrument with one of the terminals in the previous group still

connected to the instrument. The 7 terminals are then adjusted to give the same relative air flow as for the one still connected with the terminal in the previous group. This way you work your way through for the whole branch. By using this technique of always having a connection to one of the air terminals in the previous group, all of the terminals in the branch will end up having the same relative air flow as the last group of terminals that was balanced (the "proportional principle").

Each time a group of air terminals is balanced the result is saved to a disk file in a format that is accepted by common spread sheet programs (Lotus, Excel ..). This way it is easy to produce the necessary dokumentation of the work being done.

5. EXPERIENCES

In the past two years VEKST has been balancing ventilation systems in a number of buildings using the described innovative technology. Most of the buildings are buildings where it is of prime importance that the air distribution is correct. E.g. hospitals, laboratories (clean rooms), defence installations and some prestigious office buildings.

More than 3000 extraction air terminals has been pre-set using the DPM method. When controlling the air distribution after installation, it is found that less than 5 % of the terminals need to be adjusted. Pre-setting of supply air terminals is now being tested in the present project. Preliminary tests show that we can expect equally good results as for the extraction air terminals.

The principle of the DPM method may also be applied to hydronic systems, but we have no practical experiences so far.

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

THE EFFECT OF RECIRCULATION ON AIR-CHANGE EFFECTIVENESS

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The Effect of Recirculation on Air-Change Effectiveness

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Synopsis

The effect of recirculation on the age of air is described. A new effectiveness measure called the relative air-change effectiveness is defined in such a way that the air distribution pattern in a room may be quantitatively characterized even when the age of the supply air is non-zero. This admits the evaluation of air distribution patterns in single-zone systems that recirculate air, and also multizone systems with or without recirculation. It is shown that the relative air-change effectiveness may be calculated either solely from age of air measurements or from age of air measurements and flow rate measurements. Re-evaluation of previously published experimental data demonstrates how knowledge of the relative air-change effectiveness may change conclusions drawn from experiments in working buildings.

1 Introduction

Ventilation performance measures have been developed for quantifying the behavior of ventilation systems and for comparing the behavior of different systems. These measures are either related to how well fresh air is delivered or to how well contaminants are removed. Although these two concepts are closely related, this paper focuses exclusively on the former.

Several different definitions of air delivery performance have been proposed. Most often experimental determination of air delivery performance is conducted with tracer gases. Air delivery performance is determined either from steady-state concentration levels of tracer gas or from a transient tracer gas response. In [1] an air delivery performance measure called the relative ventilation efficiency was defined as a ratio of concentration differences at steady-state. Methods for determining this measure from transient response tests were discussed. In [2] a performance measure called relative air diffusion efficiency was defined as the ratio of the nominal time constant to the mean age of air. In [3] ineffective ventilation was modeled by assuming that some of the supply air bypasses the zone and flows directly to the return duct. A performance measure called the ventilation efficiency was then defined as one minus the bypass factor. In [4] a performance measure called the pollutant control index was defined as a time-averaged tracer gas concentration multiplied by a constant that is a function of the tracer source strength and the size of the space.

In many kinds of heating, ventilating, and air-conditioning (HVAC) systems, the ventilation function is coupled with the heating and cooling function. In order to deliver the required quantity of air to different zones of a building at the required temperature,

it is necessary to recirculate and redistribute some of the air that leaves these zones. This poses a problem for evaluating ventilation performance using a measure such as the relative air diffusion efficiency because it was developed under the assumption that “neither is it possible for a molecule either to return upstream once it has entered the room, or to re-enter the room once it has left it” [2].

In this paper, the effect of recirculation on air-change effectiveness calculations is described. First the age of air in the presence of recirculation is presented. Then it is shown how recirculation affects the conventional measure of air-change effectiveness. The theory of air-change effectiveness is extended to include single-zone systems with recirculation and multi-zone systems with or without recirculation by defining a new measure of air-change effectiveness. Implications of this theory for ventilation performance evaluation and control are discussed.

2 Age of Air

Consider the ventilation system depicted in Figure 1 in which a single zone is supplied by air that is a combination of recirculated return air and outdoor air. The fraction of

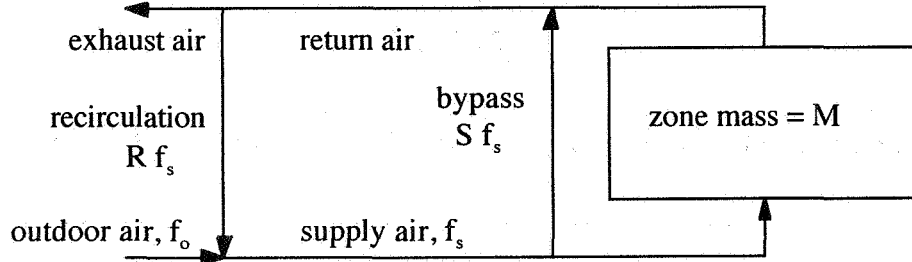


Figure 1: Schematic diagram of a single-zone ventilation system with recirculation and bypass.

recirculated air in the supply air will be denoted as R . Assume for now that the bypass factor, S , is zero. The age of air at a point in the room is the length of time that it takes a massless particle entering the system from outdoors to reach that point. The age of air will be denoted as a . The mean age of air in the room is the volumetric mean of the age of air at all points within the room. It will be denoted as $\bar{a}$.

First the age of air for two abstract systems will be described. Perfect mixing systems are the most commonly considered abstract systems used to analyze the performance of ventilation systems. When the air in the system shown in Figure 1 is perfectly mixed, then the accumulation dynamics of a component of the air (e.g., a tracer gas) can be described by the following equation

$$M\dot{\omega} = f_s(\omega_s - \omega) + r \quad (1)$$

where M is the mass of the air in the zone, ω is the mass concentration of the component in the zone, the dot notation refers to differentiation with respect to time, f_s is the mass flow rate of supply air, ω_s is the mass concentration of the supply air, and r is the rate at which the mass of the component gas is generated within the zone. It can be shown that the accumulation dynamics can also be expressed as

$$M\dot{\omega} = f_o(\omega_o - \omega) + r \quad (2)$$

where f_o is the outdoor air mass flow rate and ω_o is the outdoor air concentration. Since this is a linear, time-invariant, first-order differential equation, the mean age of the air is

equal to the nominal turnover time, which is denoted as τ_n

$$\bar{a} = \tau_n = \frac{M}{f_o} \quad (3)$$

For a perfect-mixing system, the mean age of the air is independent of the recirculation fraction.

Another kind of abstract system that is commonly considered when analyzing air-change effectiveness is the system with plug flow and no diffusion (PFND). When f_o and R are constant, the PFND system is a linear, time-invariant transport delay. The zone delay is the time that it takes a particle to cross the zone from the supply to the return. It is equal to the local nominal turnover time of the zone, denoted as T

$$T = \frac{M}{f_s} \quad (4)$$

It can be shown that for a PFND system

$$\bar{a} = \frac{T}{2} \frac{1+R}{1-R} \quad (5)$$

which implies that the age of air depends on the recirculation fraction for a PFND system. Figure 2 demonstrates how recirculation affects the age of the air. The figure shows the

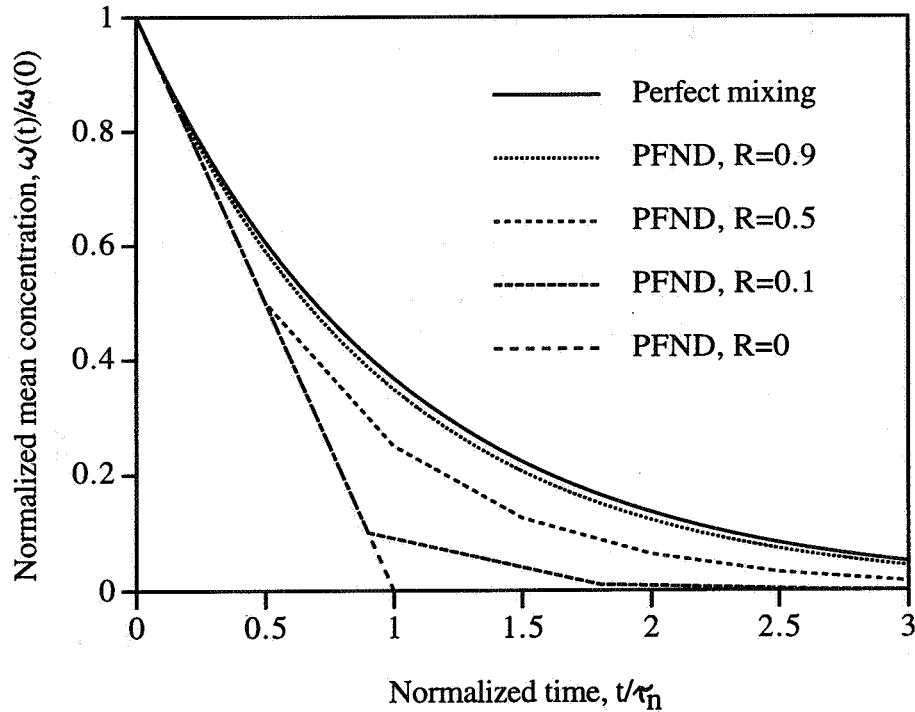


Figure 2: Normalized tracer gas decay for a perfect-mixing system and several PFND systems with different recirculation fractions.

normalized concentration decay curves for a perfect mixing system and several PFND

systems with different amounts of recirculation. As the recirculation fraction increases, the decay response moves closer and closer to the response of the perfect-mixing system.

In practice the air will not behave like it does in these abstract systems. Since the age of return air for systems with no recirculation is independent of the air distribution and is equal to T [2, 5], one can show that the age of air in any single-zone system is

$$\bar{a} = \bar{a}_{R=0} + \frac{TR}{1-R} \quad (6)$$

where $\bar{a}_{R=0}$ is the mean age of air that would exist if $R = 0$.

3 Air-Change Effectiveness

For a single-zone system, the absolute air-change effectiveness is defined as the ratio of the nominal turnover time to the mean age of air

$$\varepsilon_a \equiv \frac{\tau_n}{\bar{a}} \quad (7)$$

The absolute air-change effectiveness is the same as the air diffusion effectiveness defined in [2]. The absolute air-change effectiveness can be defined at a point by substituting the age at the point for the mean age. The absolute air-change effectiveness for a perfect-mixing system is 1, and is independent of the recirculation fraction. For a PFND system, the absolute air-change effectiveness is

$$\varepsilon_a = \frac{2}{1+R} \quad (8)$$

In general, the absolute air-change effectiveness is

$$\varepsilon_a = \frac{\tau_n}{\bar{a}_{R=0} + \tau_n R} \quad (9)$$

If there were no recirculation, then the absolute air-change effectiveness would provide a measure of the air distribution in the zone. Therefore, another air-change effectiveness measure, which will be referred to as the relative air-change effectiveness, is defined as the value of the absolute effectiveness that would have been calculated had the recirculation fraction been zero.

$$\varepsilon_r \equiv \frac{T}{\bar{a}_{R=0}} \quad (10)$$

For a single-zone system, it can be shown that

$$\varepsilon_r = \frac{\tau_n - a_s}{\bar{a} - a_s} \quad (11)$$

where a_s is the age of the supply air. Equation 11 demonstrates the similarity between the relative air-change effectiveness defined in this paper and the relative ventilation efficiency defined in [6]. As with the absolute air-change effectiveness, the relative air-change effectiveness can be computed at a point by substituting the age of air at a point for the average age of air in Equation 11. When there is no recirculation, the age of the supply air is zero, so the relative air-change effectiveness becomes the same as the

absolute air-change effectiveness. For single-zone PFND systems, the relative air-change effectiveness is equal to two regardless of the recirculation fraction. For perfectly mixed systems it is equal to 1.

It can be shown that for a single-zone system, the relation between ε_a , ε_r , and R is

$$\varepsilon_a = \frac{\varepsilon_r}{1 - R + \varepsilon_r R} \quad (12)$$

This relation is shown in Figure 3. The figure shows that as the recirculation fraction

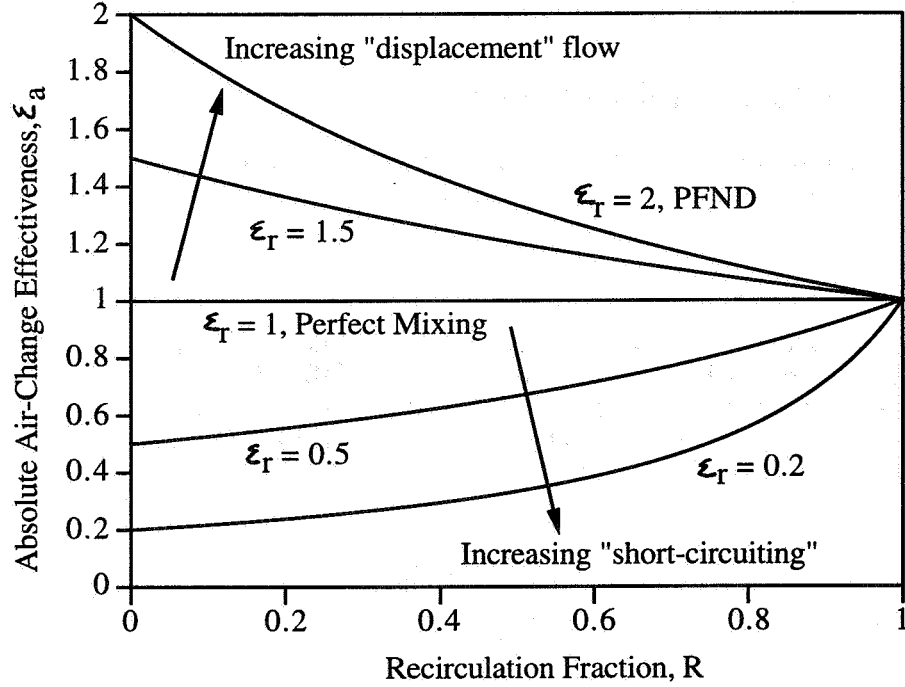


Figure 3: Schematic diagram of a single-zone ventilation system with recirculation and bypass.

approaches 1, ε_a approaches 1 regardless of the value of ε_r . It also shows that by itself, ε_a only provides a qualitative description of the air distribution pattern. If $\varepsilon_a < 1$, then one can say that there is some "short-circuiting," but one cannot say how much without knowing either ε_a and R or just ε_r .

Most buildings contain a number of zones that interact both through recirculation in an air-handling unit and directly through passages such as doorways. The theory described above can be applied to any multizone system in which the flow-weighted average age of the air entering and leaving the zones can be evaluated. For example, in Figure 4, the relative air-change effectiveness of zone 2 could be calculated as

$$\varepsilon_{r2} = \frac{a_{e2} - a_{s2}}{\bar{a}_2 - a_{s2}} \quad (13)$$

$$a_{s2} = \frac{f_{s1}a_{e1} + f_{s3}a_{e3}}{f_{s1} + f_{s3}} \quad (14)$$

where a_{e2} is the age of the air leaving zone 2, a_{e1} is the age of the air leaving zone 1

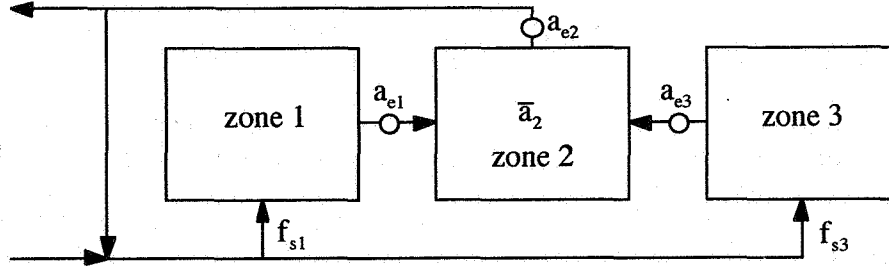


Figure 4: Schematic diagram of a multizone ventilation system with recirculation.

and flowing into zone 2, a_{e3} is the age of the air leaving zone 3 and flowing into zone 2. Note that the quantity in the numerator of Equation 13 could be computed from flow rate measurements as

$$a_{e2} - a_{s2} = \frac{M_2}{f_{s1} + f_{s3}} \quad (15)$$

Also note that ε_{r2} could not be calculated from ε_{a2} of zone 2 and R even if the age of air at the return grille in zone 2 were used to compute ε_{a2} because zone 2 is not directly supplied with air by the air-handling unit.

Sometimes ineffective ventilation is modeled by assuming that some of the supply air bypasses the breathing zone and passes directly to the return duct [3, 7]. This can be modeled by assuming that $S > 0$ in Figure 1. It can be shown that for this system

$$\varepsilon_a = \frac{1 - S}{1 - RS} \quad (16)$$

The absolute air-change effectiveness is a measure of how well outdoor air is utilized. In [3], a ventilation performance measure called ventilation efficiency is defined as

$$\eta = 1 - S \quad (17)$$

It can be shown that

$$\varepsilon_r = \eta \quad (18)$$

In other words, for a perfect-mixing system with bypass, the relative air-change effectiveness is the fraction of the supply air that is used to ventilate the zone. In general, the relative air-change effectiveness is a measure of how well supply air is utilized. A problem with the bypass model is that some ventilation systems make more effective use of ventilation air than perfect-mixing systems. When this happens, the bypass factor, S , must be negative. A negative value of S does not have a physical interpretation as does a positive value.

4 Discussion

Many published experiments on air-change effectiveness involve tests on systems that recirculate air. Typically the absolute air-change effectiveness is reported, and the tests are often carried out with large recirculation fractions (e.g., minimum outdoor air conditions). The theory presented above predicts that under these conditions the absolute air-change effectiveness will be close to unity. This prediction is consistent with many

published findings such as [8, 9, 10] although some experiments such as those described in [11] have shown that the absolute air-change effectiveness may be significantly less than one even with recirculation.

To demonstrate the significance of the theory presented above, the data reported in [11] are re-evaluated, and relative air-change effectivenesses are calculated. The article describes a tracer gas experiment on a zone in a multizone system which was divorced from the other zones. The HVAC system was operated in the heating mode, so the supply air was warmer than the zone air. Table 1 shows the values of the breathing-zone absolute air-change effectiveness reported in [11] and the relative air-change effectiveness calculated from information supplied in the article. The values of ε_r shown in the table

Table 1: Re-evaluation of the data presented in [11]

configuration	ε_a from [11]	ε_r
H1, ceiling supply & return	0.73	0.31
H2, ceiling supply & return	0.66	0.34
H3, ceiling supply, floor return	0.76	0.56
H4, ceiling supply, floor return	0.73	0.47
H5, “short-circuiting” design	0.57	0.44

were not calculated with Equation 11 even though the age of air at the supply, return, and breathing zone were all reported in the paper. This is because one of the configurations delivered air to a ceiling plenum rather than directly to the zone, and because there was some uncertainty about leakage of air into the return duct. Instead, ε_r was calculated as follows

$$\varepsilon_r = \frac{T}{a_b - a_p - a_s} \quad (19)$$

where T was calculated from the reported nominal ventilation rate, volume of the space (total volume minus the volume of the ceiling plenum) and recirculation fraction (67%), a_b was the age of air in the breathing zone, a_p was the additional age of the supply air induced by supplying the air to the ceiling plenum rather than directly to the room, and a_s was the age of the supply air, which was measured as it left the rooftop air-handling unit. The results reported in [11] indicate that the configuration designed to induce short-circuiting had the lowest absolute air-change effectiveness. However, by calculating ε_r , one can see that the system with ceiling supply and return induced more “short-circuiting” than the system designed to induce it. This is probably because the supply air velocities were higher with the system designed for “short-circuiting,” and the higher velocities probably entrained surrounding air and induced more mixing.

Another implication of the theory described in previous sections pertains to the use of experimentally-determined effectiveness parameters in controlling ventilation systems. It is proposed in [12] that values of ε_a determined from an experiment on the system be used to adjust the minimum outdoor air flow rate. This may cause the effective outdoor air flow rate to be too high or too low depending on the conditions of the test and the operational behavior of the system. For example, if the system tested in [11] had been a variable-air-volume (VAV) system, and if the value of ε_a calculated in test H1 were used to adjust the outdoor air flow rate, then if the system were to operate with 100% outdoor air and the adjusted minimum flow rate, the effective outdoor air delivery to the breathing zone would be only 42% of the design value because with 100% outdoor air would equal ε_r , which was 0.31 rather than 0.73.

5 Conclusions

1. Recirculation of air in buildings confounds the quantitative description of air distribution patterns when only absolute air-change effectiveness is reported.
2. This paper extends the air-change effectiveness theory to single-zone systems with recirculation and multizone systems with or without recirculation by accounting for the age of the supply air.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

SIMULATIONS AND TRACER GAS MEASUREMENTS OF AIR CHANGE AND AIR EXCHANGE EFFICIENCY

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(Full paper not available at time of print - see supplement to proceedings)

17th AIVC Conference

Optimum Ventilation and Air Flow Control in Buildings

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Abstract

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This paper concerns about simulation of air flow and measurements of air change and air exchange efficiency in a room. The age of air theory has been implemented in the CFD simulation program, and tracer gas measurements are compared to the simulation results. In the simulations, particles with the same physical properties as air, are initially distributed uniformly in the room. The ventilation is then activated, and the particle concentration decay is simulated. The transient particle concentration is used for determination of the age of air, and finally, the air exchange efficiency. Tracer gas measurements are performed under equal conditions in a similar room. The comparisons between simulations and measurements show reasonable agreement.

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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HOW EFFECTIVE IS NATURAL VENTILATION? - A STUDY OF LOCAL MEAN AGE OF AIR BY MODELLING AND MEASUREMENT

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HOW EFFECTIVE IS NATURAL VENTILATION? - A STUDY OF LOCAL MEAN AGE OF AIR BY MODELLING AND MEASUREMENT

by R R Walker, S D Hayes and M K White

SUMMARY

A condition often assumed when designing a naturally ventilated building is where air enters at low level and leaves at high level due to the stack effect. It then follows that, at upper levels, the air may be relatively 'stale' since it has previously passed through the lower storeys. An analogous situation may arise when wind is blowing, in which the air entering through the windward face becomes stale as it passes through the building to the downwind sections.

It is not well understood how ventilation may, in reality, be affected by this. To address these issues, this report describes a modelling approach using BREEZE and complementary measurements using the Passive Tracer Gas technique to study local ventilation rates in multiroomed office buildings. Calculations show that simple ventilation flow rates, as conventionally calculated at the design stage, cannot be relied upon to indicate the true 'freshness'. Measurements show that effective ventilation can be less than current minimum occupancy requirements.

INTRODUCTION

This report describes work carried out as part of a broader study to investigate ventilation efficiency and indoor air quality strategies to minimise energy liabilities in naturally ventilated office buildings. The study was funded by the UK Department of Environment's Energy Related Environmental Issues research programme.

In the design of a naturally ventilated building with many stories, a condition often assumed is where air enters at low level and leaves at high level due to the stack effect. It then follows that, at upper levels, the air may be relatively 'stale' since it has previously passed through the lower storeys. An analogous condition occurs where wind is blowing, and the air entering through the windward face becomes stale as it passes through the building to the downwind sections. Internal partitions (eg closed doors) may also affect the true adequacy of the supply of outside air. At present, it is not well understood how the effectiveness of ventilation may be affected by these factors, and how improved design may avoid the possible problems. To address these issues, BRE has been investigating^{1,2,3} the possible application of the concept of 'ventilation efficiency'⁴ to naturally ventilated buildings.

The objectives of the current work were to apply the concept of ventilation efficiency to examples of typical, multi-story naturally ventilated office buildings and to:

- assess the distribution of 'fresh' (outside) air supply'
- assess the adequacy of ventilation for the intended occupancy, and
- explore the possible implications for the design and operation of buildings and strategies to ensure adequate air quality, whilst avoiding excessive ventilation.

Two complementary approaches were followed. Computer software for modelling airflow and contaminant movement (BREEZE)⁵ was used to calculate local ventilation rates in a multiroomed complex naturally ventilated office building during design and construction.

Complementary to this, the Passive Tracer Gas Technique developed at BRE was applied to measure local ventilation rates in two multistorey office buildings while in use.

MODEL STUDY

The building used for the model study had recently been constructed and ventilation modelling had previously been carried out to provide input to decisions during the design and construction. It incorporated the following key architectural features typical of many recent designs for natural ventilation in the UK:

- a central atrium or covered street, as a means of providing ventilation to adjacent offices either side via windows,
- deep offices with ventilation openings to outside on only one face, with the other face opening to an internal street or atrium, and
- multistorey, with significant vertical ventilation routes.

It consists of an east and west wing linked in a line to a central section. A covered 'street' forms an atrium between the north and south buildings in each section. There are five storeys along the north side of the street and four storeys along the south side. The offices are mainly open plan throughout, except on the fourth and fifth floors where some cellular offices have been considered as options at the design stage. Although the whole building was modelled, this study focuses on the west wing for the purposes of simplicity and clarity. A perspective view (model), of the whole building, and schematic floor plan and cross-section of the west wing are shown in Figure 1. Ventilation is provided by windows to outside and to the 'internal' covered street. Trickle ventilators are installed in each external window to provide controllable background ventilation during cooler months, when windows are expected to be closed.

Approach

The model study approach was based on a protocol previously developed² (summarised below) for measuring local mean ages of air using the Passive Tracer Gas Technique. In this, constantly emitting tracer gas sources are distributed evenly throughout a building to give a uniform distribution of tracer emission rate per unit volume. The local mean age is then determined from measurements of the local concentration of tracer gas and its known emission rate.

For the model study, the multi-cell ventilation and contaminant prediction model BREEZE was used to predict the bulk air exchanges between the various ventilated spaces in the building. A uniform 'tracer' emission rate was simulated using the contaminant prediction facility and steady-state tracer gas concentrations (C) were calculated in selected rooms. The local mean age τ_j of the air in room ' j ' was calculated as follows:

$$\tau_j = C_j / (S / V) \quad (1)$$

where S is the source emission rate and V is the room volume.

It is convenient to define a local ventilation rate², analogous to the conventional air change rate, as follows:

$$r_j = 1 / \tau_j \quad (2)$$

We have previously proposed a quantity called the equivalent mean ventilation flow rate², defined as equal to the equivalent flow rate of fresh air which would maintain the observed concentration of tracer. This is obtained by multiplying the local ventilation rate by the volume of the space to which it applies. It is convenient to express this as a normalised flow rate per unit floor area, Q_{eq} , eg. in units of litres per second per square metre ($l\ s^{-1}\ m^{-2}$). If the local ventilation rate is expressed in h^{-1} , and the room floor to ceiling height H is in metres, then the normalised equivalent flow rate is given by the following expression:

$$Q_{eq(j)} = r_j \times H / 3.6 \quad (3)$$

Test cases

The study focused on how local ventilation rates might vary throughout the building and whether they would meet the minimum requirements in conditions when windows are closed.

Three basic cases were considered as follows:

- (i) buoyancy dominated - no wind, with the air temperature outside (T_o) lower than inside'
- (ii) buoyancy and wind combined - prevailing wind, with air temperature outside (T_o) lower than inside, and
- (iii) wind dominated - prevailing wind, but with equal air temperatures inside and outside.

In each of the above cases both average and extreme weather conditions for the south of England⁶ were modelled, although only the case with average conditions are reported here. The internal air temperature was taken to be either 18°C or 21°C.

The possible effect on local ventilation rates due to closing the internal doors to the cellular offices was also considered. A small gap of 0.041 m^2 was assumed around each closed door, and the calculations carried out for average conditions in the combined wind and buoyancy case as above.

Results

The results were compared with recommended and minimum ventilation rates⁷ expressed per person and per unit floor area for occupancies of 10 m^2 and 14 m^2 , taken to represent minimum and recommended values⁸, as shown in Table 1.

Table 2 shows the result for the three cases. Figure 2 shows the local ventilation rates at increasing storey height above ground. Where wind is blowing, zone numbers corresponding to the left of Figure 1b have at least one wall generally facing windward.

For the buoyancy dominated case, it can be seen from Table 2 that the total flow rates meet or exceed the minimum recommended rate ($0.35\ l\ s^{-1}\ m^{-2}$) on all but the third level. Ventilation is at a minimum on the third level, which is approximately at the neutral plane. Below this level, air enters the offices from outside. Above it, air enters from the internal street and leaves through the external facade.

As expected, the equivalent ventilation rates are highest and similar to the total rates on the lower storeys, where air enters from outside. Equivalent rates are small near the neutral level

(at mid-height) where air exchanges are at a minimum. On the two levels above the neutral plane, equivalent ventilation rates are approximately one-third of the total rates and are less than the minimum requirement. This is because the supply air is generally from within the building.

In the combined case of buoyancy and wind, the total flows achieve the minimum throughout the building. The equivalent flows, however, reveal a more complicated picture. Rooms with at least one windward wall generally have equivalent ventilation rates which meet the minimum requirement. Effective ventilation is not achieved in a significant number of rooms at all levels. There were no significant changes in these results when calculations were repeated with doors closed (but for a small gap).

The situation deteriorates further in the wind dominated case. Although total ventilation rates appear to be adequate, equivalent rates show that ventilation is below requirements on the lower three storeys and in the cellular office on the top storey.

FIELD MEASUREMENTS IN TWO OFFICE BUILDINGS

To complement the model study, measurements were carried out in two naturally ventilated office buildings as described below, using a protocol for the application of the Passive Tracer Gas Technique developed at BRE^{2,3}. The measurements are described only in brief here to present the key results; more details will be published separately⁹.

The general principle is the same as used for the model simulations. Passive tracer gas permeation sources are placed in rooms and corridors throughout the building to achieve a source emission rate which is broadly in proportion to floor area (within about 10%). Diffusion-type air samplers are subsequently placed in a selection of typical rooms, and the average concentration (C) is determined from the mass collected over the measurement period (days or weeks) and the known diffusive sampling rate. The local mean age, τ_p , within a room of volume V is then calculated using equation (1).

Procedure

Tests were carried out in winter to measure the average ventilation rates at different locations in the buildings over a range of weather conditions. Samplers were placed in triplicate (to check accuracy) at 25 locations in Building A and 24 in Building B. These were collected for analysis after four weeks in Building A and after three weeks in Building B. A limited set of preliminary measurements were also carried out over 48 hours in Building A at five locations.

Building A

Building A was a fairly typical 1960's narrow plan, four storey office block in a suburban location north of London (Figure 3). A building of similar height stood immediately to the north, whilst to the east the ground was built-up and steadily rising and open to the west and south. The building was primarily naturally ventilated, but with mechanical extract in some photographic studio rooms on the third floor. The openable windows were mainly side hung casements, but with some louvred units in stairwells and toilets. Offices were mainly cellular (around 13 m²), but with some open plan rooms (approximately 40 to 90 m²).

Building B

Building B was a four storey office block, built in the mid eighties in an urban location in the midlands. Built over an existing single storey office block, it was basically rectangular in plan (Figure 4) with each storey overhanging the lower. It is unusual in that it was naturally ventilated from the perimeter but deep in plan, with no central atrium for additional ventilation. To the north and west were a pair of two-storey buildings whilst the other directions were unobstructed.

Offices were arranged around a central core (for services and fire escape) and an open stairwell surrounded by open circulation or meeting areas. An almost isolated mezzanine floor was sandwiched between part of the ground and second floors. The ground floor consisted of an open plan office and a large, tall reception area. The first floor was open plan. The second floor was mainly open plan but with some cellular offices. The third floor consisted of mainly single or double occupancy offices, some of which were air-conditioned, with a few large rooms for occasional use. Window bays consisted of an inwardly opening top 'hopper' pane above a light shelf and a lower 'tilt or turn' section.

Results

Building A

As before, the results are compared with the recommended ventilation supply rates (Table 1). For Building A, Table 3 shows the measured local ventilation rates, normalised equivalent ventilation supply rates and nominal floor areas for each room for both the preliminary and long term tests. The average meteorological data are also given for the same period. It may be seen that long term averaged local ventilation rates were spread over an order of magnitude, ie 0.3 to 3.6 per hour. The standard error of measurement is better than 15% except where indicated.

The variation in rates appeared to be dominated by window opening behaviour. It was not expected that room size should be a significant factor, since most rooms differed only in their length, along which openable windows were generally regularly spaced. Results for the large cross-ventilated rooms 012, 222b and 229 were not exceptional.

Corresponding to the above, equivalent supply rates were in the range 0.25 to $2.8 \text{ l s}^{-1} \text{ m}^{-2}$ and mostly just above the minimum requirement (Table 1). Exceptions were some small rooms on the south face of the second floor with double occupancy. The low ventilation rate of $0.25 \text{ l s}^{-1} \text{ m}^{-2}$ in the large room 229 was adequate for the low occupancy (two persons) noted.

Room 001 showed a relatively high ventilation rate. This was used by several office messengers and was fitted with a number of window slats which were usually open. On the third floor, where ventilation rates were generally high, occupants were noted to frequently open windows, perhaps to dilute odours from film processing activities carried out on that floor.

Ventilation rates in the 48-hour tests were broadly half to two-thirds of the long term values. This may be consistent with limited CO₂ measurements which showed peaks exceeding 1400 ppm over short periods. This level corresponds to a ventilation rate of 5 l s^{-1} per person for an average person in office type activity¹⁰.

Building B

Results for Building B are shown in Table 4. Time-average local ventilation rates were in the range 0.26 to 0.78 per hour and equivalent air supply rates in the range 0.22 to 0.64 l s⁻¹ m⁻². Ventilation rates were generally lower on the second floor. It is not clear whether this could be related to a possible neutral level for stack-driven ventilation, as no supplementary measurements were carried out to investigate this.

The observed floor area per workstation was approximately 10 m² on the ground and second floors respectively. The minimum recommended ventilation rate of 0.5 l s⁻¹ m⁻² was met on the ground floor but not on the second. First floor occupancy was approximately 18 m² per person, for which the minimum recommended ventilation rate of 0.28 l s⁻¹ m⁻² was exceeded. On the third floor, which mainly contained single and double occupancy rooms and a large 'boardroom', the minimum recommended ventilation rate per person was either met or exceeded.

CO₂ concentrations recorded near location 2.1 frequently exceeded 1400 ppm (corresponding approximately to 5 l s⁻¹ per person). This is consistent with the measured ventilation rates which were between one-half to four-fifths of the recommended minimum for the given occupancy.

CONCLUSIONS

The objectives of this study were to apply the concept of ventilation efficiency to assess the effectiveness of natural ventilation in typical examples of multi-storey office buildings, and to explore the possible implications for the design, operation and ventilation strategies in buildings. Both a computer modelling and experimental approach were followed. The local mean age of air was determined which was used to calculate an equivalent ventilation rate of 'fresh air' in l s⁻¹ m⁻².

The model study was carried out for average meteorological conditions in three cases, (i) buoyancy dominated, (ii) combined wind and buoyancy driven and, (iii) wind dominated. In all cases it was predicted that ventilation, as expressed by equivalent ventilation rates, may be expected to be less than currently recommended minimum ventilation supply rates for occupants in a significant number of rooms. In general, however, the simple ventilation air flow rates were greater than the equivalent flow rates and, consequently, were not an accurate guide to the true adequacy of ventilation for diluting internally generated contaminants.

Local ventilation rates were measured in two multi-storey office buildings in winter using a protocol for measuring local mean ages developed for the Passive Tracer Gas Technique. The time-averaged equivalent ventilation rates were shown to vary by over an order of magnitude throughout one building, but mostly just above the minimum requirements for the occupancy. Equivalent ventilation rates in the second building varied by over a factor of three and achieved the minimum requirements on all but the second storey.

Taken together, these results suggest that the concept of ventilation efficiency, applied in the form of the local ventilation rate and equivalent ventilation rate, provides a valuable measure of the ventilation performance throughout naturally ventilated buildings. Both measured and predicted equivalent ventilation rates showed that ventilation can be less than current minimum

occupancy requirements. Simple ventilation flow rates, as conventionally calculated at the design stage, cannot be relied upon to indicate the true 'freshness' or capability of the ventilation supply for diluting internally produced contaminants.

Further full scale measured data are necessary to establish a representative view of the ventilation performance of the UK office building stock, together with model studies to understand the implications for design in more general terms. However, the following preliminary recommendations are suggested to help avoid problems identified above:

- (i) The optimum arrangement of inlets and outlets (ventilation strategy) for background ventilation should be sought.
- (ii) The design should consider the meteorological conditions likely to occur for significant numbers of days (the case when there is no wind cannot be generalised as the worst case for the UK).
- (iii) To ensure effective control, specialist attention should be given to ensure background leakage is small compared to purpose provided ventilation.
- (iv) An effective control strategy should be evaluated for opening small vents or fine control of small window openings.
- (v) It may be necessary to consider special inlets and outlets (eg ducted outlets via towers etc.) to ensure that air flows in the required direction within the building.
- (vi) Consider sensors (eg CO₂) to control openings automatically according to occupancy requirements if necessary.

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(l s ⁻¹)	Per person	Per m ² floor area	
		(10 m ²)	(14 m ²)
Recommended:	8	0.8	0.57
Minimum:	5	0.5	0.35

Table 1. Required ventilation rates (l s⁻¹) per person and per unit floor area with either 10 m² or 14 m² per occupant.

Location:	1.2	1.4	2.2	2.4	3.2	3.4	4.2	4.3	4.5	4.6	5.2	5.3
Area (m ²)	910	839	900	830	900	830	150	750	680	150	750	150
(i) Buoyancy Only												
r (h ⁻¹)	0.77	0.72	0.56	0.54	0.11	0.11	0.19	0.22	0.22	0.19	0.22	0.17
Q _{eq} (ls ⁻¹ m ²)	1.01	0.94	0.66	0.63	0.12	0.12	0.22	0.26	0.26	0.22	0.25	0.20
Q _m (ls ⁻¹ m ²)	0.95	0.88	0.59	0.60	0.06	0.06	1.60	0.76	0.74	1.72	1.03	0.53
acr (h ⁻¹)	0.73	0.68	0.51	0.51	0.05	0.05	1.37	0.65	0.64	1.47	0.88	0.45
(ii) Combined Buoyancy & Wind												
r (h ⁻¹)	0.18	0.50	0.16	0.41	0.14	0.22	0.86	0.33	0.45	1.25	0.24	0.17
Q _{eq} (ls ⁻¹ m ²)	0.24	0.65	0.19	0.48	0.17	0.25	1.00	0.38	0.52	1.46	0.28	0.20
Q _m (ls ⁻¹ m ²)	0.42	0.61	0.80	0.45	0.96	0.51	1.17	0.38	0.88	1.29	0.62	0.36
acr (h ⁻¹)	0.32	0.47	0.68	0.38	0.82	0.44	1.00	0.32	0.75	1.11	0.53	0.30
(iii) Wind Only												
r (h ⁻¹)	0.15	0.21	0.17	0.22	0.19	0.24	1.18	0.41	0.67	1.86	0.60	0.24
Q _{eq} (ls ⁻¹ m ²)	0.19	0.27	0.20	0.25	0.23	0.28	1.38	0.47	0.78	2.17	0.70	0.28
Q _m (ls ⁻¹ m ²)	1.06	0.53	1.00	0.54	1.01	0.54	1.36	0.45	0.93	2.13	0.76	0.34
acr (h ⁻¹)	0.82	0.41	0.86	0.46	0.87	0.46	1.16	0.38	0.80	1.83	0.65	0.29

Note: Q_m = total inflow divided by zone area; acr = air change rate (total inflow / zone volume)

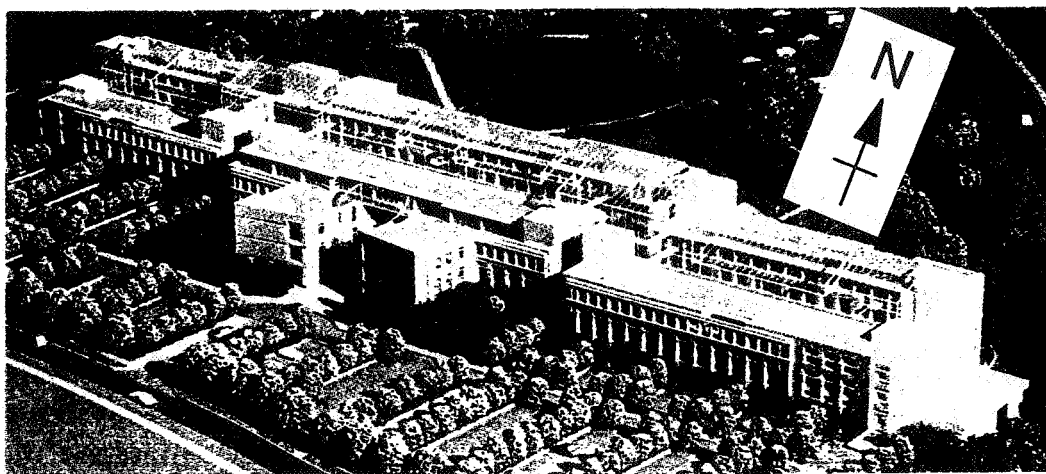
Table 2: Equivalent and total ventilation rates calculated from model study

Location:	323	319	311	309a	309	307	303	229	225	224	222b	219	218	214	210	206	204	202	
Area (m ²)	13	13	10	11	10	29	11	81	21	62	91	40	13	13	13	13	30	16	
Q _{eq} (ls ⁻¹ m ²)	0.35	0.46	1.4	1.3	2.8	0.52	0.93	0.36	0.28	0.41	0.51	0.53	0.60	0.46	0.43	0.42	0.48	0.50	
r (h ⁻¹)	0.43	0.59	1.8	1.7	3.6	0.66	1.2	0.46	0.36	0.53	0.66	0.69	0.77	0.59	0.55	0.54	0.62	0.64	
Location:	022	016	012b	012	001	OS	ON	48 hr tests:				307	229	222b	206	012			
Area (m ²)	45	13	13	129	16	31	16	(ON = corridor north)				28.6	80.7	91	13	129			
Q _{eq} (ls ⁻¹ m ²)	0.25	0.44	0.30	0.33	1.8	2.0	1.1	(OS = corridor south)				0.23	0.21	0.20	0.21	0.17			
r (h ⁻¹)	0.32	0.57	0.38	0.41	2.3	2.6	1.4					0.30	0.27	0.26	0.27	0.22			
Notes: room prefix denotes storey; ceiling height 2.8 m; average test conditions as follows:-																			
(i) 4 wk test: T _i = 17.9 °C, T _o = 6.7 °C, ws = 4.2 ms ⁻¹ , wd = 210°																			
(ii) 48 hr test: T _i = 19.3 °C, T _o = 4.7 °C, ws = 4.4 ms ⁻¹ , wd = 210°; where																			
Q _{eq} = equivalent flow rate, normalised by floor area; r = 1/τ; τ = local mean age;																			
T _i , T _o = air temperature (i) inside, (o) outside; ws/wd = wind speed/direction (clockwise from N)																			

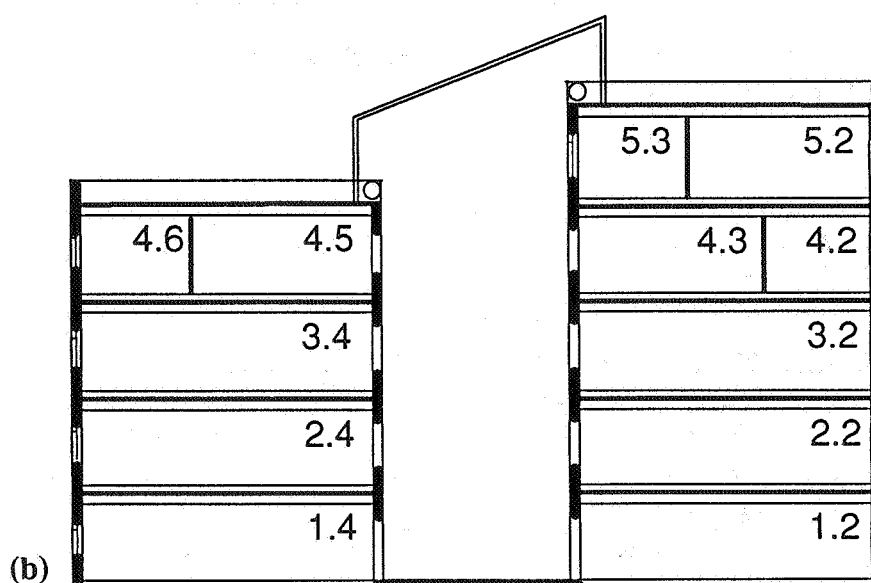
Table 3. Winter - Building A ; measurements over 4 week period (2.2.95 to 2.3.95) and 48 hr period (25.1.95 to 27.1.95)

Location:	3.1	3.2	3.3	3.4	3.5	3.6	3.7	3.8	3.9	3.10	2.1	2.2	2.3	2.4	2.5
Area (m ²)	49	21	-	39	31	43	40	40	38	124	(---312---)		42	25	95
Q _{eq} (ls ⁻¹ m ²)	0.32	0.34	'lost'	0.24	0.41	0.46	0.54	0.47	0.57	0.37	0.22	0.27	0.38	0.27	0.27
r (h ⁻¹)	0.39	0.41	'lost'	0.3	0.5	0.55	0.65	0.57	0.69	0.45	0.26	0.32	0.46	0.33	0.33
Location:	1.1	1.2	1.3	1.4	1.5	1.6	1.7		0.1	0.2					
Area (m ²)	(-----444-----)					26		(---134---)		(--- --) denotes common					
Q _{eq} (ls ⁻¹ m ²)	0.48	0.64	0.50	0.46	0.47	0.34	0.51	0.54		0.58	open plan area				
r (h ⁻¹)	0.57	0.78	0.61	0.55	0.57	0.41	0.62	0.66		0.71					
Notes: room prefix denotes storey; ceiling height 3.0 m; average test conditions as follows:-															
T _i = 23.0 °C, T _o = 5.6 °C, ws = 4.5 ms ⁻¹ , wd = 240°; where															
Q _{eq} = equivalent flow rate, normalised by floor area; r = 1/τ; τ = local mean age;															
T _i , T _o = air temperature (i) inside, (o) outside; ws/wd = wind speed/direction (clockwise from N)															

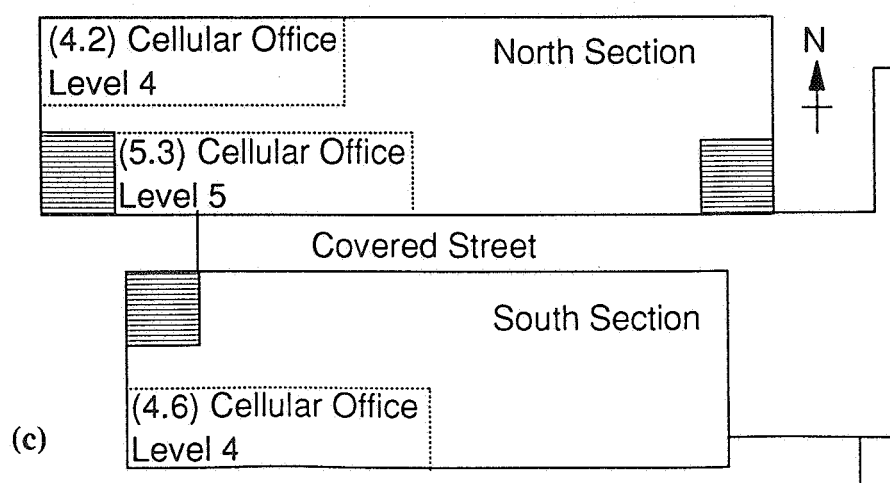
Table 4. Winter - Building B; measurements over 3 week period (7.3.95 to 28.3.95)



(a)



(b)



(c)

Figure 1. (a) Perspective view (model) of whole building, (b) cross section showing zone numbers, and (c) floor plan, of west wing.

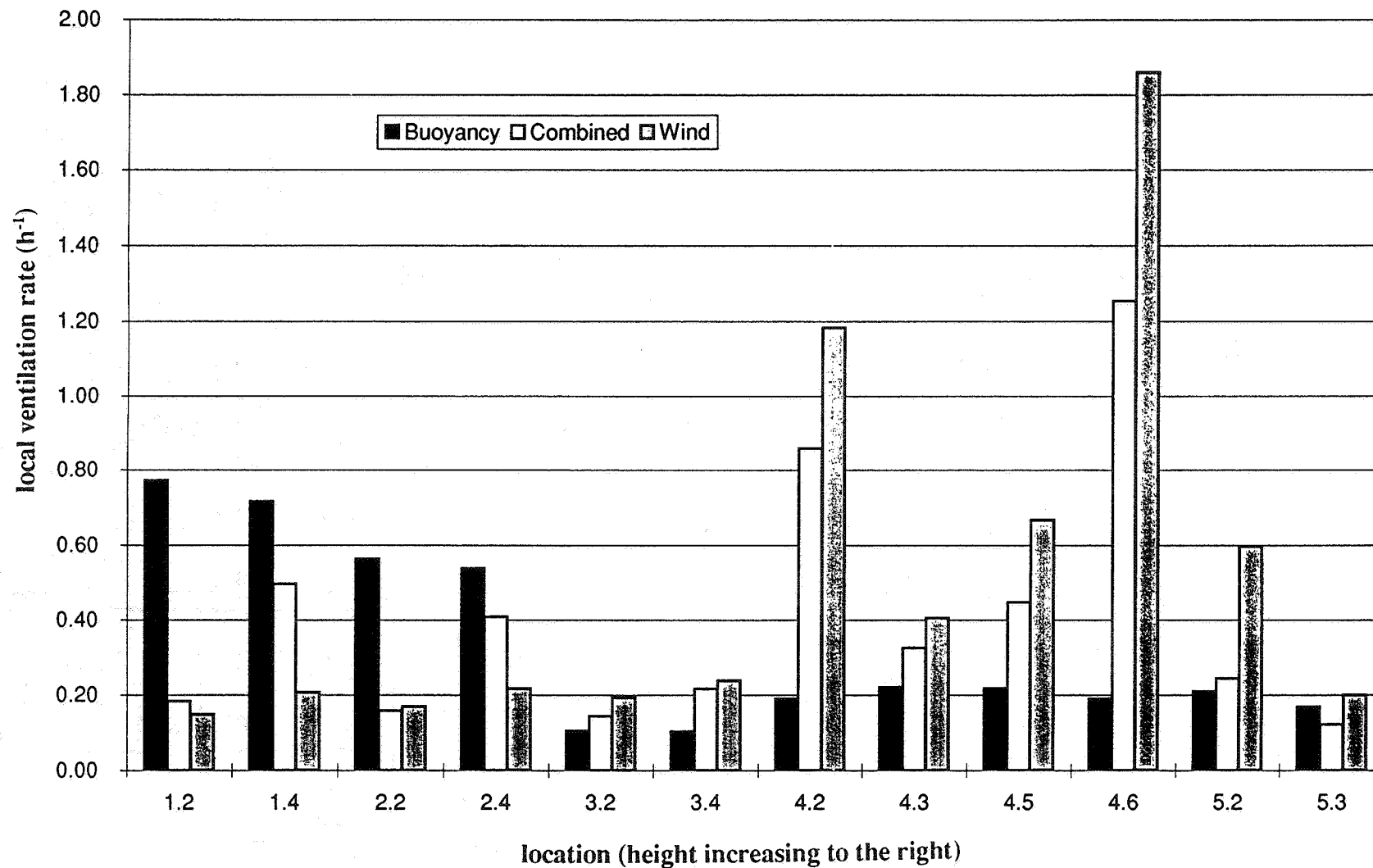


Figure 2. Local air change rate for each zone, shown with height increasing to the right.

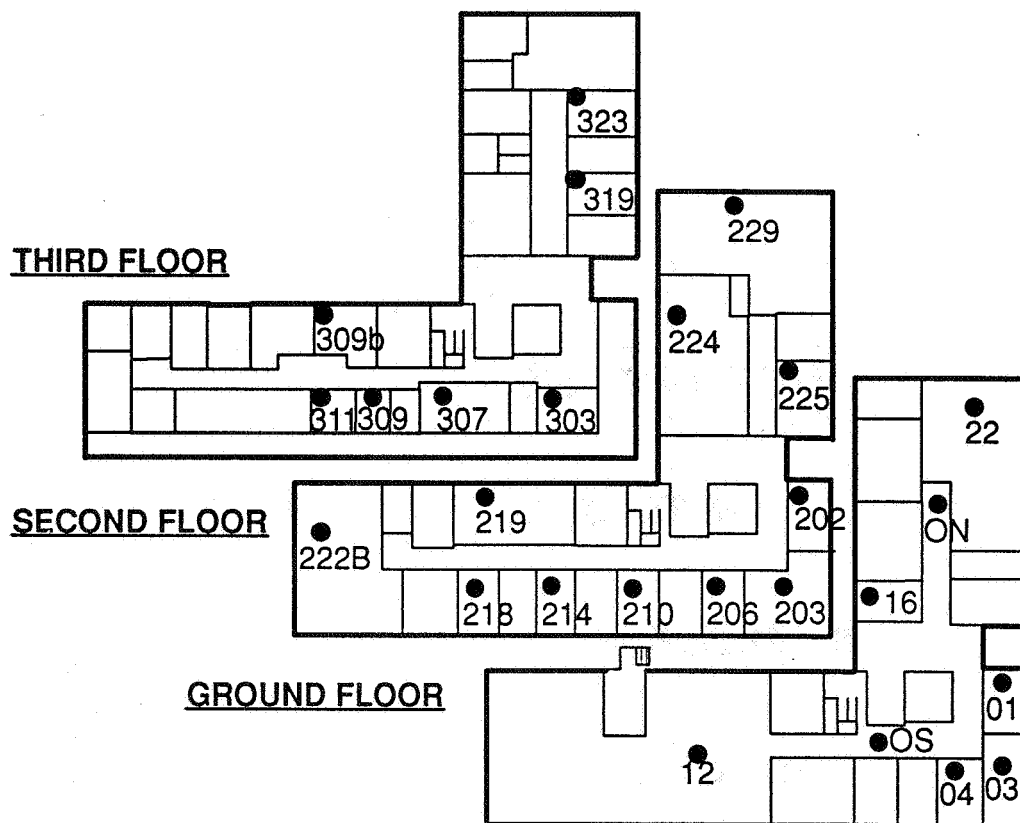


Figure 3. Building A; floor plans and measurement zones.

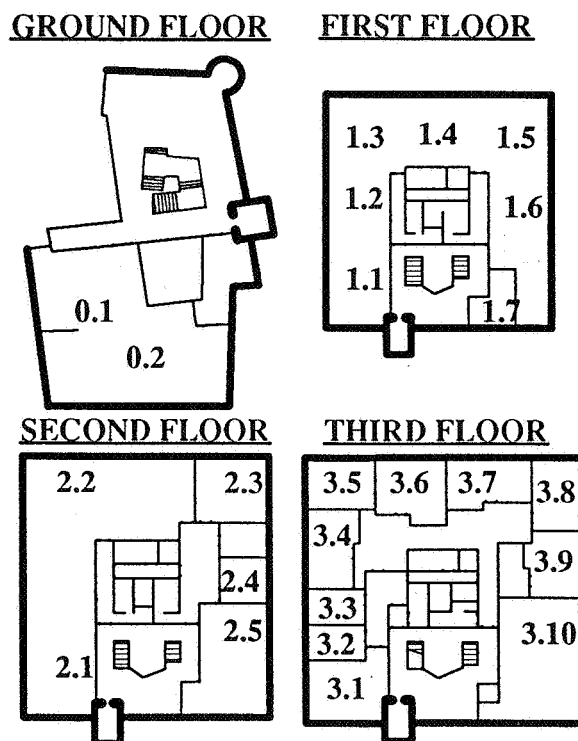


Figure 4. Building B; floor plans and measurement zones.

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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The Effect of External Atmospheric Pollution on Indoor Air Quality

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ABSTRACT

This paper reports the findings of a pilot field study carried out to investigate the internal and external air pollution levels of two adjacent buildings, one naturally-ventilated and the other air-conditioned in an urban area, to investigate their relative attenuation of external pollution levels and to compare internal levels with existing air quality guidelines.

Concentration levels of sulphur dioxide, nitrogen oxides, carbon monoxide and carbon dioxide were monitored. Simultaneously, measurements of ventilation rates within the buildings and periodic video recordings of the traffic were also carried out.

As expected the concentrations of external pollutants in the buildings followed the daily external variation, but at reduced levels. Generally, pollutant levels were higher in the naturally ventilated building than in the air-conditioned building. However, on a number of occasions, combustion products from heating boilers were entrained into the air-conditioned building via the high level air intake of the ventilation system raising the levels of nitrogen oxides, sulphur dioxide and carbon dioxide inside the building to higher than those found externally.

A comparison of the results with existing air quality guidelines or standards for exposure showed that in both buildings the level of contamination was less than the relevant standard, except during a limited period at the weekend when combustion products were possibly entrained into the air-conditioned building. There is thus no clear distinction between the two types of ventilation strategies in terms of providing adequate indoor air quality to the occupants of the buildings.

1. BACKGROUND

Increased concerns over the adverse environmental impact of high energy usage required for mechanical ventilation and air-conditioning has encouraged the design and construction of energy-efficient buildings with many of them employing natural ventilation strategies. However, in urban areas and city centres where external air pollution levels are relatively high, it is usually assumed that natural ventilation may not be able to provide adequate indoor air quality. Mechanical ventilation and air-conditioning systems are thus being installed to 'clean' the incoming air, even though there is evidence that such systems do not always provide clean fresh air to the occupants of the building (1).

The air quality in non-domestic buildings has been widely studied in the context of contaminants arising from internal processes or emanating from materials within the building. However, there have been comparatively few studies of the internal air quality of a building arising from the normal occurrence of external air pollution and the supply of this air to the occupants. Furthermore, there have been few attempts to relate this internal air quality to accepted standards of exposure to known contaminants.

A number of guidelines or standards exist which deal with human exposure to contaminants which are potentially injurious to health. In the UK these are contained within the Control of Substances Hazardous to Health (COSHH) Regulations 1988 (2) and the Health and Safety Executive (HSE) Occupational Exposure Limits EH40/95 (3). The relevant standards are

expressed in terms of concentrations of pollutant to which the working population may be exposed for the relevant periods, (eg. an 8-hour day, a 15-minute period or an absolute peak value) to avoid any risk to their health. Standards which deal more explicitly with air quality for the general populace are those of the World Health Organisation (WHO)(4), which deal with both indoor and outdoor air quality and those of the UK's Department of the Environment Expert Panel on Air Quality Standards (EPAQS), which has recently begun to publish reports on the health effects of certain pollutants and recommends air quality standards for these. Other more stringent guidelines are those which are included in the UK's Her Majesty's Inspectorate of Pollution (HMIP) Technical Guidance Note (5). The European Community (EC) also sets guideline or limit values for some pollutants (6).

2. CURRENT STUDY

This pilot study was carried out to investigate the following:

- internal and external pollution levels of a naturally ventilated and an air-conditioned building in close proximity to one another,
- their relative attenuation of external pollution levels, and
- a comparison of the levels recorded indoors with currently available air quality guidelines.

Previous work in this area is limited. Turner and Binnie (7) studied the CO levels in a number of naturally and mechanically ventilated buildings and found that externally produced CO was more prevalent in the mechanically ventilated buildings than those naturally ventilated. This was thought to be a consequence of ingress into the air handling units of contaminated air from underground car-parks. A study by Phillips, et al (8), of four naturally ventilated buildings, concluded that the air change rate was the determining factor for air quality; the greater the supply of external air the greater the presence of external pollutants indoors.

However, Ekberg (9) showed that it is unlikely that the relationship is related to the air change rate alone since there is the potential for sinks and sources within the building. He also showed that the effect of the rapidly changing concentrations of external pollutants was important in obtaining a more complete understanding of the relationship between indoor and outdoor air quality and that short term peaks in the concentration of external pollutants are significantly affected by the general response of the building. This is also highlighted in Treple's (10) studies of ventilation strategies in cases of external pollution events.

Although evidence from recent buildings employing natural ventilation as a design strategy (rather than simply providing openable windows) shows that acceptable ventilation rates can be achieved without resort to mechanical ventilation (11), the level of externally generated pollution experienced in these buildings has not been considered directly with reference to indoor air quality and air quality standards. Thus to enable development of suitable design guidance for low energy ventilation of non-domestic buildings in urban areas, the manner in which buildings can attenuate the effects of external pollution levels to achieve adequate indoor air quality needs to be known.

3. EXPERIMENTAL SETUP

3.1 Buildings

Two buildings, one naturally-ventilated and the other air-conditioned and located adjacent to each other on an eight-lane major road in a major urban centre (Figure 1) were selected for investigation. The naturally-ventilated building was a four storey building with openable vertical sash windows and secondary glazing. The measurements were made in a ground floor office with windows facing onto the main road on one side and an internal court-yard on the other.

The air-conditioned building was ten storeys high with a facade which was mostly sealed although there was a limited number of unused openable windows. The third floor open-plan office was chosen for monitoring because the lower floors were recessed from the road and shielded by the access stairs to a foot-bridge. The mechanical ventilation system drew external air from the tenth floor level via the roof-top plant room. The air was then filtered and heated prior to distribution to the ceiling voids for terminal re-heat and cooling. There was no heat recovery. A radiator system also provided heating to the offices. The mechanical ventilation system was taken as found and there were no reported problems. Its performance was assessed by the measurement of the air change rate as part of this pilot study.

The offices in both buildings were in normal use with variable occupancy and normal office activities. Both offices had a no-smoking policy and there were no gas appliances or any other significant internal sources of the measured pollutants apart from metabolic CO₂ from the occupants.

3.2 Measurements

Measurements of the following were made:

- sulphur dioxide (SO₂)
- carbon monoxide (CO)
- oxides of nitrogen (NO_x)
- carbon dioxide (CO₂)
- building air change rates
- traffic density

Each building was provided with a set of high quality gas analysers and the outputs from these instruments were recorded at five minute intervals for the whole of the monitoring period (13 to 20 February 1996) using a data-logging system. By using a switchable pump sampling system, the instruments in the naturally ventilated building were also used for measuring the external levels of the pollutants. They were calibrated immediately prior to the monitoring and the instruments 'zeroed' on alternate days by using a supply of uncontaminated air. The building air change rates were measured by using the conventional technique of observing the decay of an injected tracer gas (sulphur hexafluoride) seeded into the areas of interest. During unoccupied periods, the decay of CO₂ generated by the occupants previously in the building, was also used as a measure of the air change rates. Traffic densities on the main road were taken by periodic direct observation and by using video recording. The wind speed and direction were obtained from the local meteorological site.

4. RESULTS

4.1 Pollutant Concentrations

Measurements were carried out for a seven day monitoring period (12). However, for clarity of presentation, and to show interesting events which occurred over the weekend period concentration results for the various gases for Friday to Monday inclusive of a weekend only are shown in Figures 2-6.

Over the complete monitoring period, there were times when no significant levels of pollutants were recorded - either outside or inside the buildings. The buildings appeared not to retain the external pollutants for long periods. The ability of the buildings to attenuate the external pollutants is shown in most cases by the peak internal concentrations being lower than that recorded externally. Furthermore, it is apparent that neither building reacted to the rapid and sudden fluctuations seen in the external levels. The buildings tended to smooth out the external pollution levels over a time period of about an hour.

For occupants, the period of interest is the occupied day of 08:30 to 17:30, during the working week. Table 1 shows the ratio of the internal to external pollution concentration for this period. As a first approximation, it is suggested that this can be used as a measure of the effectiveness of each of the buildings in attenuating the external pollutants.

Table 1. Concentrations of Measured Gases for the Occupied Periods for Monday to Friday.

Measured gas	External		Internal							
			Naturally ventilated				Mechanically ventilated			
	Peak	Mean	Peak	Mean	Internal/External Ratio		Peak	Mean	Internal/External Ratio	
					Peak	Mean			Peak	Mean
CO (ppm)	9.6	1.8	4.1	1.7	0.4	1.0	3.0	0.9	0.3	0.5
CO ₂ (ppm)	478	366	1197	650	1.5	1.8	800	646	1.0	1.8
NO (ppb)	798	114	241	97	0.3	0.9	256	78	0.3	0.7
NO ₂ (ppb)	70	31	33	28	0.5	0.9	45	25	0.6	0.8
SO ₂ (ppb)	40	11	10.6	4.4	0.3	0.4	13.4	3.9	0.3	0.4

Generation of metabolic CO₂ by occupants of the buildings indicated a internal/external ratio greater than one. However, it is important to note that for the pollutants, the levels indoors are always less than those outdoors.

4.2 Air change rates

Average air change rates obtained for the two buildings over the monitoring period were as follows:

Naturally ventilated building:	Occupied	- 1.6 ach ⁻¹
	Unoccupied	- 0.8 ach ⁻¹
Mechanically ventilated building:	Occupied	- 1.2 ach ⁻¹
	Unoccupied	- 0.4 ach ⁻¹

In the mechanically ventilated building, the air change rate varied little whilst the system was operating. However, as expected, the air change rate in the naturally ventilated building was more variable and was dependent upon the circumstances of operation and weather conditions. Over the period of the tests, wind speeds were recorded at the nearby Meteorological site as being higher than normal, ranging from 5 ms⁻¹ to 10 ms⁻¹ with gusts of about three times the mean wind speed (Figure 7).

5. DISCUSSION

The short period of monitoring in this pilot study allowed only a limited scope for investigation of the performance of the buildings with regard to the indoor and outdoor air quality. However, a number of useful observations were made with regard to comparison with guidelines on exposure levels of the contaminants studied.

5.1 Sources of Pollution

In this study traffic appeared to be the major source of CO and NO_x. The concentrations of these pollutants varied with the traffic density on the main road. However, SO₂ concentrations were not so clearly associated with the traffic as can be seen from the low levels which occurred on the Saturday evening (Figure 6) when the high levels of other pollutants suggest a considerable flow of traffic.

The most prominent feature of the data is the exceptionally high levels of CO₂, SO₂, NO and NO₂ in the mechanically ventilated building on the Saturday morning from 06:00 to 12:30 - coincident with the period between the start-up and shut-down of the building's ventilation system. Levels were well in excess of the external concentrations and not seen inside the naturally ventilated building. The obvious source for these gases in such high concentrations was either cross-contamination from the ventilation exhaust or high level discharges from boiler plant being drawn into the ventilation inlet. There are a number of boiler flues close to the building which may have been responsible for this, if not those on the building itself. A similar event on the Monday morning suggests that this was not an isolated incident.

5.2 Health risk from external air pollutants

Table 2 shows the measured daily mean concentrations for the working week taken from Table 1 and compares them with some of the most appropriate guidelines for air quality. It can be seen that for all of the gases measured there was little risk in either of the buildings when these standards were applied. However, if the results from the weekend (Figures 2-6) are compared then the NO concentration, for a limited period, does exceed the HMIP guideline and the NO₂

Table 2. Measured mean concentrations inside the buildings for the working week compared with existing air quality guidelines.

Measured gas	Measured mean concentrations		Air quality standards/guidelines				
	Naturally ventilated	Mechanically ventilated	DoE*	HSE**	WHO [#]	HMIP ^{##}	EC [~]
CO (ppm)	1.7	0.9	-	50	10 ¹	50	-
CO ₂ (ppm)	650	646	-	5000	-	-	-
NO (ppb)	97	78	-	25000	-	830	-
NO ₂ (ppb)	28	25	<50	3000	80 ¹	100	105 ¹
SO ₂ (ppb)	4.4	3.9	<60	2000	122 ²	170	94 ²

* Department of Environment - very good air quality

** Health and Safety Executive 8-hour exposure TWA (time weighted average)

World Health Organisation

UK's Her Majesty's Inspectorate of Pollution short-term exposure limit

~ European Community guidelines

1 8-hour mean

2 1-hour mean

concentration exceeds almost all the guidelines. The possible cause for this is discussed in section 5.1. These results suggest that for this particular building there may be a cause for concern if these occurred when the building was occupied.

5.3 Attenuation of Pollutants

The results show that the buildings significantly attenuated the concentration of the external pollutants. The ratio of the internal/external peak values in the occupied periods (Table 1), indicate that the levels of the external pollutants monitored inside the buildings were generally less than 50% of the peak external concentrations. This was consistent over the period of monitoring and shows that this damping of the fluctuations of external pollutant levels would be valuable in reducing short-term exposure levels. There was less attenuation of the mean concentration but both buildings show better air quality indoors than outdoors in terms of the externally generated pollutants. The attenuation of the CO in the mechanically ventilated building was greater than in the naturally ventilated building. However, for the other pollutants the difference was less marked.

5.4 Correction for building air change rates

In order to allow for the fact that they experienced different air change rates and sources of external air, a form of correction procedure is required. Investigations into suitable techniques for performing this 'correction' are currently being carried out. When a building is to be located in an urban area it would be desirable if the ventilation system designer could predict the internal air quality from records of external pollution events for that site. The development

of the proposed correction procedure should enable such a prediction to be carried out.

6. CONCLUSIONS

From the measurements made in this pilot study a number of conclusions can be drawn about the attenuating capability of the buildings and the comparison of the internal measured concentrations with existing air quality guidelines.

1. In both the naturally and mechanically ventilated buildings the indoor air quality followed the trend of that of the external air to which they were exposed. However, the concentrations of the external pollutants were attenuated by the building and the transient peak concentrations measured externally were approximately halved in value.
2. The possibility of drawing combustion products into buildings at high level has been shown to exist in the mechanically ventilated building. The building was chosen only for its location with respect to traffic density. However, the findings highlight the real danger of cross-contamination between ventilation exhausts and inlets at roof level and contamination from other sources.
3. The indoor air quality in both buildings over the main occupied period did not exceed any of the main health standards. However, most of the NO₂ guideline values were exceeded over the weekend when combustion products were possibly drawn into the air conditioned building.
4. In terms of indoor air quality it is not clear which guidelines should be followed when designing buildings. If the more stringent guidelines are applied then the designer needs to think carefully when designing for non-domestic buildings and their ventilation systems in urban areas such that these guideline values are not exceeded.
5. It has been shown in this short pilot study that there is no clear distinction between the two ventilation strategies in providing adequate indoor air quality for the occupants of the buildings with respect to externally generated air pollutants other than when combustion products were entrained into the air-conditioned building.

ACKNOWLEDGEMENTS

This research was supported by the Department of the Environment (DoE) and carried out under the DoE Construction Sponsorship Directorate's Energy-Related Environmental Issues (EnREI) programme. The research team would like to express their gratitude to the owners and occupants of the buildings for their help and in assisting with the project. The assistance of R Woolliscroft (Sheffield Hallam University) when working at BRE, R Watkins and W Pane of John Palmer Associates and I Ridley of University of Westminster is also gratefully acknowledged.

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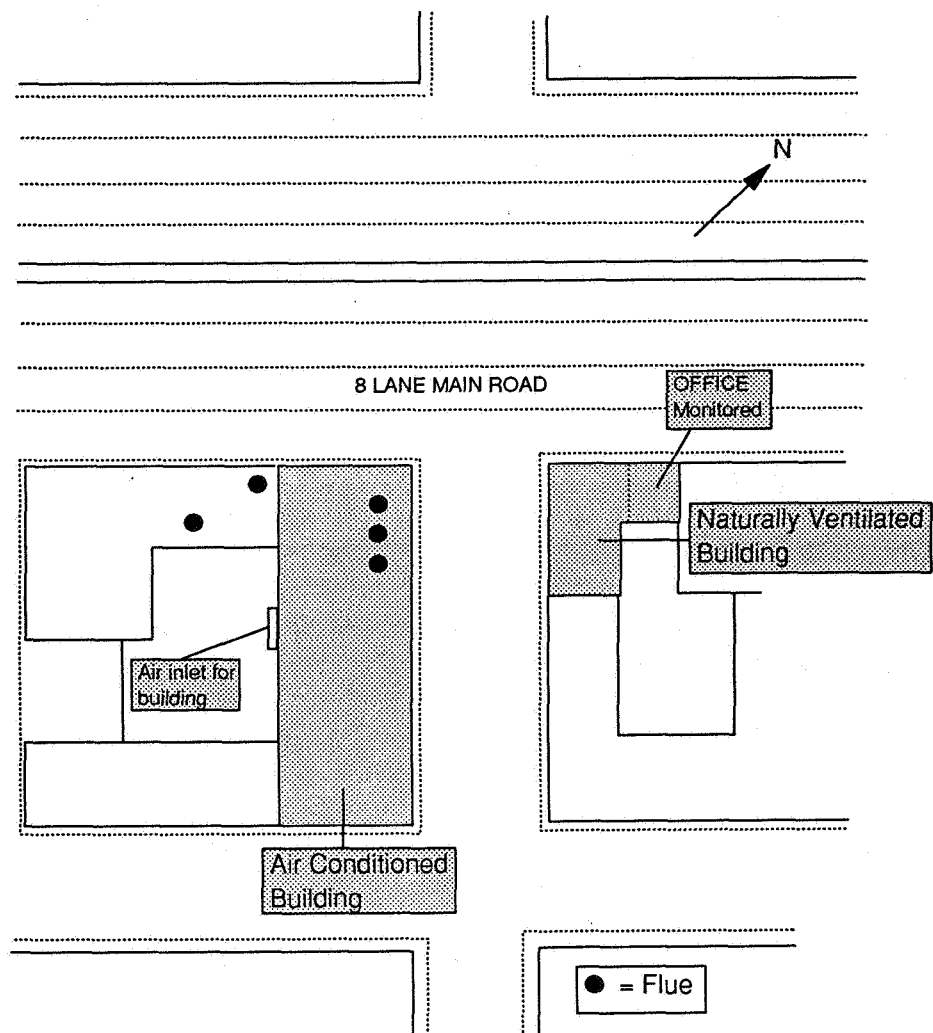


Figure 1. Site plan of monitored buildings

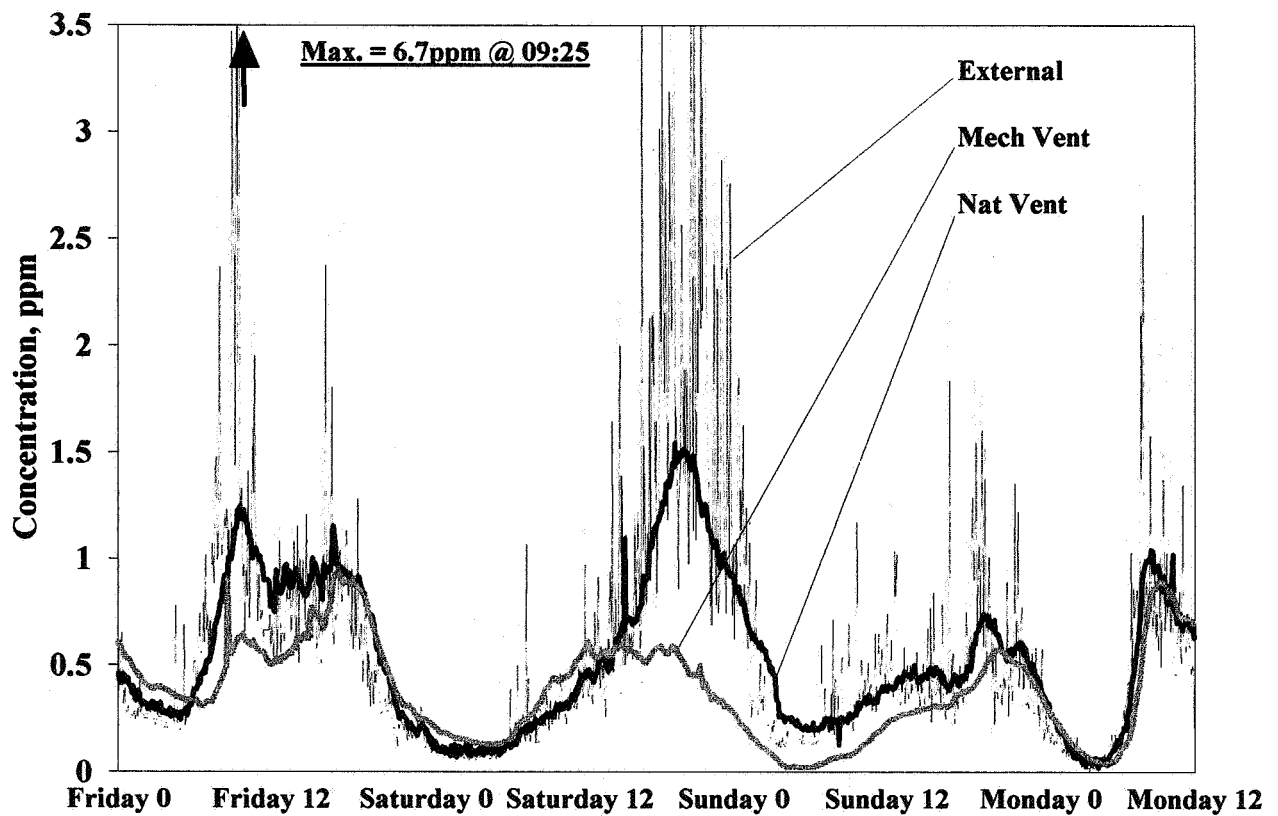


Figure 2. Carbon Monoxide

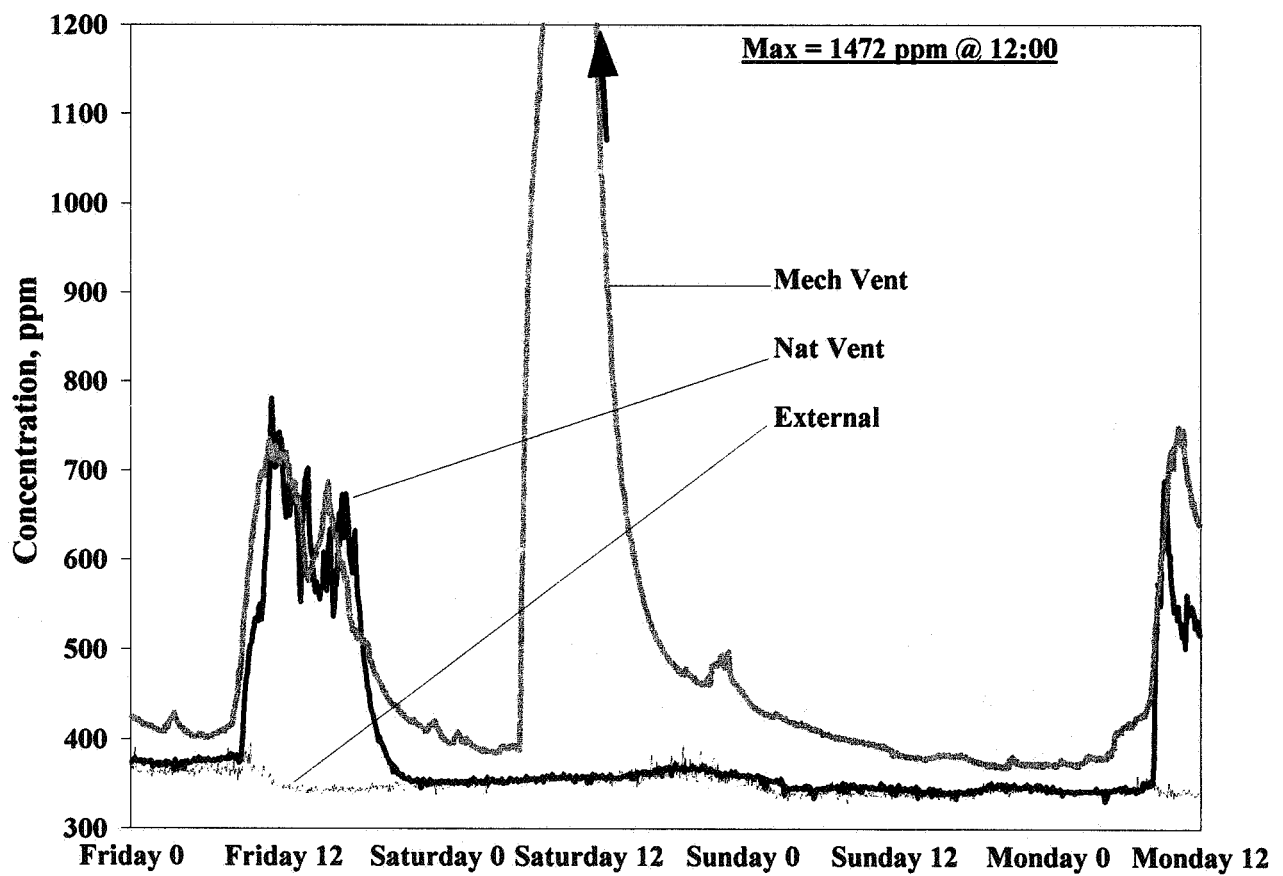


Figure 3. Carbon Dioxide

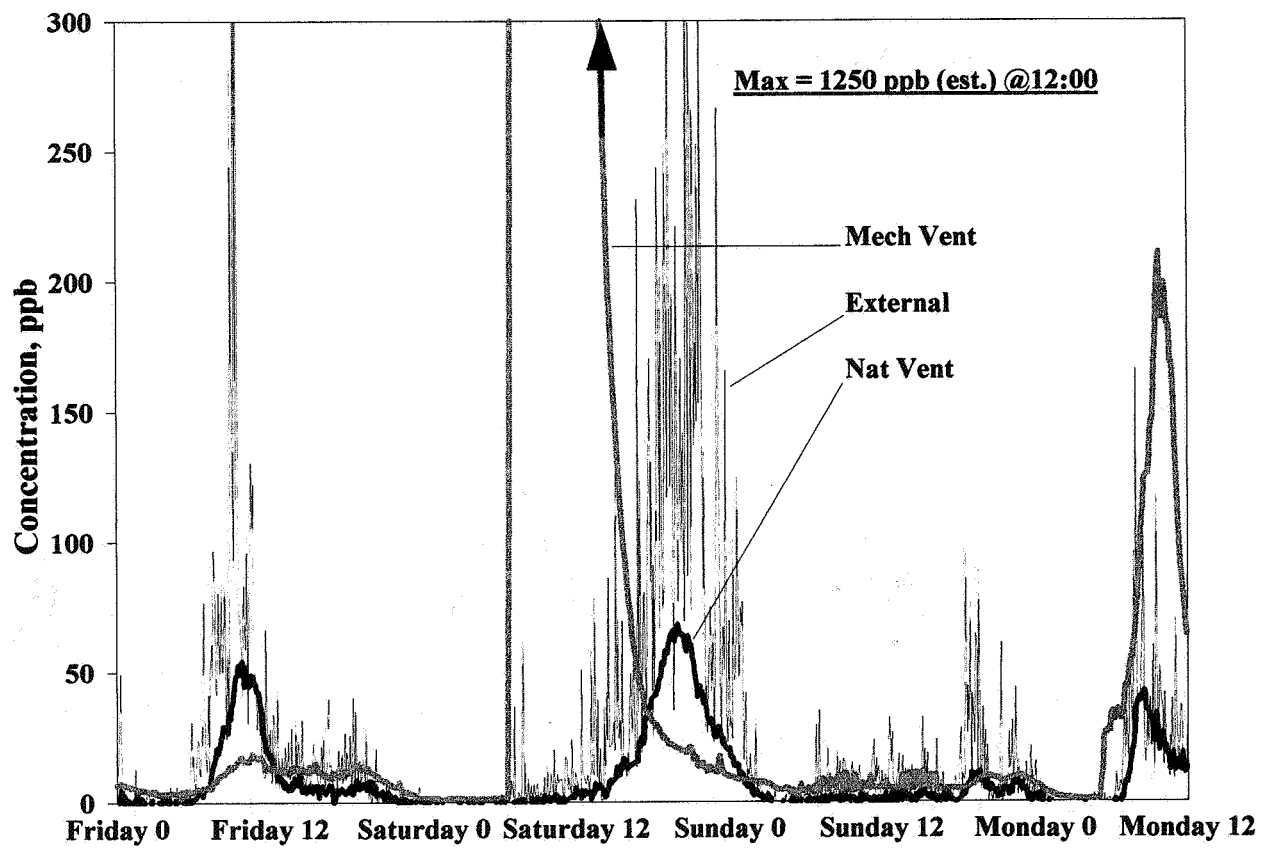


Figure 4. Nitric Oxide

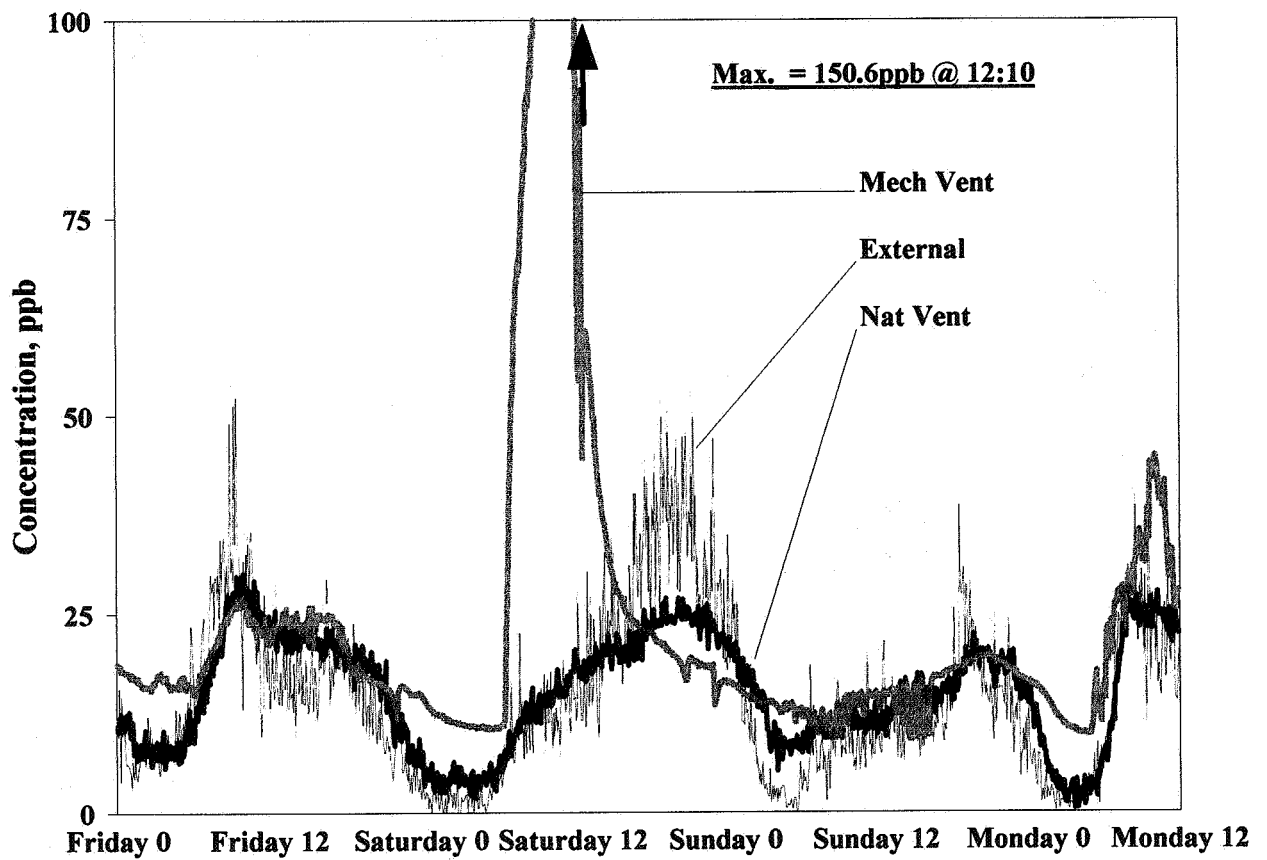


Figure 5. Nitrogen Dioxide

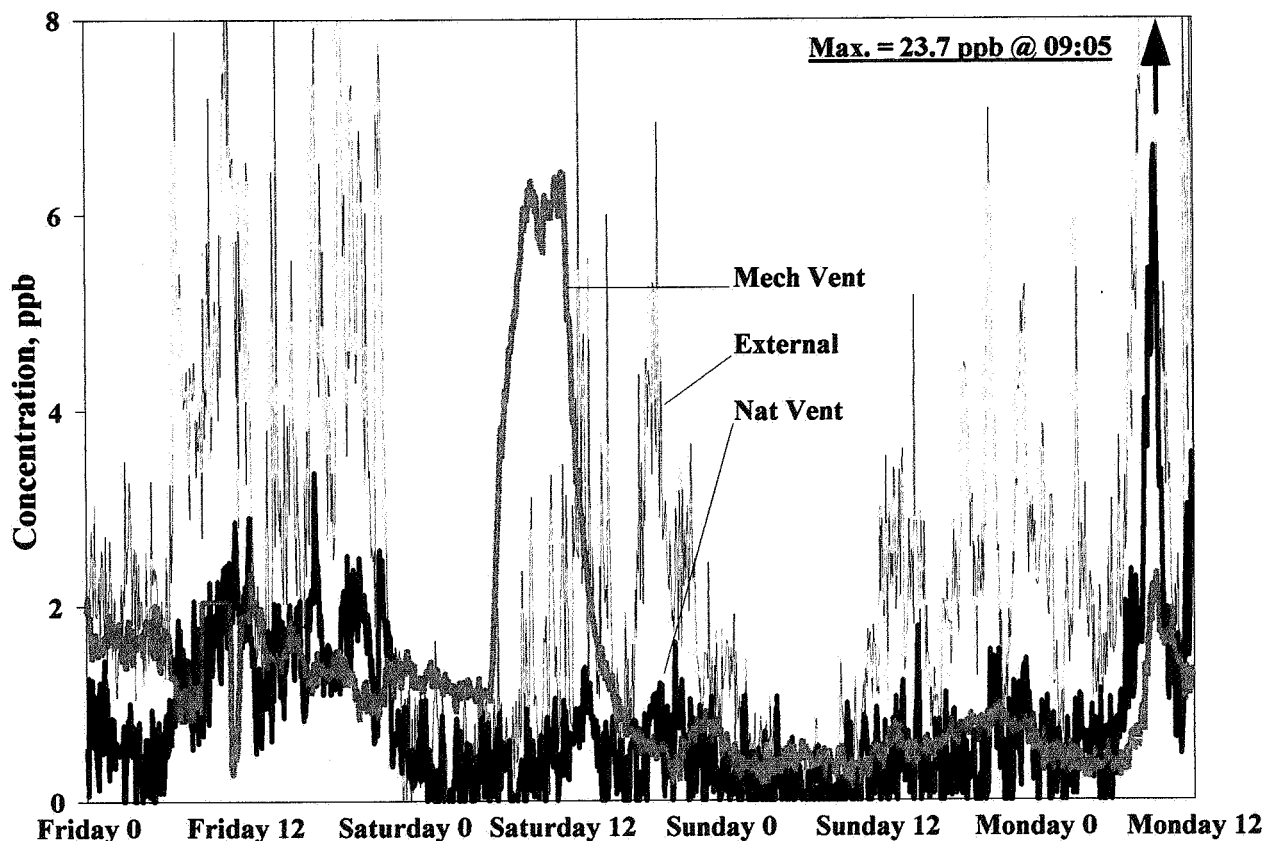


Figure 6.

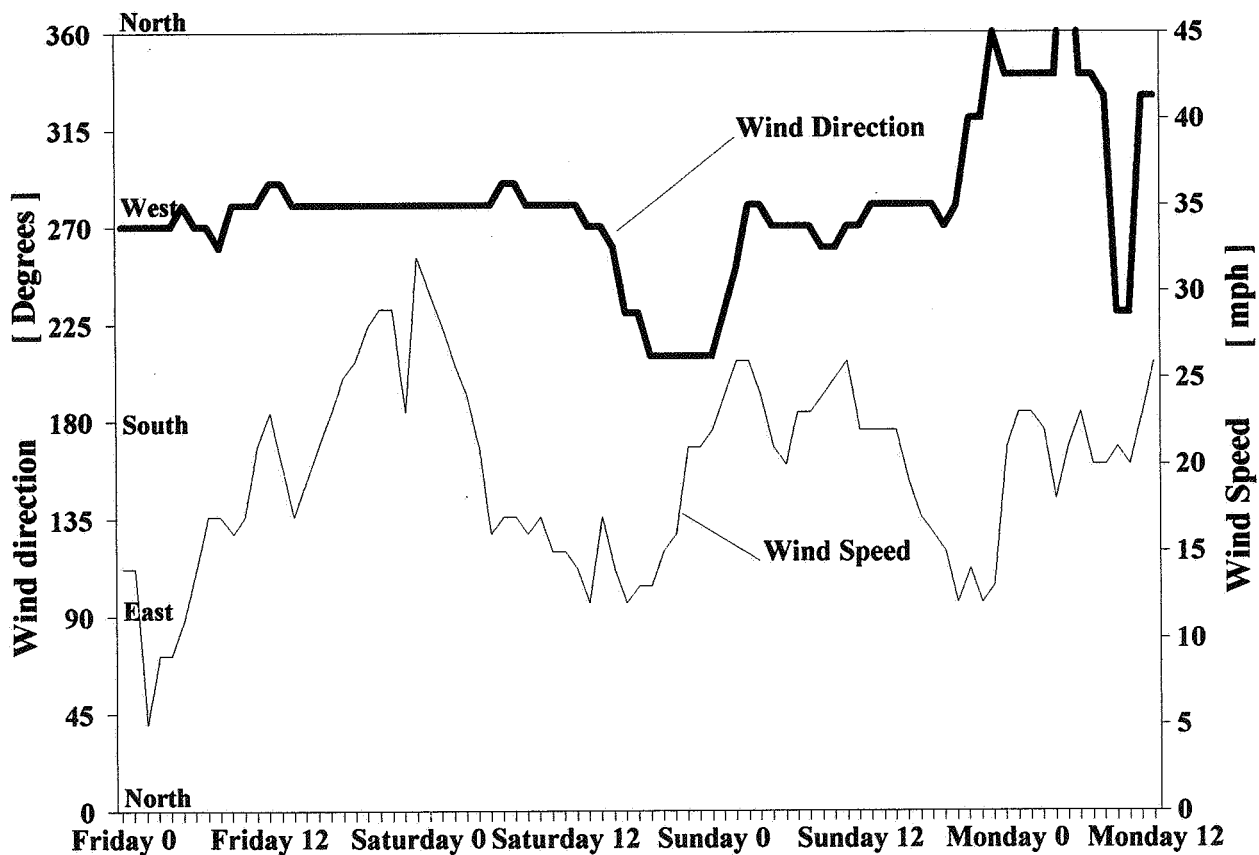


Figure 7.

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

(Title) Comparison of Indoor Levels of Radon Between Workplaces
and Homes Located nearby in Different Parts of Finland

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SYNOPSIS

The aim of this study was to compare the radon levels at workplaces and in homes located nearby. Homes (number of 57) and partly or fully underground workrooms (number of 55) have been studied at the four workplaces in southern Finland and one workplace in northern Finland. Radon concentrations both at workplaces and in homes seemed to be at the same level in the same district. The mean radon concentration in workrooms was 406 Bq.m^{-3} , and in homes concentration was 398 Bq.m^{-3} . At the workplaces having mere exhaust ventilation the mean radon concentration was higher (arithmetic mean of 677 Bq.m^{-3} , $n=14$), than the places having mechanical exhaust and supply ventilation (arithmetic mean of 207 Bq.m^{-3} , $n=33$). In an average the naturally ventilated workplaces ($n=8$) had the lowest level of indoor radon (arithmetic mean of 133 Bq.m^{-3}). The highest radon level, both in the workrooms (2937 Bq.m^{-3}) and in the homes (3080 Bq.m^{-3}), was found in the northern Finland. The high values of indoor radon might be partly explained by the hill-construction of buildings without sealed constructions against soil, and partly by depressurisation caused by mechanical exhaust ventilation.

INTRODUCTION

Radon is a radioactive noble gas, which is the decay product of radium. Radon enters a building mainly from soil below the building. Radon could also be exhaled from tap water or building materials. In addition, indoor radon concentration depends on meteorological factors, subgrade structures, air exchange rates and pressure conditions. Effective mechanical ventilation reduces indoor radon concentration, if the constructions of the buildings have been properly sealed and the underpressure is not increased /1, 2/. Houses having crawlspace usually have lower radon levels than slab-on-grade constructed houses. Variation in both outdoor temperature and wind induce the subterranean air-flows in the esker, which was observed to result in winter/summer radon concentration ratios of 3-30, in esker top areas, whereas in slope areas with amplified summer concentrations, the ratio is typically 0.1-0.5. Typical values in flat areas are 1.5-2. /2/. In the Chinese study at underground workplaces the radon concentrations increased from 13.6 Bq.m^{-3} to 119.0 Bq.m^{-3} , with the underground depth of 5 meters to 35 meters, respectively /3/. In Finland the radon levels in homes have been studied by the Finnish Centre for Radiation and

Nuclear Safety (STUK). The overall radon level in flats and in single-family houses was 82 Bq.m⁻³ and 145 Bq.m⁻³, respectively /4/. Radon levels at workplaces have been studied by STUK and the radon levels at partly or fully underground workplaces have been studied by our group. The concentrations have been found to vary a lot in different parts of Finland. The mean concentrations during working hours have been 90 Bq.m⁻³ in central Finland and 300 Bq.m⁻³ in southern Finland /5/.

MATERIAL AND METHODS

The workplaces in this study were located in southern Finland in four different districts (places A-D) and one place in northern Finland (place E). The total number of measured workplaces was 55 and number of the homes was 57 (table 1.). Most of the workrooms had mechanical supply and exhaust (n=33) and the rest had either mere mechanical exhaust (n=14) or natural ventilation (n=8) (table 1.). All the homes had natural ventilation and a ground level foundation.

Table 1. The number of workrooms with different types of ventilation, different types of foundations and number of the homes in each places.

	Workplaces							Homes
	Types of ventilation			Types of foundation			N	N
	MSE	ME	NV	U	G	H		
Place A	7	-	1	7	1	-	8	4
Place B	9	1		3	6	1	10	11
Place C	6	5	7	3	15	-	18	1
Place D	6	1	-	5	2	-	7	1
Place E	5	7	-	-	1	11	12	40
Number of places	33	14	8	18	25	12	55	57
MSE	mechanical supply and exhaust			U	underground workroom			
ME	mechanical exhaust			G	ground level workroom			
NV	natural ventilation			H	hill-constructed workroom			
				N	number of workrooms or homes			

Radon levels were analysed continuously near the workers' breathing zone by using the Lucas cell method /6/ with a Pylon AB-5 assembly, which includes a detector, a photomultiplier and a

system of data collection based on a microprocessor. The output data of the Pylon detector were processed with SP-55 software run on a PC. The flow rate of air was 0.4 l/min. The interval of continuous measurements was 30 minutes (averaged to one hour). Concentrations were measured during periods ranging from few hours to several days. The integrated long-term radon levels at workplaces and homes were determined by alpha track etch films and analysed by STUK /7/. At workplaces alpha films revealed the average radon level during one month. This integrated radon concentration also included the radon levels at nights and weekends, when the ventilation was not usually operating at full capacity. In homes alpha films revealed the average radon level during two months. The pressure differences across the wall, either separating or external, were monitored by an electronic manometer together with a datataker. The pressure differences were measured continuously averaging every 30 minute to one hour. During daytime working hours, air exchange rates were measured by the tracer gas technique and by the dilution method using nitrous oxide as the tracer gas and an infrared spectrophotometer, Miran 1A, as the analyser.

RESULTS

The continuously measured and integrated radon levels varied from 27 to 2937 Bq.m⁻³ (arithmetic mean of 352 Bq.m⁻³, n=40) and from 20 to 405 Bq.m⁻³ (arithmetic mean of 205 Bq.m⁻³, n=20) at the workplaces, respectively. The integrated radon levels varied from 20 to 3080 Bq.m⁻³ (arithmetic mean of 398 Bq.m⁻³, n=57) in homes (figure 1. and table 2.). The arithmetic mean of the air exchange rates in the underground workrooms or workrooms on the ground level were 3.1 h⁻¹ (n=35) and the arithmetic mean of pressure difference was -5.6 Pa (n=11) (table 2). The highest average value, 658 Bq.m⁻³, when comparing the different types of ventilation, was found from the places having mere mechanical exhaust (table 3.). The mechanical exhaust causing underpressure (tables 2. and 3, place E) elevated the radon level. The underground location of the workroom did not seem to increase the radon concentration (fig. 2. and table 4.).

The radon concentrations in the place A were quite low (table 2.), although 7 of those places were underground (table 1.). This may be due to effective ventilation in the underground workrooms (7 h⁻¹) (table 2.). The highest radon concentration, measured both with continuous

and integrated methods, existed in the workroom located ground level. This workroom had only natural ventilation (tables 3. and 4.). Also the integrated radon level in the home was at the same level than the level in ground level workroom, but higher than the levels in the underground workrooms (figure 1. and table 4.).

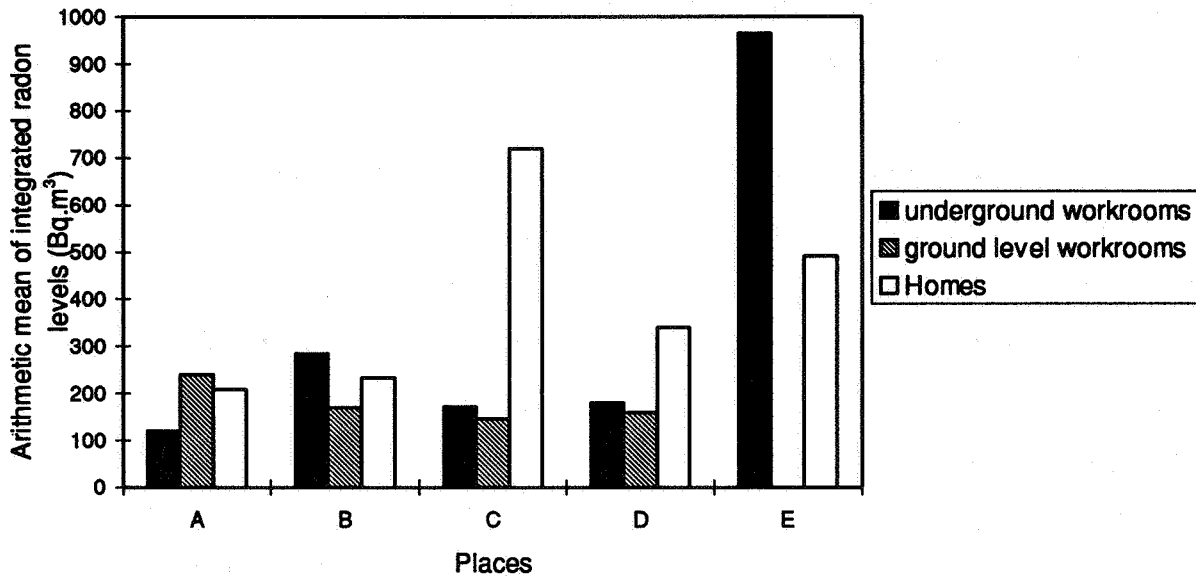


Figure1. The arithmetic means of integrated radon concentrations (Bq.m^{-3}) on the ground level and underground workrooms and homes in each places.

Table 2. The arithmetic mean, the range and the number (#) of the continuous and integrated radon concentrations (C_{Rn} , Bq.m^{-3}), air exchange rates (h^{-1}) and pressure differences (Pa) in the workrooms and integrated radon concentrations in the homes.

Places	Workplaces				Homes
	Continuous C_{Rn} (Bq.m^{-3})	Integrated C_{Rn} (Bq.m^{-3})	Air exchange rates (h^{-1})	Pressure differences (Pa)	Integrated C_{Rn} (Bq.m^{-3})
Place A	68 (27-174, 8)	120 (70-240, 5)	7.0 (0.4-12.0, 7)	1.3 (3.7-1.2, 4)	208 (20-400, 4)
Place B	288 (48-653, 6)	285 (160-450, 4)	2.6 (2.0-4.0, 5)	-	233 (60-460, 11)
Place C	134 (36-349, 9)	172 (20-370, 9)	1.3 (0.1-3.2, 6)	5.1 (1)	720 (1)
Place D	306 (41-1163, 5)	243 (160-200, 2)	2.8 (1.3-4.7, 5)	0.6 (0.3-0.8, 2)	340 (1)
Place E	965 (56-2937, 12)	-	1.6 (0.4-2.7, 12)	10.4 (19.3-2.8, 5)	491 (20-3080, 40) ^e
Average	352 (27-2937, 40)	205 (20-450, 20)	3.1 (0.1-12.0, 35)	5.6 (19.3-0.8, 11)	398 (20-3080, 57)

^e the measurements have been done by STUK

In the place B the integrated radon levels were at the same level both at workrooms and at homes (table 2.). The highest radon concentration was at the workroom having mere mechanical exhaust (table 3.). In this district the continuously measured radon levels were higher on the ground level rooms than underground, and integrated levels were lower (table 4.). In the place C the integrated radon concentration in the home was 720 Bq.m^{-3} , which is the highest radon concentration in this district (tables 2. and 3.). With the same type of ventilation, natural ventilation, radon levels in workrooms were lower than the level in the home (table 3.). The ventilation in these workrooms was not so effective, which might cause elevated indoor radon levels. In this place integrated and continuously measured radon concentrations were at the same level on the ground level rooms (table 4.). In the place D the highest continuously measured radon concentration was 1163 Bq.m^{-3} . This workroom located underground and had mechanical exhaust and supply ventilation (tables 2., 3. and 4.). The circulation of the exhaust and supply air were used which may have increased the radon level. The home that located little further from the ground level workrooms had higher radon concentration than the workrooms (tables 3. and 4.). The highest radon concentrations were observed in place E, both in the workrooms (arithmetic mean of 965 Bq.m^{-3}) and in the homes (arithmetic mean of 491 Bq.m^{-3}). Also these workrooms were depressurized (down to -20 Pa) and the air exchange rates were quite low ($0.4\text{-}2.8 \text{ h}^{-1}$, $n=12$) (table 2.). The workrooms, except one, and homes were located on the slope of the esker. All these buildings, where former workrooms locate, had one wall against the ground (table 4.) and most of them had mere mechanical exhaust (table 3.).

Table 3. Continuous and integrated radon levels (C_{Ra} , Bq.m^{-3}) and numbers (N) of workplaces and homes with the different types of ventilation.

	Arithmetic mean of radon concentration (Bq.m^{-3}) (N)			
	MSE	ME	NV	Homes, NV
Place A, continuous / integrated	53 (7) / 90 (4)	- / -	- / 174 (1)	208 (4)
Place B, continuous / integrated	260 (5) / 285 (4)	426 (1) /	- / -	233 (11)
Place C, continuous / integrated	184 (5) / 175 (2)	75 (2) / 230 (3)	71 (2) / 136 (5)	720 (1)
Place D, continuous / integrated	306 (6) / 200 (1)	- / 160 (1)	- / -	340 (1)
Place E, continuous / integrated	252 (5) / -	1474 (7) / -	- / -	491 (40)
Total average	211 (28) / 188 (11)	658 (10) / 195 (4)	71 (2) / 155 (6)	361 (57)
MSE	mechanical supply and exhaust	ME	mechanical exhaust	NV natural ventilation

Table 4. The continuous and integrated radon levels (Bq.m^{-3}) and numbers (N) of different types of foundation in workrooms in each places.

	Arithmetic mean of radon concentration (Bq.m^{-3}) (N)		
	underground	hillside	ground level
Place A, continuous / integrated	53 (7) / 90 (4)	- / -	174 (1) / 240 (1)
Place B, continuous / integrated	59 (1) / 400 (1)	48 (1) / -	405 (4) / 170 (2)
Place C, continuous / integrated	68 (1) / 275 (2)	- / -	142 (7) / 146 (8)
Place D, continuous / integrated	372 (4) / 180 (1)	- / -	41 (1) / 160 (1)
Place E, continuous / integrated	- / -	1041 (11) / -	126 (1) / -

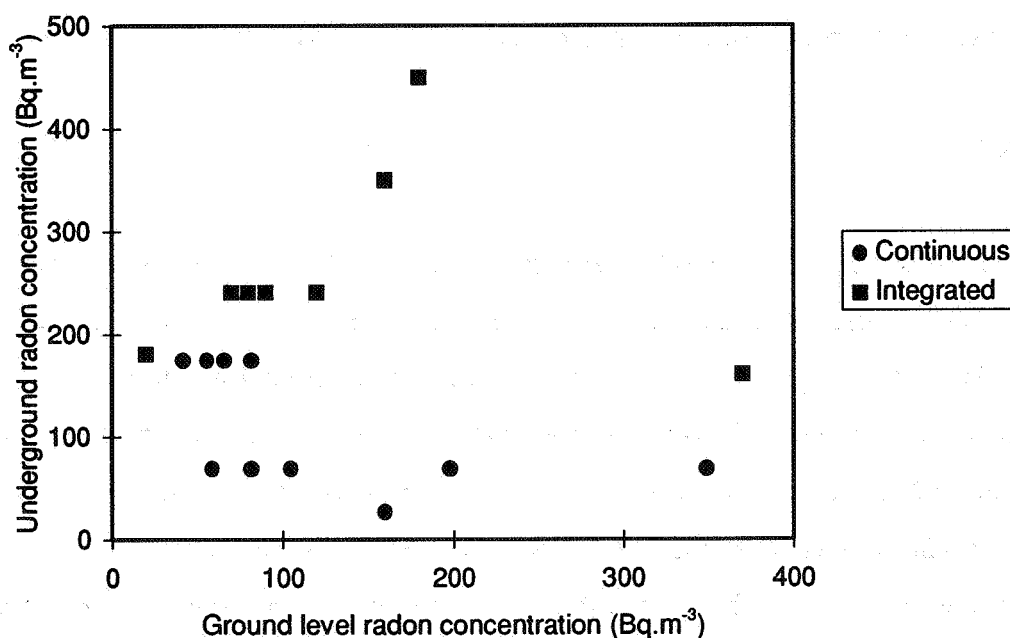


Figure 2. The radon levels (Bq.m^{-3}) measured by continuous and integrated methods in underground and ground level workrooms.

In the place A continuous and integrated measurements have been done at the same time in 5 of the workrooms. The radon concentrations seemed to be at the same level with both used methods (figure 3.).

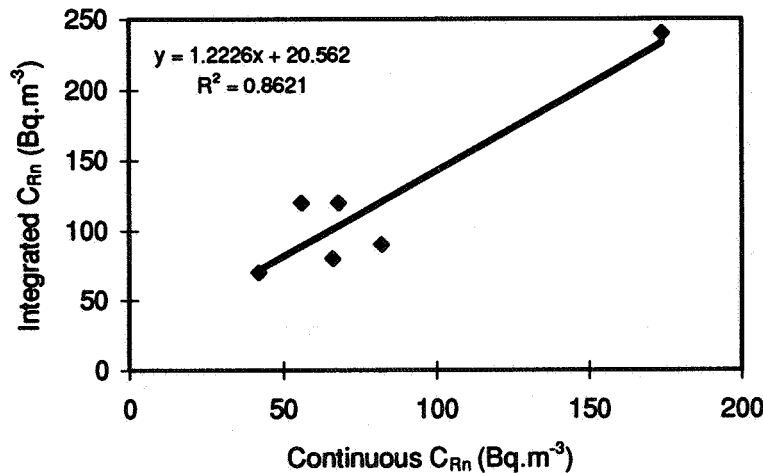


Figure 3. Comparison of the continuous and integrated methods of radon measurements.

CONCLUSIONS

The radon levels were highest on the hillside area in northern Finland where the hill-constructed workrooms had mere mechanical exhaust. The possible explanation to the high radon concentration could be that the parts of the walls of these buildings were constructed against the soil, which made radon entry possible. Also the high underpressure due to mechanical exhaust increased radon entry from soil. Also the homes on the hillside area had highest radon levels, when compared to other homes. Generally, the radon levels at workplaces and in homes were observed to be at the same level in the same district. The underground location of workrooms did not rise the indoor radon levels at all the places. Some places had even lower radon level in underground workrooms than the ground level workrooms. This was possibly caused by effective dilution by mechanical supply and exhaust ventilation at underground workrooms.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

(Title)

**Particulate Pollution Interactions with Indoor Surfaces:
Measurement and Modelling for Risk Assessment and
Contaminant Control.**

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Particulate Pollution Interactions with Indoor Surfaces: Measurement and Modelling for Risk Assessment and Contaminant Control.

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Synopsis.

In the urban environment, there is strong evidence that fine particulates associated with vehicular emissions are linked with respiratory problems and an increase in mortality. The population sector most at risk is the elderly who spend much of their time indoors; consequently, the infiltration of these particles and their subsequent behaviour indoors is of primary concern.

The deposition of aerosol particles in the respiratory system and on indoor surfaces is a process governed by particle size; in addition to providing risk assessment data, an understanding of the interaction of particles with indoor surfaces can be applied to the design of systems for enhancing indoor aerosol deposition and thus inhibiting inhalation exposure. This paper describes experiments using tracer-labelled aerosols, in a range of monodisperse size distributions representative of real particulate pollutants, to study aerosol deposition on surfaces with representative roughness. Some preliminary data, exhibiting electrostatically-enhanced aerosol deposition are also presented. In order to make these data widely accessible, the experimental results are used to aid in the development of a CFD code by providing validation data. Simulations are described for a room-sized enclosure with representative indoor surfaces, to illustrate the influence of internal building surface characteristics on indoor aerosol concentration modification.

1. Introduction.

Air pollution studies in cities of the United States and in London have demonstrated a strong association between urban aerosol concentrations (largely attributed to vehicular emissions) and excess numbers of deaths from respiratory diseases. Since excess deaths are most clearly seen in the elderly sector of the population, the U.K. Department of the Environment's Expert Panel on Air Quality Standards conclude that particulate pollution episodes are most likely to exert their effects on mortality by determining the time of death of those rendered susceptible to pre-existing disease; however the possibility also exists that prolonged exposure to air pollution may contribute to disease development (1).

Since the elderly sector of the population spends much of its time indoors, and often in naturally-ventilated dwellings, indoor inhalation exposure to particulates of outdoor origin is of primary concern. Ingressed aerosol is either removed from indoor air through ventilation or is deposited on internal building surfaces; aerosol deposition rate measurements (2) have shown that, particularly at low air exchange rates, particulate deposition is an important modifier of inhalation exposure. Aerosol deposition, both inside the respiratory system and on indoor surfaces, is a particle size-dependent phenomenon.

The motion of the largest particles (those greater than $1\ \mu\text{m}$), which lodge in the upper airways, is gravitationally-dominated; the inhalation of these particles could therefore be inhibited by the promotion of turbulent deposition and impaction on indoor surfaces.

Particles smaller than $1\mu\text{m}$ penetrate more deeply into the lung (3); particles at the upper end of this range ($0.1 - 1\mu\text{m}$), are neither strongly influenced by diffusive effects or gravity. It is expected, therefore, that external forces would play a role in enhancing the deposition of these particles on indoor surfaces, thereby reducing their inhalation potential; in the indoor environment, such forces commonly arise, for example, from electronic visual display units which generate electrostatic gradients. Particles smaller than $0.1\mu\text{m}$ are highly susceptible to alveolar deposition, due to Brownian diffusion; since these particles behave essentially like a gas, their inhalation potential is largely governed by air-exchange.

To provide risk assessment data for airborne particulate exposure and to examine whether the forced enhancement of aerosol deposition on indoor surfaces might be an effective means of inhalation dose modification, a detailed understanding of the mechanisms controlling aerosol deposition is required. Few direct measurements of deposited aerosol particle masses on internal building surfaces exist since such studies require sensitive multi-elemental analytical techniques which are not widely available; the techniques employed at Imperial College for measuring aerosol deposition rates to internal building surfaces are described in the next section. In order to make these data widely accessible and limit the necessity for further costly and labour-intensive measurements, the experimental results are used to aid in the development of a CFD code by providing validation data. The ultimate objective is to develop a code which predicts accurate particle/surface interactions for a range of ventilation conditions, surface features and particulate characteristics.

Following the description, in the next section of this paper, of the techniques used for measuring aerosol deposition to surfaces, the main features of the computational model are summarised. The current status of model development is then illustrated, by comparing predicted aerosol-surface interactions with the measured values.

2. Experimental.

2.1 Aerosol generation and labelling.

The techniques used for generating, dispersing and detecting tracer labelled particles have been previously described (4). Porous silica particles, available in a variety of supra-micrometre monodisperse size distributions (with mass median aerodynamic diameters of $2.5\mu\text{m}$, $4.5\mu\text{m}$ and $5.4\mu\text{m}$, respectively), are agitated in a tracer salt solution so that tracer ions become bound to the particles' surfaces. The labelled particles are dispersed using a rotating brush aerosol generator. Sub-micrometre tracer particles, with a mass median aerodynamic diameter of $0.7\mu\text{m}$, are generated by atomisation and subsequent evaporation of a tracer salt.

Salts containing the rare earth elements dysprosium and indium are used as tracers in this work. Both exist naturally in a stable state but become unstable (i.e. radioactive) when bombarded with neutrons. Subsequent to a period of aerosol deposition in a test room/chamber, neutron irradiation of tracer aerosol-bearing materials (such as air filter papers or samples of domestic furnishing materials) in a nuclear reactor, followed by gamma-spectrometry, allows a quantitative determination of the aerosol mass present. Since dysprosium and indium occur naturally in low concentrations, the possibility of analytical interferences from particle-bearing media are minimised.

2.2 Aerosol deposition measurements in a test chamber.

The sensitive aerosol detection techniques summarised above have been employed to study aerosol deposition in the Imperial College test chamber. The chamber is an aluminium cube of side 2m , fitted with externally-accessible air sampling ports. An internal fan, situated on the central axis and pointing towards the floor, provides a source of inhomogeneous turbulence, in the range 22-43%. After a period of aerosol injection and deposition, during

which air sampling is carried out, filter papers attached in regular arrays to the internal surfaces of the test chamber are analysed. Since the initial aerosol concentration in the chamber can be calculated by air sample analysis, the proportion of aerosol particles deposited on each internal chamber surface can be inferred from the surface filters. The integrity of this approach has been proven by aerosol mass balance tests; the correction factors (which includes the aerosol particle mass deposited on the fan blades) involved in the mass balance calculations are discussed elsewhere (5).

Rough materials can be mounted over the smooth internal aluminium surfaces of the test chamber so that the effect of surface roughness on aerosol deposition can be studied. Direct sampling of the rough surface, subsequent to the aerosol deposition period, is avoided in order to minimise particle losses and aerosol deposition enhancement to the rough surface is inferred by comparison with the smooth surface case.

Using the four particle size distributions available, the relative proportions of aerosol particle mass deposited on the walls, ceiling and floor of the unlined test chamber were determined. The chamber walls were then lined successively with wallpaper, short-pile carpet, and astroturf and the enhancement in deposited aerosol particle mass was measured. The roughness of the surfaces was quantified by friction velocity measurements; using a hot-wire anemometer in a small-wind tunnel, the variation in air velocity with height above each surface was measured. The results, for two particle sizes, and smooth and rough chamber surfaces, are compared with model predictions in section 4.

In the indoor environment, persistent soiling of computer monitors and television screens provides evidence of the preferential deposition of aerosol particles on electrostatic surfaces. The deposition of charged aerosol particles on charged surfaces of opposite sign is easily understood; the mechanism by which uncharged particles deposit can be explained by considering that the charged surface induces an "image charge" of opposite sign on the particles, promoting attraction towards the surface and subsequent deposition (6). Using the test chamber, preliminary measurements were carried out whereby the deposition velocity (i.e. the aerosol particle mass deposited on a surface of unit area in unit time, divided by the particle mass concentration in the core of the enclosure) of $0.7\text{ }\mu\text{m}$ and $2.5\text{ }\mu\text{m}$ particles to vertical polythene sheeting was determined. In these measurements, the air turbulence in the test chamber was minimised, so that electrophoresis would be the dominant influence on aerosol transport. The results are shown in section 4.

3. Computational.

3.1 Introduction.

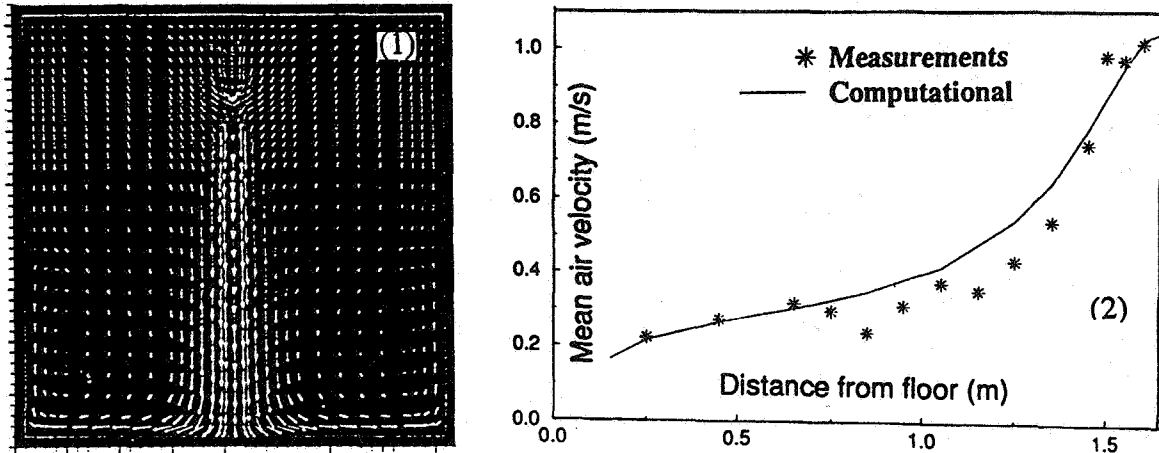
While a large number of researchers have studied the physical mechanism underlying particle deposition on surfaces, the research has largely been confined to the viscous sub-layer close to a surface, and deposition has been considered as a process isolated from the main turbulent flow field. The mechanism by which particles are transported to the boundary layer by the turbulent dispersion of particles in the main stream is an important and frequently overlooked phenomenon and to address this, an analytical method called the "Prediction of Evolving Probability" (PEP) model, in which the evolution in time of the particles' velocity probability density is derived from first principles, has been developed (7). This method considers all possible realisations of the turbulent flow field to obtain the evolution of the particles' velocity probability density function, with the result that the need for time-consuming stochastic trials is eliminated.

The present study adopts the PEP method to model the particle-turbulence interaction, along with the more conventional free-flight model to simulate the whole deposition process. The main features of the computational methods are summarised below.

3.2 Modelling the gas-phase.

The CINAR CFD code, which solves the governing equations in a Cartesian co-ordinate framework, has been applied to predict flows in large multi-burner industrial furnaces and other complex configurations. The standard two-equation 'k-ε' closure is employed to model the gas-phase turbulence and a finite difference/finite volume technique is used to solve the Navier-Stokes equations on a conventional staggered grid.

Using the mean air velocity value at the centre of the test chamber (measured using a Dantec 5410 hot film anemometer) to define the momentum source term due to the mixing fan, the CINAR code was used to simulate the gas-turbulence interaction, as shown in Figure 1. Figure 2 shows the computed mean velocity profile above the floor; good agreement with the measured velocity values is observed.



Figures 1 & 2. CFD simulation of the airflow generated by the use of the mixing fan in the test chamber (1) and comparison of experimental and computational air velocity values on the central axis of the chamber (2).

3.3 Modelling the particulate phase: the PEP model.

The particulate phase is simulated in a Lagrangian frame and the contributions of the particulate phase to the source terms of the gas-phase equations are evaluated according to the scheme proposed by Migdal and Agosta (8). The effects of the particle-turbulence interaction on the motion of the particulate phase are accommodated by the PEP model, assuming that the joint gas-particle velocity distribution is Gaussian. The evolution equation for the probability density function (pdf) of an ensemble particle velocity may be derived as follows (9) :

$$C_0[C_1 + C_2 v_p] P_{v_p} + P_t + C_0 C_1 P = 0 \quad [1]$$

where

$$C_0 = (\rho_g A_p C_D / 2 m_p) \cdot u_g - u_p ,$$

$$C_1 = \bar{u}_g - \frac{\rho_{gp} \sigma_g}{\sigma_p} \bar{u}_p ,$$

$$C_2 = \frac{\rho_{gp} \sigma_g}{\sigma_p} - 1 .$$

A_p and m_p are, respectively, the particle mass and surface area. C_D is the drag coefficient and ρ_g is the gas velocity. P denotes the pdf $P(v_p, t)$, v_p being the fluctuating particle velocity and t , the time; the subscripts v_p and t denote the partial derivatives with respect to these variables. u_g and u_p denote the mean values of gas and particle velocity in real space. σ_g and σ_p are the rms values of the gas-phase and particle-phase velocity fluctuations, respectively, and ρ_{gp} is the correlation coefficient for the gas-particle velocity fluctuations. The solution of equation [1] can be obtained using the method of characteristics. The discrete time step for the particle tracking is estimated as the minimum of either the particles' dynamic relaxation time or the time step based on the local cell dimensions.

3.4 The deposition model.

The deposition model used in the present simulation is based on the "stopping distance" concept and is similar to the commonly-used free-flight model (10). The steps in the deposition model are as follows: all cells are successively searched until a cell containing a particle is located. That particle is then traced through the flow field until it either exits the domain or arrives within a pre-calculated distance (which depends on the friction velocity) of the wall. If the particle leaves the domain, calculations for this particle are terminated and a new particle is initialised. Alternatively, if the particle reaches the edge of the sublayer, the velocity component normal to the wall is calculated. This velocity is used as a free flight velocity to determine whether the particle will penetrate through the sublayer and if the particle's incoming velocity is less than or equal to a critical velocity, the particle is assumed to be deposited. The critical velocity for each particle is calculated by incorporating a particle bounce model (11). In this, the energy loss mechanisms are theoretically modelled by considering the relationship between contact deformation mechanics and contact surface energy.

In the case of a rough wall surface, an additional deposition mechanism is provided and the deposition model must be modified. The mean velocity profile due to the roughness of the surface is shifted, using the correlation of Im and Ahluwalia (12), so that the zero velocity level is no longer at the wall, but some height above it. It is assumed that if a particle reaches a distance equivalent to the mean plus one standard deviation of the roughness elements with respect to the zero velocity level, it will be captured by the wall.

Computations were carried out for two monodisperse particle size distributions: 2.5 μm and 4.5 μm particles. In successive computations, the roughness heights of the chamber wall surfaces were adjusted to correspond to those of aluminium, wallpaper, carpet and astroturf.

4. Results.

Figure 3 shows the predicted proportion of 2.5 μm and 4.5 μm particles which deposit on the test chamber surfaces, compared with the experimental results. The model predicts that a significantly greater proportion of the particle mass is deposited on the floor than on the walls of the test chamber, in agreement with the measurements. These findings seem reasonable since the direction of airflow in the chamber, as dictated by the presence of the fan, is vertically downwards. In addition, the actual percentage of particles deposited on the floor, as predicted by the model, is greater in the case of the 4.5 μm particles than the 2.5 μm particles; this result is also in agreement with the experimental observations and confirms the anticipated gravitational influence in the larger particle size range.

Figure 4 shows the measured and modelled proportions of aerosol particle mass deposited on the rough test chamber walls. It can be seen that the measured values exceed those predicted by about a factor of two, and that the model is less sensitive to changes in surface

roughness. While the interaction of the particles with the rough surface is modelled within the sub-layer, the height of the roughness elements (11mm in the case of astroturf) is such that it is likely that they penetrate into the core of the flow field, causing inertial impaction of particles. The difficulty in incorporating this process in the model is thought to be largely responsible for the discrepancy between the predicted and measured values.

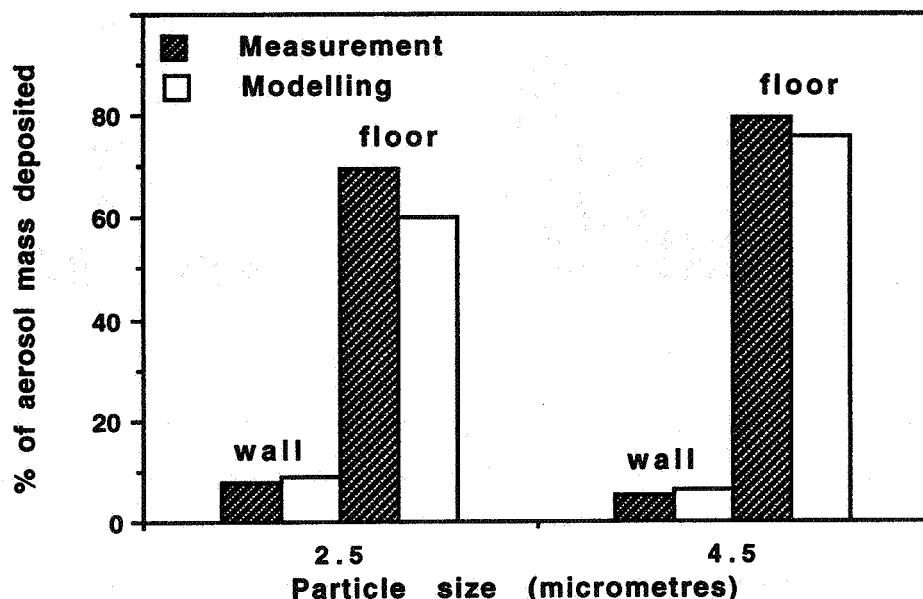


Figure 3. The proportion of aerosol particle mass deposited on the floor and one wall of the aluminium test chamber: measured and predicted values for 2.5 µm and 4.5 µm particles.

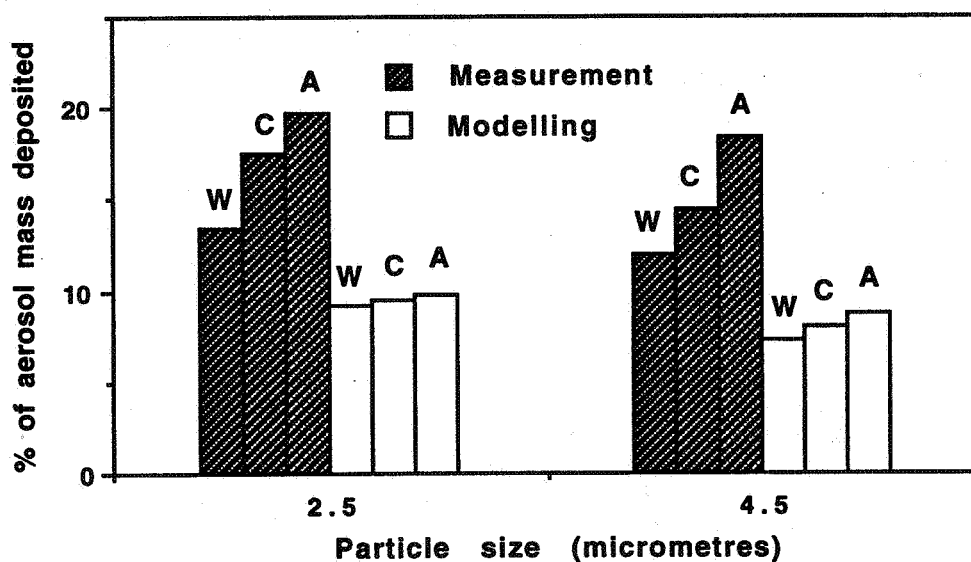


Figure 4. The proportion of aerosol particle mass deposited on a test chamber wall lined successively with wallpaper (W), carpet (C) and astroturf (A): measured and predicted values for 2.5µm and 4.5µm particles.

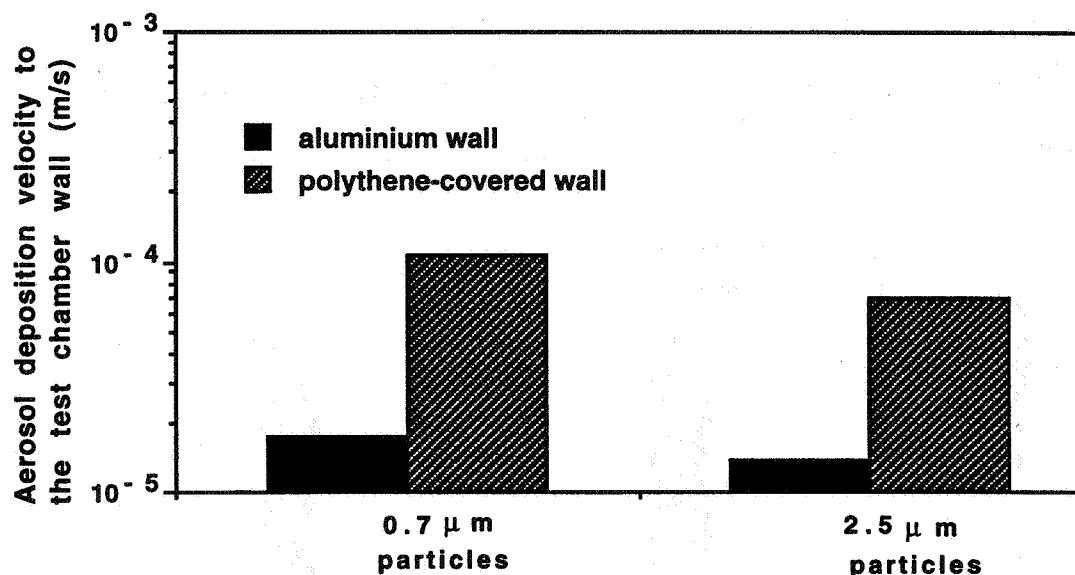


Figure 5. Measured aerosol deposition velocity to the test chamber wall, for 0.7 μm and 2.5 μm particles; comparison of a conducting (aluminium) and an electrostatic (polythene) wall surface.

The observed enhancement in aerosol deposition velocity to the wall of the test chamber, when that wall is lined with an electrostatic material, is shown in Figure 5. It can be seen that the effect is most pronounced for the 0.7 μm particles, since they are less strongly influenced by gravity than the 2.5 μm particles. The measured deposition velocity of the 2.5 μm particles to the aluminium floor of the chamber was $2.5 \times 10^{-4} \text{ ms}^{-1}$, which was significantly greater than that measured to the polythene-covered wall. This finding confirms earlier research (13) which concluded that electrostatic effects were most influential for particles less than 1 μm in diameter.

5. Conclusion.

It has been shown that the deposition of particles on indoor surfaces, which is an important modifier of indoor inhalation dose for particulate pollutants, is strongly influenced by the characteristics of those surfaces. Results suggest that indoor air pollution control strategies should exploit the sensitivity of supra-micrometre particle transport to turbulence and impaction effects and the preferential deposition of sub-micrometre particles to electrostatic surfaces.

A computational model has been shown to accurately predict the interaction of particles with smooth surfaces in a room-sized enclosure. The prediction of aerosol deposition to rough surfaces is complicated by the process of inertial impaction of particles on roughness elements. Future work will aim to incorporate this process in the model; simulation of the deposition of particles to electrostatic surfaces will also be carried out.

Acknowledgement.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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(Title) Checking of ventilation rates by CO₂ monitoring

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CHECKING OF VENTILATION RATES BY CO₂ MONITORING

Synopsis

The present paper presents results from measurements of outdoor airflow rates and air change rates carried out simultaneously with measurements of the indoor concentration of carbon dioxide (CO₂). The measurements were made both under controlled laboratory conditions and in the field. The field experiments were performed in a conference room, an assembly hall and an office room, and the laboratory investigation was carried out in a 19 m³ test chamber.

CO₂ measurements can be successfully used to estimate the outdoor airflow rate in occupied rooms, but it is vital that the methodology takes into consideration a number of possible sources of errors. For example, it can not be presupposed that steady-state conditions prevail. However, inaccuracies due to non steady-state conditions may be reduced by a thorough analysis of the measurement results. Furthermore, it is demonstrated how the air change rate can be determined using a method based on analysis of the CO₂-concentration decay in a room.

1. Introduction

The concentration of CO₂ is frequently measured with the objective of indicating the indoor air quality or to check the outdoor airflow rates in buildings. Instruments for CO₂ measurements are often easy to use but, nevertheless, the applicability of the method is often disputed. Many of the difficulties with the interpretation of the results could be avoided by primarily regarding the method as a tracer gas method. It is also vital that the user has sufficient knowledge not only about the operation of the instruments, but also about the building and ventilation system where measurements are planned to take place.

The CO₂ concentration can be used as an indicator of pollutants from humans, and a CO₂ concentration of 1000 ppm is often referred to as a limit, below which the air quality can be regarded as acceptable. According to the olf/decipol concept [1] the limit 1000 ppm represents a situation where about 18% of persons not adapted to the pollution in question, i.e. visitors to the building, are dissatisfied with the air quality, see Figure 1.

However, the use of the CO₂ concentration as an air quality indicator is only valid in situations where the release of pollutants from sources other than humans is not a dimensioning factor. In rooms designed for a relatively high occupant density, air pollutants originating from the various indoor activities often have a dominant influence on the indoor air quality, while the importance of pollutants, for example, emitted from building materials can be expected to be confined to the first couple of months after a new building has been completed.

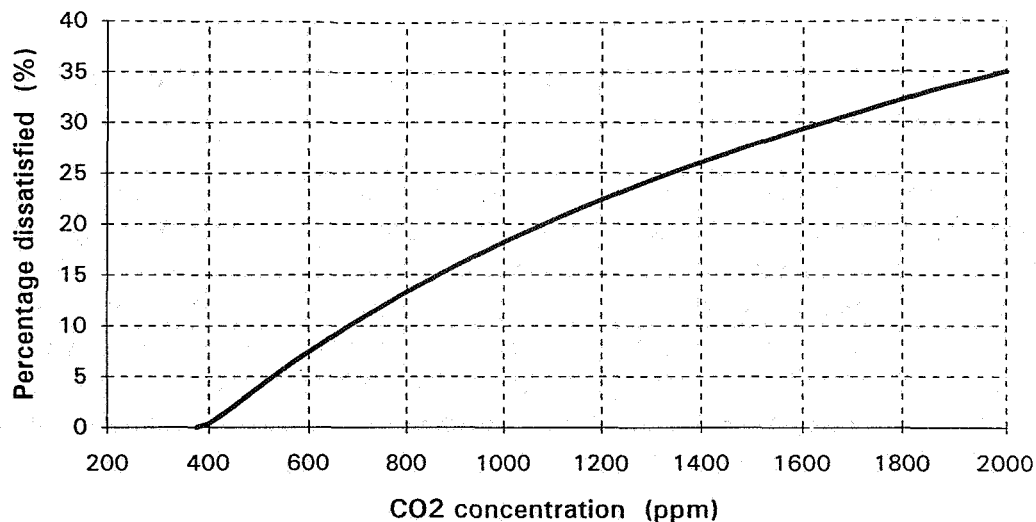


Figure 1. The perceived air quality as a function of the indoor CO₂ concentration. The function is based on the olf/decipol concept, and the calculation is carried out assuming an outdoor CO₂ concentration of 375 ppm and a CO₂ production of 18 l·h⁻¹ per person.

Procedures for calculation of the outdoor airflow rate required to achieve an acceptable indoor air quality are available [2, 3]. Such calculations are based on knowledge about the relative importance of different types of pollution sources (e.g. building materials, office machines and persons) and assumptions about how the concentrations of the pollutants emitted from various sources should be added in order to describe the indoor air quality adequately. Although these assumptions differ between different recommendations, and often can be regarded as quite rough, the resulting recommendations are in approximate accordance. For example, The Nordic Committee on Building Regulations recommends 10.5 l·s⁻¹ per person in offices without tobacco smoking [3], while The American Society of Heating Ventilating and Air-Conditioning Engineers suggests a minimum of 10 l·s⁻¹ per person in the same type of room [4]. The calculation procedure presented in [2] results in recommendations over quite a wide span since a number of parameters, that may vary from case to case, are included in the calculations. The outdoor air quality, the pollution load from people depending on the physical activity and the pollution load from building materials are examples of parameters that should be considered using the olf/decipol concept.

Although different approaches are used, several international recommendations of minimum outdoor airflow rates lie between 7 and 10 l·s⁻¹ per person. Significantly higher values are often recommended for rooms where tobacco smoking may occur, or in buildings where other strong pollution sources can be expected.

Several factors influence the prevalence of discomfort, allergies and other types of health related problems. The knowledge about these factors is still limited, but there are strong indications that there is a correlation between low outdoor airflow rates and discomfort and health problems. Thus, it is important to have access to methods for checking outdoor

airflow rates. Two important features of such methods are that they should be both reliable and easy to use.

The present paper shows results from measurements of outdoor airflow rates and air change rates carried out simultaneously with measurements of the CO₂ concentration indoors. Field experiments were performed in a conference room, an assembly hall and an office room, and laboratory measurements were carried out in a full-scale test chamber. Typically, there is a good agreement between outdoor airflow rates obtained by steady-state CO₂ measurements and simultaneous measurements using calibrated throttle flanges. Furthermore, it is demonstrated how the air change rate can be accurately determined by analysis of the CO₂-concentration decay in a room.

2. Materials and Methods

The CO₂ data were collected using an instrument based on photoacoustic spectroscopy (PAS), which measures the absorption of infrared light at a wavelength of 4.4 μm . The instrument enables a maximum sampling frequency of 2 min^{-1} and a computer-controlled solenoid valve system enabled automatic switching between up to four sampling locations in sequence. The time required for switching between measurement locations, purging the sampling tube and taking a sample was about 2 minutes. According to the manufacturer the lowest detectable CO₂ concentration is 1.7 ppm and the repeatability of the measurement is ± 1.7 ppm. The linearity of the output signal is $\pm 1\%$ and the inaccuracy of the measured concentration is estimated not to exceed $\pm 5\%$ of the measured value, including the inaccuracies in the calibration procedure.

The equipment described above was also used for measurements of sulphur hexafluoride (SF₆) used as a tracer gas. The detection limit for SF₆ is 0.005 ppm and the repeatability is ± 0.005 ppm. The inaccuracy of the measured concentration is estimated not to exceed $\pm 5\%$ of the measured value, including the inaccuracies in the calibration procedure.

A part of the investigation was carried out in a test chamber with an internal free volume of 19.3 m^3 . The airflow rate in the chamber can be varied between a few $\text{l}\cdot\text{s}^{-1}$ and about 50 $\text{l}\cdot\text{s}^{-1}$, which means that the chamber can be operated with air change rates up to about 10 h^{-1} . The airflow rate in the chamber was determined using a calibrated throttle flange in the exhaust air duct, over which the pressure drop was measured. The chamber was operated with a static pressure of +10 Pa relative to the ambient air. At this pressure the air leakage from the chamber through its envelope is 1.0 $\text{l}\cdot\text{s}^{-1}$. The inaccuracy of the measured airflow rate in the test chamber is estimated not to exceed $\pm 5\%$ of the measured value. Also in the field study, the airflow rates were determined using throttle flanges with a maximum inaccuracy of $\pm 5\%$.

3. Outdoor CO₂ concentrations

Figure 2 shows the result from measurements of CO₂ over an eleven-day period in the outdoor air 3 meters above ground level, outside an office building located close to a busy street in Göteborg. The CO₂ concentrations show pronounced variations with time, with office hour averages (8am - 5pm) between 348 and 416 ppm. The average concentration for the entire period presented is 377 ppm, and the maximum value is 435 ppm. Furthermore, it should be noted that the measured CO₂ concentrations show a significant co-variation with simultaneously measured concentrations of carbon monoxide. Carbon monoxide in the outdoor air can often be used as an indicator of traffic emissions. Consequently, there is a strong indication that the observed variations of the outdoor CO₂ concentration is a result of varying traffic intensity on the street close to the measurement location. Similar results have been presented previously [5], including the possible influence of both traffic and other outdoor pollution sources.

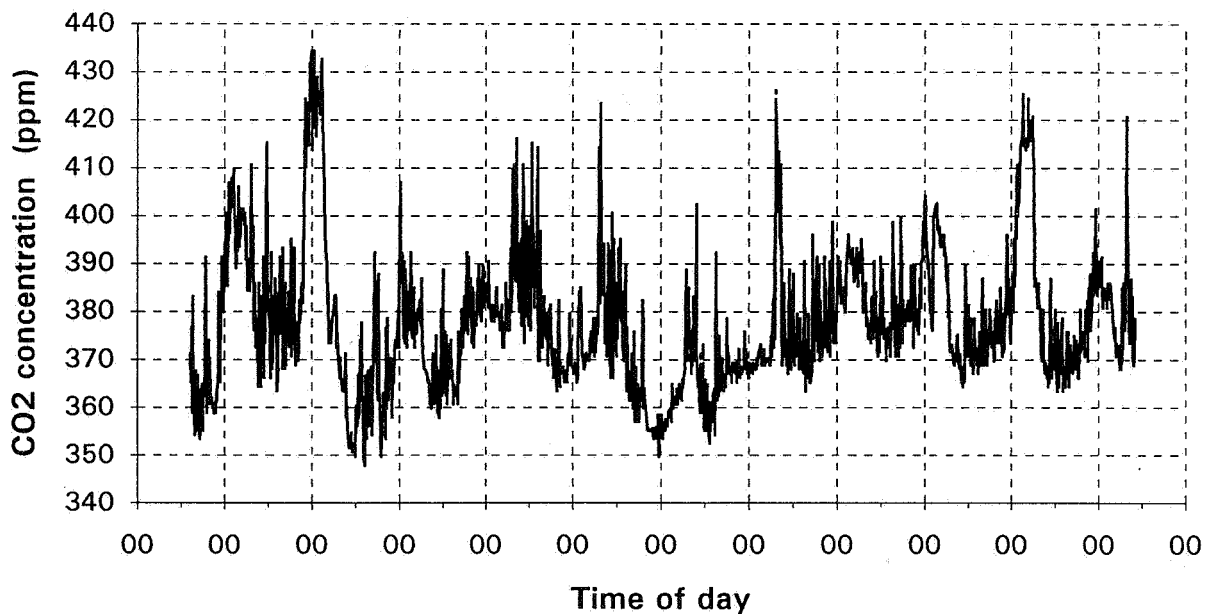


Figure 2. Concentrations of CO₂ outside an office building located close to a busy street in Göteborg.

Corresponding measurements of CO₂ were also carried out in a location not directly influenced by traffic or other sources of outdoor pollution. These measurements were made in the countryside about 25 km from the centre of Göteborg, and in this case the concentrations ranged from a minimum of 355 ppm up to a maximum of 375 ppm, with an average value of 363 ppm. This set of data did not show any drastic variations with time.

4. Occupant-generated CO₂

The human production of CO₂ and other bioeffluents have been found to vary approximately linearly with the level of physical activity [2, 4]. For adults the metabolic CO₂ production varies from about 10 l·h⁻¹ per person when sleeping up to about 170 l·h⁻¹ per person at high levels of physical activity. For persons at sedentary activities (1.0-1.2 met), such as office work, the CO₂ production is about 19 l·h⁻¹ per person [2]. The corresponding figures according to [4] are 18 l·h⁻¹ per person at a metabolic rate of 1.2 mets. However, the CO₂ production may vary between individuals depending on, for example, body weight. Physical activities in the range between sedentary and high level exercise correspond to the range 1 to 10 mets (1 met equals 58 W/m² skin area, i.e. approximately 100 W for an average person). Also the generation of olfactory pollutants can be expected to increase with the metabolic rate.

For children, the relationship between the CO₂ production and the metabolic rate is different compared to that for adults. In, for example, kindergardens an activity level of 2.7 met, corresponding to a CO₂ production of 18 l·h⁻¹ per person is a realistic assumption [2]. Also in schools with children aged between 14 and 16 years, the CO₂ production is about equal to that for an adult at sedentary activity, 19 l·h⁻¹ per person. It may be noted that, according to the same reference, the olfactory pollution load per person is about 20-30% higher for children than for adults producing the same amount of CO₂.

5. Determination of outdoor airflow rates

If a tracer gas is released at a constant rate in a room the steady state concentration in the exhaust air will equal the sum of the outdoor concentration and the concentration determined by the ratio between the tracer gas generation rate and the outdoor airflow rate. If the ventilation is characterised by complete mixing the steady state concentration will be equal in each location of the room.

As indicated in the previous section, a CO₂ production of 18 l·h⁻¹ per person can be assumed in many cases. The outdoor airflow rate can then be calculated from equation (1) if the measurement is carried out under conditions of steady-state.

$$\dot{V} = \frac{5000}{C_i - C_o} \quad (1)$$

where

$\dot{V}$ = outdoor airflow rate (l/s, p)

C_i = indoor concentration (ppm)

C_o = outdoor concentration (ppm)

A prerequisite for a correct application of equation (1) is that neither of the following assumptions are violated:

- steady-state conditions prevail
- the CO₂ production is constant
- the outdoor concentration is correctly assessed
- the measured indoor CO₂ concentration represents the average concentration in the room.

Measurements in a conference room showed a good agreement between airflow rates measured using a throttle flange in the supply air duct and the airflow rates calculated from measured steady-state CO₂ concentrations. The measurements were carried out in 1991 in a location not directly influenced by outdoor sources of CO₂. The data presented below refer to measurements when the room was occupied by 14 adults at sedentary activities and it is assumed that the CO₂ production was 18 l·h⁻¹ per person, and that the outdoor CO₂ concentration was 360 ppm. After 3 hours both the CO₂ concentration and the supply airflow rate were almost constant at 840 ppm and 145 l·s⁻¹ respectively. From the CO₂ data the airflow rate could be calculated to 146 l·s⁻¹.

Good agreement between measured and calculated airflow rates were also shown at tests carried out in an assembly hall dimensioned for 60 persons [6]. The ventilation of the hall is provided by a system with variable-air-volume (VAV system). The results from three tests performed with 30, 51 and 44 persons respectively are presented below. Steady-state conditions prevailed during the first two tests, while equilibrium was not reached during the last one.

With 30 persons in the hall, the measured CO₂ concentration was 750 ppm and the measured airflow rate was 430 l·s⁻¹. A calculation of the outdoor airflow rate in a way corresponding to the procedure described above, gives a value that is 10% less, i.e. about 385 l·s⁻¹. With 51 persons in the hall, the measured CO₂ concentration and the measured airflow rate was 820 ppm and 550 l·s⁻¹ respectively, which is in good agreement with the calculated value 554 l·s⁻¹.

As mentioned, equilibrium was not reached during the test with 44 persons in the hall. The test was terminated 50 minutes after the occupants entered the hall. At this time, the CO₂ concentration was 1020 ppm and the supply airflow rate 270 l·s⁻¹. Considering the free volume of the hall (390 m³), the CO₂ concentration can be expected to have reached about 87% of the steady-state concentration, assuming conditions of complete mixing. The calculated airflow rate will then equal 290 l·s⁻¹, which is about 7% higher than the airflow rate measured with the throttle flange.

6. Non steady-state conditions

If equilibrium is not reached during the measurement period the data cannot directly be used for determination of the outdoor airflow rate. However, if the measurement period is sufficiently long, it is possible to make a more or less rough estimation of the steady-state concentration by regression analysis of the data using an exponential function. From the steady-state concentration calculated, the outdoor airflow rate can then be obtained using a mathematical relation similar to that shown in equation (1).

Field measurements

Other methods for analysis of non steady-state CO₂ concentrations can be based on determination of the time constant for the ventilation system. Such a method is to measure the decay of CO₂ when the room has been left unoccupied after a certain period of occupancy. Figure 3 shows the result from such a measurement carried out in an office room ventilated by mechanical exhaust ventilation. The sampling interval was 2 minutes and a sampling sequence with two samples in the exhaust air followed by one sample in the outdoor air was selected. The time constant obtained from this set of data is 0.58 h, which corresponds to an air change rate of 1.72 h⁻¹. Before an exponential curve was fitted to the measured indoor concentration the background outdoor concentrations were subtracted. Before the person left the room an attempt was made to mix the air by walking around in the room for a few minutes.

Figure 4 shows a simultaneously measured decay curve for the tracer gas sulphur hexafluoride (SF₆). The air change rate obtained by the SF₆ measurement is 1.64 h⁻¹, which deviates by 5% from the rate obtained by the CO₂ measurement.

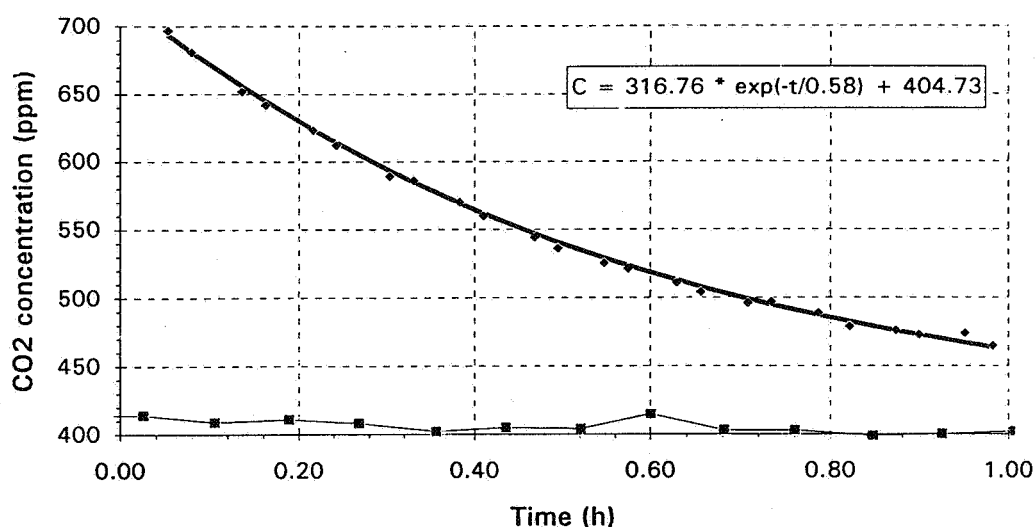


Figure 3. The CO₂ concentration in an office room with mechanical exhaust ventilation. Prior to the measurements the room was occupied by one person, who left the room at the time t=0.

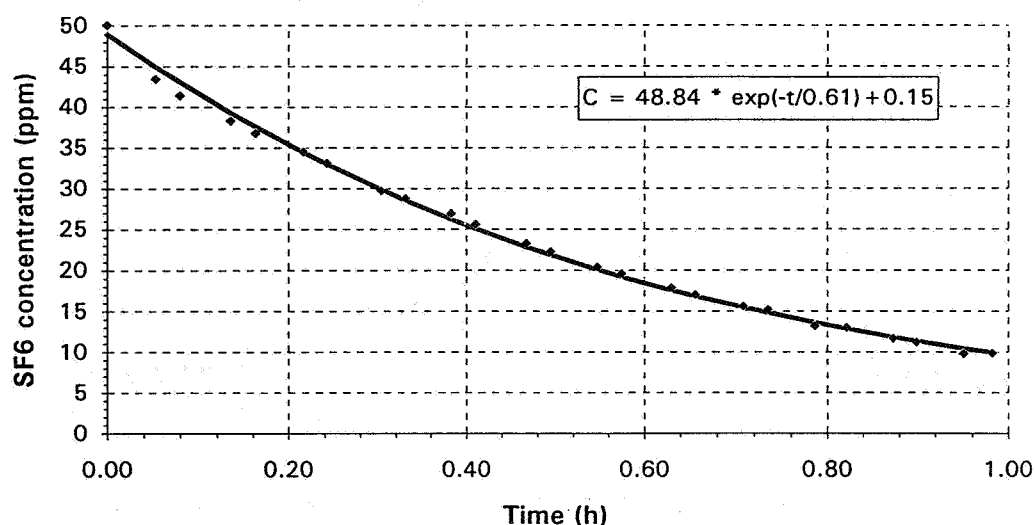


Figure 4. The decay of SF₆ measured in the office room simultaneous with the CO₂ concentration measurements shown in Figure 3.

Laboratory measurements

The results from measurements carried out in the test chamber described above are shown in Figure 5. The capacity of the supply air fan and the exhaust air fan were controlled in order to maintain an air change rate of about 2 h⁻¹. When using a throttle flange the exhaust airflow rate was determined to 10.8 l·s⁻¹, while measurements using an exhaust-hood/anemometer attached to the exhaust outlet gave the result 10.2 l·s⁻¹.

The period of occupancy was not long enough to reach concentration equilibrium, but the steady-state concentration (above the outdoor concentration) can be estimated to 497 ppm by extrapolation of the measured data. Under the assumption that the CO₂ production in the chamber was 18 l·h⁻¹, this procedure indicates an outdoor airflow rate of 10.1 l/s. The time constant for the ventilation system is calculated to 27 minutes by analysis of the concentration decay curve measured after the person had left the chamber. The internal free volume of the chamber is 19.3 m³, which gives an outdoor airflow rate of 12.1 l/s.

Since the air leakage through the envelope of the chamber is known (1.0 l·s⁻¹ at a static pressure of +10 Pa) the results from all of the methods used can be compared. A summary of the outdoor airflow rates determined with the different methods is given in Table 1.

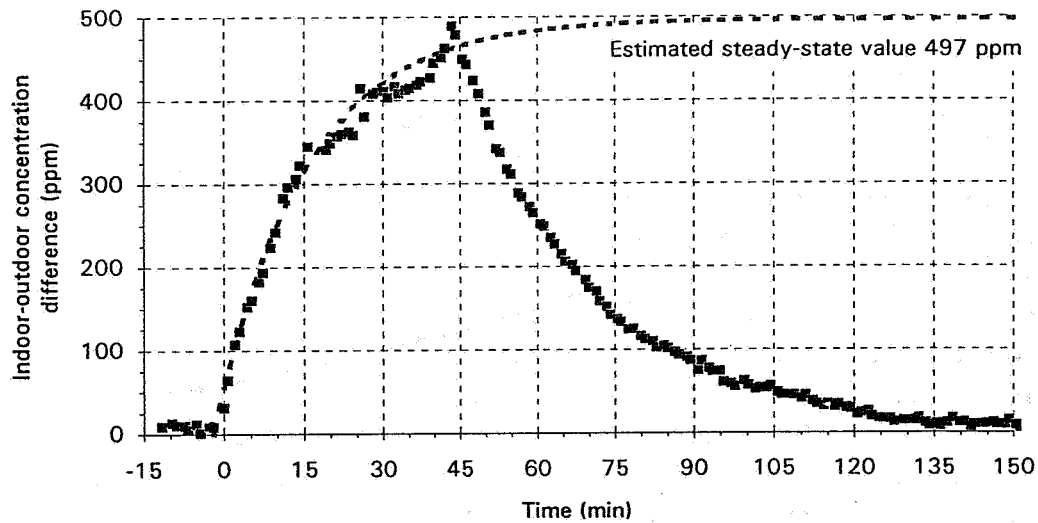


Figure 5. CO₂ concentrations measured in a full-scale test chamber. The broken line shows extrapolated data using an exponential function.

Table 1. The results obtained by different methods for determination of the outdoor airflow rate in the test chamber.

Method	Outdoor airflow rate (l·s ⁻¹)
Throttle flange	11.8
Exhaust-hood / anemometer	11.2
CO ₂ -concentration decay	12.1
CO ₂ steady-state estimation	10.1

7. Discussion and Conclusions

Measurements of CO₂ concentrations indoors can be a valuable tool for determining the outdoor airflow rates and air change rates. However, there are a number of factors that may lead to unacceptable measurement errors, but accurate results can often be obtained if the planning of the measurements include consideration of these possible sources of errors. It is a prerequisite that the measurements are carried out by a person who

- is familiar with the operation of the instruments being used,
- is aware of the limitations of the method, and
- has sufficient knowledge about the building and the ventilation system where measurements are to be carried out.

It cannot be presupposed that the outdoor CO₂ concentration is constant with time, especially not in locations influenced by vehicle exhaust. In addition to traffic, industries and other outdoor sources may influence the CO₂ concentration in the outdoor air. Therefore, measurements in the outdoor air should be carried out continuously or at least before and after the indoor measurements. If more than one instrument is used in the same investigation, the correlation between the concentrations measured by the different instruments should be established.

Assessment of the outdoor airflow rate under steady-state conditions requires accurate information about the total CO₂ production in the room, and that occupancy is constant during a period long enough to allow the concentration to reach equilibrium. This is easily checked in single rooms, but may be practically impossible to check if the measurement is carried out with the objective of determining the airflow rate in a whole building.

The air change rate can also be accurately determined by analysis of the decay of the CO₂ concentration in a room. One prerequisite for this is that the initial concentration is sufficiently high compared to the outdoor concentration.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

Reducing draught problems in the cold working rooms

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SYNOPSIS

This study is done by order of a Finnish air cooler manufacturer and it is founded to a master's thesis of the same subject matter. The company produces e.g. unit air coolers with electric fans and so the aim of the study was to reduce draught problems caused by cold air jets.

The desired values were:

- air throw 50 percent better than before
- draught free working area 30 percent larger than before

Because there are two different ways to reach and maintain low temperatures in cold working rooms we first compare the advantages and disadvantages of both method and then get acquainted with cold air jets and reducing draught problems they can cause.

The cooling coil can be mounted in the ventilating unit in which case the cold air has to be blown into rooms through ducts and conventional air outlets. Cooling coils can also be mounted directly to the ceiling in cold rooms. In that case there is often used forced convection which is carried out with electric fans. The forced convection improves heat transfer and so it is possible to make coils of compact size. The installation with cooler coils and electric fans is called unit air cooler.

The former method makes it possible to have silent cold rooms with good and even air distribution. The air distribution can be still improved by using porous textile ducts and let the cold air flow through the cloth into the room. However, there can be some hygienic problems. Hygienic problems can be minimised with the latter method, unit air coolers, because they are quite easy to maintain and keep clean. The disadvantage is that there can be some noise and draught problems. Noise problems can be solved by using fans with low RPM, but the draught problem still remains. It can be reduced with air deflectors which intensify the so-called Coanda effect. The results of this study are utilised in a novel series of air coolers.

1. COMPARISON OF THE COOLING METHODS OF COLD ROOMS

As already mentioned, in practice there are two basic methods to make cool and maintain low temperatures in cold rooms: cooling coil (evaporator) in the ventilating unit or directly in the cold room. If the cooling coil is mounted in the ventilating unit the cold air has to be distributed into rooms through ducts. If it is mounted separately in the cold room there are two alternative manners to handle the cooled air: it can simply be blown into the room with fans or there can be textile ducts or perforated sheet metal ducts which give good air distribution with low air velocity. In this report we classify the ventilation units with cooler and unit air coolers with ducts into same group and unit air coolers with deflectors into their own. So we can compare the two groups to each other.

1.1. Air distribution with ducts /1/

These solutions have the following advantages:

- even distribution of cooling effect
- good air mixing (especially with textile ducts or perforated sheet metal ducts; air can also be distributed through ordinary supply air outlets)
- possible to limit fan noise region outside of the working area (especially with the ventilation unit + cooler)
- possible to direct the cooling effect to limited region

There are also some disadvantages:

- the water condensation risk (with sheet metal ducts and non-porous, perforated textile ducts)
- the risks of bacteria (porous textile ducts)
- increased need of fan power
- location constraints caused by form and size of the ducts (Figure 1)
- cleaning of the ducts

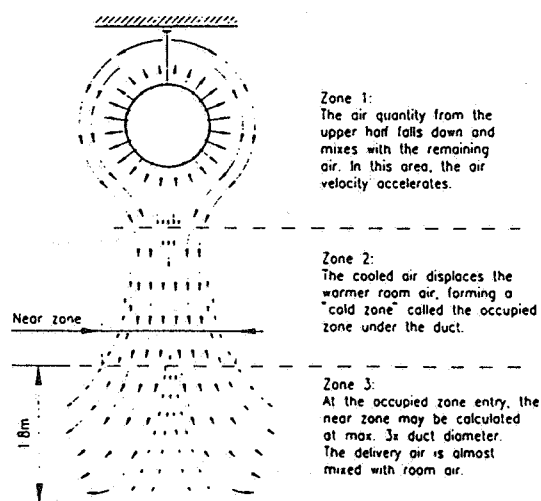


Figure 1 /2/

It is important that the considered face velocity is not exceeded because the air velocity rises beneath the duct due to the temperature difference. The maximal air velocity can be 2.5-4 times as much as the face velocity when the temperature difference is 4 K. So there should not be used big temperature differences due to the small air mixing between surroundings and delivery air. The picture included visualises air flow beneath a textile duct. /2/

1.2. Unit air cooler with deflector /1/

When using unit air coolers (cooling coils + fan), there are advantages as follows:

- possibilities to adjust the air throw with deflectors and achieve desired air circulation
- the deflectors can be adjusted after the cooler installation so that the air throw can be readjusted when the needs have changed
- easy to maintain and keep clean = less bacteric problems

There are some restrictive factors which should be known when designing a cold working room with these coolers intending to avoid draught problems, e. g. attention should be paid to constructional things such as beams and lighting fixtures because they can turn air jets and get them to the occupation zone too early.

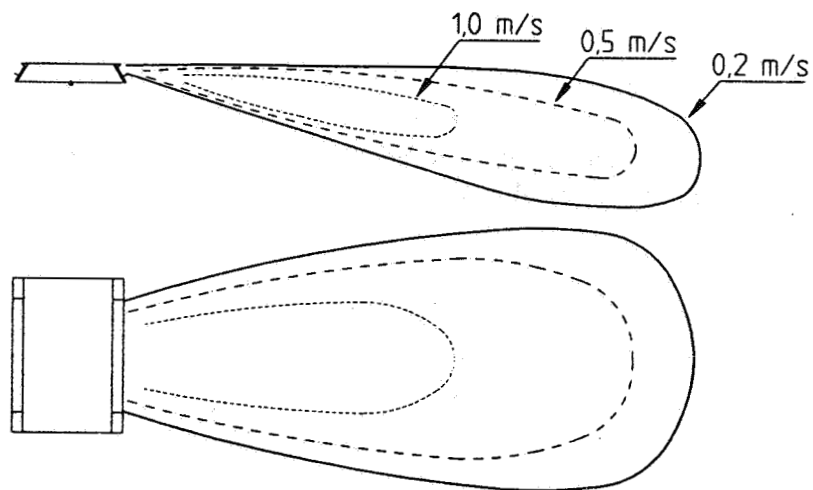


Figure 2. Unit air cooler with deflector.

2. THE COANDA EFFECT

When an air jet is blown parallel with a wall or ceiling it will seize to the surface if the air outlet is near enough. This can be explained with the Bernoulli's law: the sum of the dynamic and static pressure is constant, so, the higher is the dynamic pressure (due to the air velocity) the lower is the static pressure. So, when comparing the air jet and surroundings there is a difference between the static pressures. The pressure difference makes air flow in the direction of the lower pressure. This causes surrounding air to mix to the jet. When the air jet is flowing near a surface the mixing is possible only on the free side of the air. It can also be imagined that when the force caused by the pressure difference is bigger than the opposite directed force caused by the differences of air density the air jet keeps to the ceiling; when the velocity lowers so that the pressure force is lower than the density force the jet begins to bend downwards and then the jet loosens from the ceiling. The distance between air outlet and the loosening point can be estimated with the equation

$$x = d_e \cdot 0.63 \cdot \frac{K}{[K_\Theta \cdot Ar]^{1/2}} \quad (1)$$

where

x is the distance between air outlet and the loosening point

d_e is diameter of the air outlet

K is an air throw coefficient

K_Θ is a temperature coefficient

It is also known that the Coanda effect doesn't exist when the distance between air outlet and the ceiling is more than 300 mm. However, it can be put into effect by directing the jet slant towards the ceiling surface.

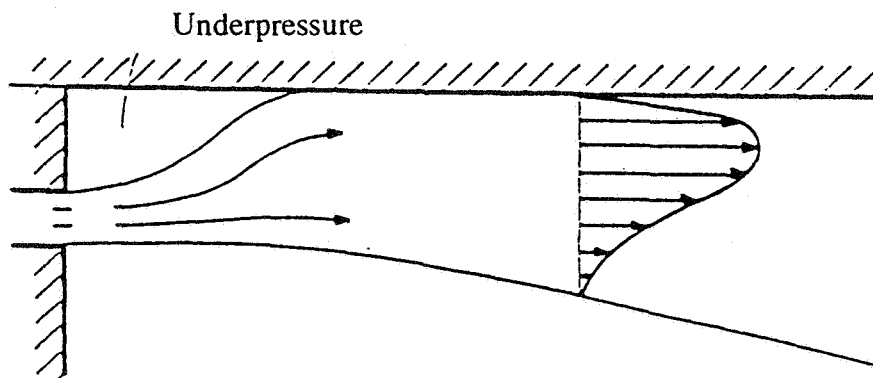


Figure 3. The Coanda effect.

3. THE NEW SERIES OF UNIT AIR COOLERS

Figures 4 and 5 present an air cooler from the new series. Note the air deflector with constant angle on the lower edge of air outlet and adjustable deflectors in the outlet. The cooling capacity output values cover an area of 4-48 kW and numbers of fans are 1-4. The deflectors make it e. g. possible to compensate the influence of distance between the cooler and ceiling. If the distance is long, then the jet can be directed up towards the ceiling and make the Coanda effect work.

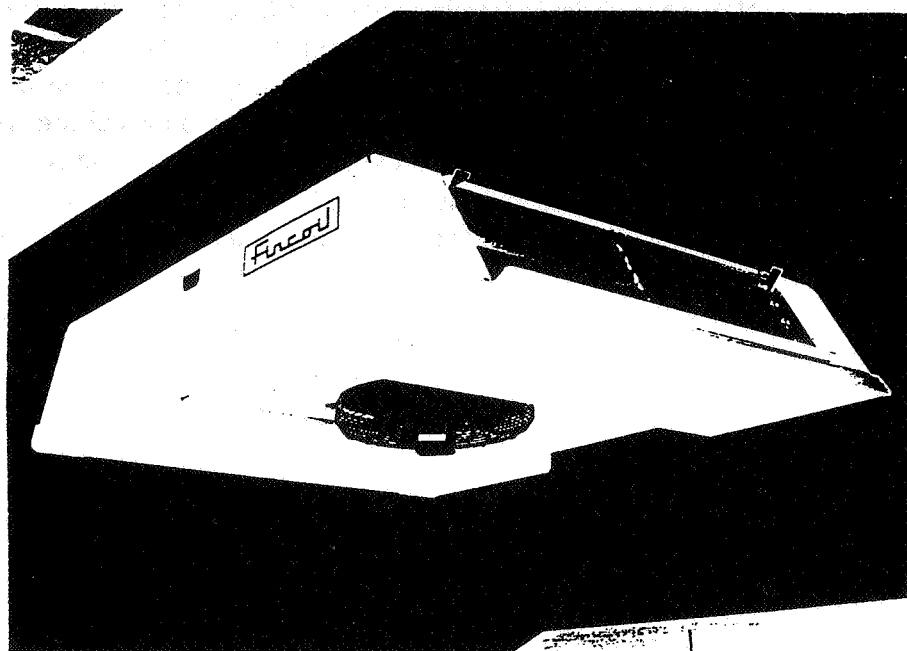


Figure 4. Polar Bear Duo - a new air cooler

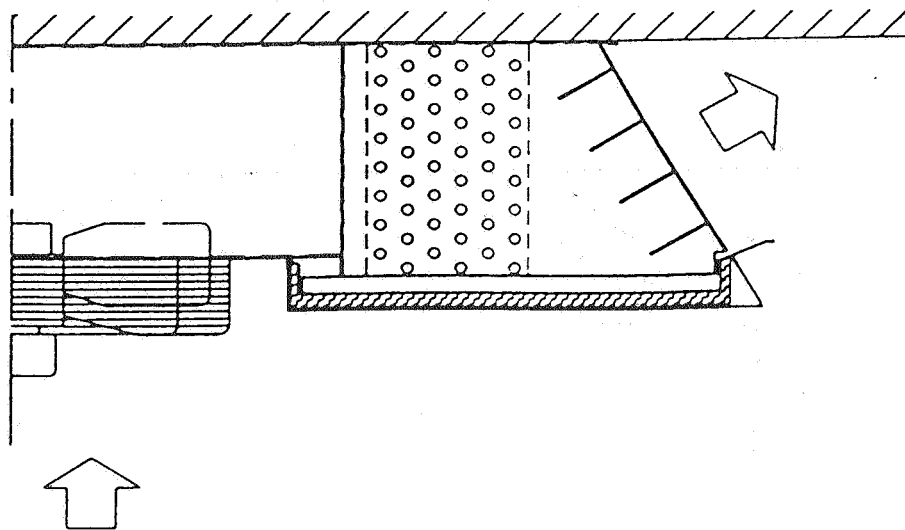


Figure 5. Cross Section of the Polar Bear Duo. Note air deflectors in the outlet.

4. SOME HINTS TO COLD ROOM LAYOUT

When the cooling is proposed to carry out with this kind of coolers the correct positioning of them is as important as it will be when using porous or perforated ducts as air distributors. Wrongly selected and positioned coolers can cause serious draught problems and make the workers complain about it.

First the air throw of the desired air velocity should not be longer than the distance to the opposite wall. This can cause draught near the wall. If the air flow is insufficient with suitable air throw there should be chosen a larger device if the smaller one can not be positioned better. As we know, the air flow (m^3/s) is the air outlet area multiplied with air velocity, so the larger is the area the lower is air velocities we need. This leads to shorter air throw. (Figure 6.)

Second the coolers should not be installed as the Figure 7 shows. The air jets will make each other to bend steeply to occupied zone. A better way is also shown in the same picture.

Third there should not be any beams and light fittings so that they can disturb the air flow. Instead of this the coolers have to be positioned so that air jets flow parallel with the beams and lamp boxes. Figure 8.

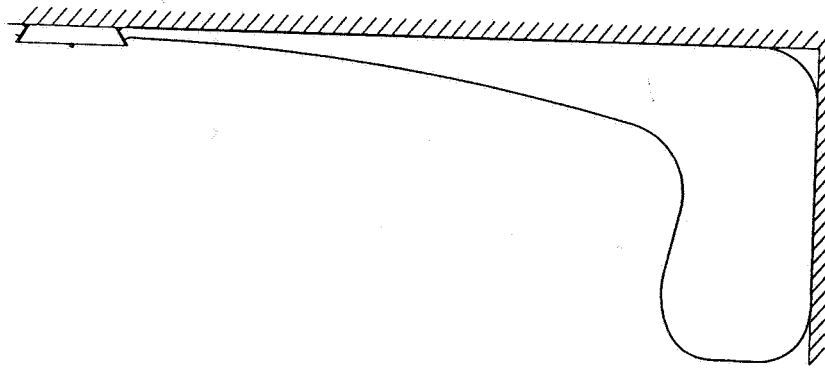


Figure 6. The air throw is too long to this room.

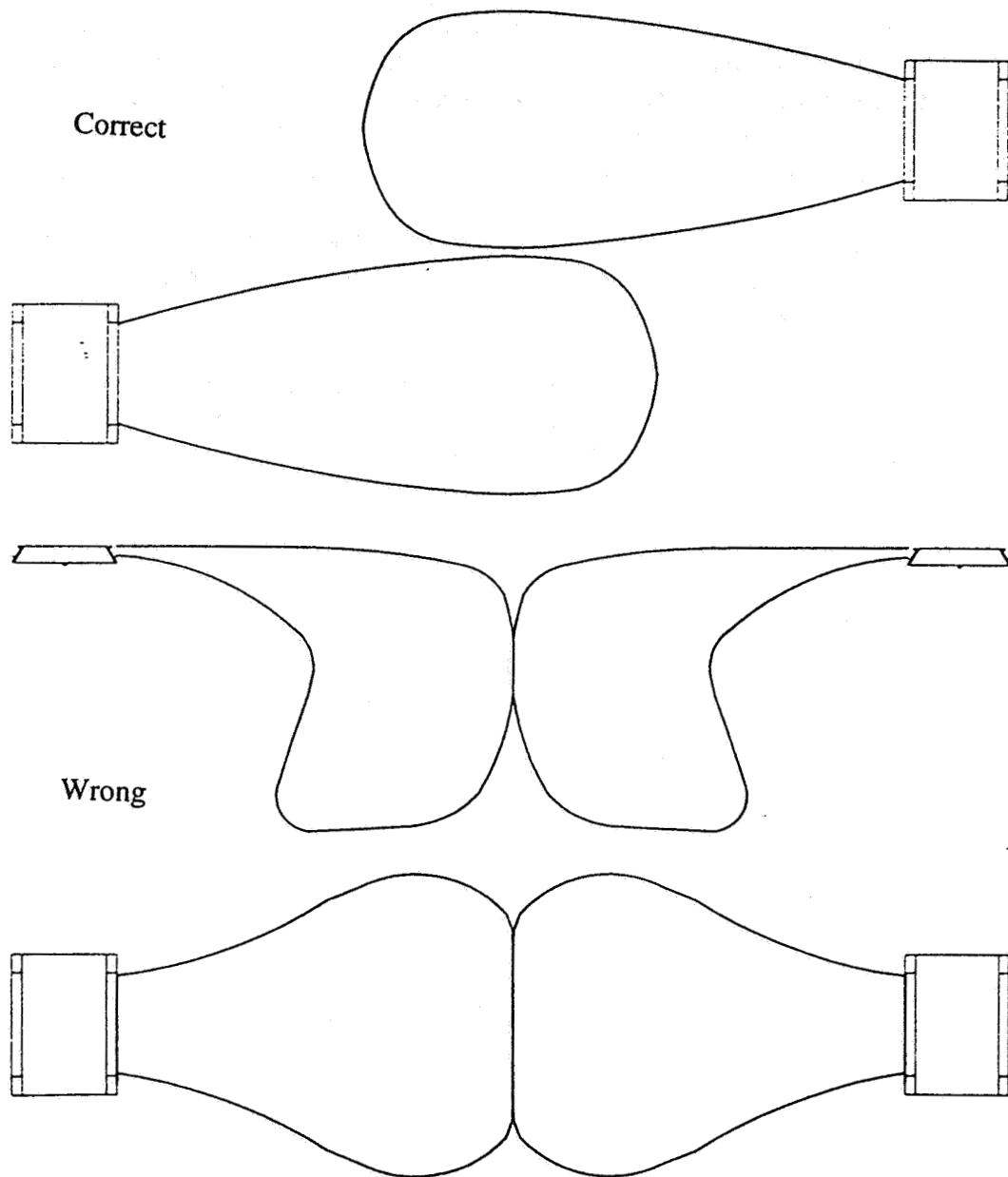


Figure 7. Correct and wrong manners to locate air coolers.

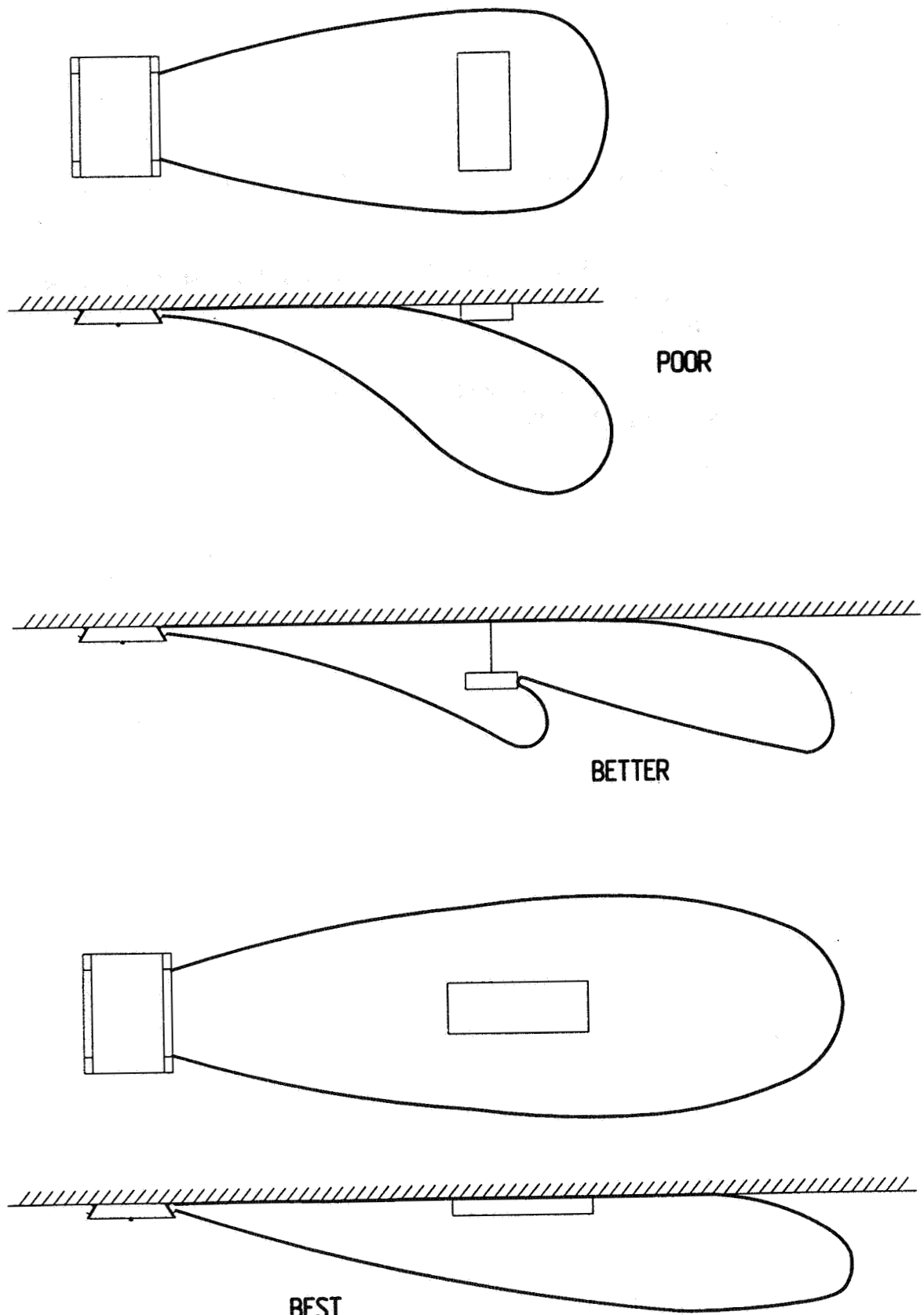


Figure 8.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

(Title)

**AIOLOS : CREATION OF AN EDUCATIONAL STRUCTURE ON THE USE OF
PASSIVE COOLING VENTILATION TECHNIQUES FOR BUILDINGS**

(Authors)

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AIOLOS : CREATION OF AN EDUCATIONAL STRUCTURE ON THE USE OF PASSIVE COOLING VENTILATION TECHNIQUES FOR BUILDINGS

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ABSTRACT

AIOLOS is a project partly financed by the European Commission, DG XVII for Energy, within the frame of the ALTENER programme. The purpose of the project is the creation and dissemination of educational material on the use of passive ventilation cooling systems and techniques as applied in buildings. All information will be coupled with the problems of passive solar heating, daylighting and visual comfort. The project is oriented to South European but also North European countries with moderate climate presenting cooling problems. Existing information and knowledge together with recent European research results will be compiled into an educational package. The package will include brochures of case studies, slides, didactic software, technical manuals, guidelines and a handbook with basic and up-to-date knowledge acquired by recent research projects in the field. The aim of the project is to provide building professionals with all necessary knowledge and tools on the efficient use of passive ventilation in buildings. All material will be in a flexible form so that it can be updated on a regular basis. Dissemination will be achieved by seminars attended by building professionals. The overall duration of the project is 18 months.

1. INTRODUCTION

Recent research activities have shown that application of passive cooling techniques in Europe can significantly reduce the energy consumption for cooling purposes. Research in the field of building related topics has lately made such rapid progress that it has been difficult to pass new technological achievements to professionals. Lack of knowledge transfer in the field of passive cooling technologies has created confusion regarding their

application, which is the main reason for non appropriate use of passive elements and techniques in buildings.

Evaluation of existing educational activities regarding the integration of passive cooling techniques in building design has shown that the related topics are rarely treated as an entity involving all building professionals. Appropriate education of all professionals involved in the field of building design could lead to energy conscious building design.

AIOLOS is a project partly financed by the European Commission, DG XVII for Energy, within the frame of the ALTENER programme. The project, having an overall duration of 18 months, is currently in progress and it is due to expire in May 1997. The University of Athens acts as a coordinator, with the participation of six European Research Institutions:

- CONPHOEBUS (Italy)
- LASH/ENTPE (France)
- University of La Rochelle (France)
- BBRI
- University of Porto (Portugal)
- University of Seville (Spain)

The general aim of the AIOLOS project is to create educational material on the efficient use of natural ventilation for buildings. All material should have a complete, flexible, modular and transferable educational structure, which should have the following specific targets:

- To evaluate and translate all existing information as well as the knowledge acquired within the frame of the European research on passive ventilation of buildings into a complete educational package dealing with the efficient use of passive ventilation for buildings.
- To create the necessary educational infrastructure that can be transferred to all educational activities and can be used by all professionals involved in the field of buildings.
- To provide building professionals with all necessary knowledge, tools and information on the efficient use of passive ventilation in buildings in order to decrease the energy consumption for cooling purposes, increase the indoor thermal comfort levels and improve indoor air quality.

Based on these targets, the work of AIOLOS project will result in the following deliverables:

- a handbook on Natural ventilation
- a series of case studies on naturally ventilated buildings
- a tool for the evaluation of the performance of natural ventilation as well as the thermal performance of buildings where it is applied as a cooling technique

This paper presents a description of the contents of the handbook, a presentation of the didactic software and an analysis of the case studies that have been carried out within the framework of the AIOLOS project.

2. A HANDBOOK ON NATURAL VENTILATION

In order to fill an existing gap in the current bibliography concerning the topic of natural ventilation, a handbook has been under development within the frame of the AIOLOS project. The handbook is addressed to architects, engineers, technicians and, in general, scientists involved in the field of building physics. Its contents cover a wide range of issues related to the subject, including fundamental as well as state of the art information. Practical methods to integrate natural ventilation in urban, suburban and rural environment are discussed together with coupling procedures with conventional systems, thermal comfort notions and indoor air quality aspects. Evaluation methods for the efficiency of natural ventilation and examples of successful design are presented. The book is structured in six chapters.

The first chapter is introductory, presenting passive cooling as an alternative concept to deal with energy and environmental problems caused by the excessive use of air conditioning systems. Natural ventilation is introduced as a passive cooling strategy.

In the second chapter, the basic principles of natural ventilation are described. The interaction of the wind with built environment is presented in a section on air flow in urban environment. Existing methods for wind data transfer from standard meteo stations to specific locations and heights are presented. The impact of various parameters, physical and geometrical on the ventilation efficiency is discussed. Specifically, the impact of climatic/microclimatic parameters on the promotion of natural ventilation is analyzed and the combined effect of the wind and buoyancy as driving mechanisms is presented. Finally, the contribution of natural ventilation to the improvement of the indoor climate, from the aspect of thermal comfort and air quality is discussed.

The third chapter presents all the methodologies that have been developed so far for the air flow prediction. Principles on various levels of modeling are presented: empirical models and computational models. Special focus is given on the possibilities and ranges of application for each modeling level. Empirical methods and simplified models are presented together with the limits of their applicability. Computational models include network, zonal and CFD modeling.

The fourth chapter is on diagnostic techniques. Tracer gas and PFT techniques for the air flow rate measurement are presented. Advantages and disadvantages of each technique are discussed. Temperature measurements as a means to characterize indoor thermal comfort conditions are also presented.

Chapter five is on critical barriers involving all parameters that may prohibit the use of natural ventilation. These are related to daylighting, shading, acoustics, fire regulations,

building type, safety, regulations and outdoor pollution. The interaction of natural ventilation with each of these parameters is discussed.

Chapter six presents techniques and solutions from practice for a successful incorporation of natural ventilation principles in the design stage. Examples of successful designs are given.

3. THE AIOLOS DIDACTIC SOFTWARE

The aim of this tool is to help the user understand the possibilities of natural ventilation as well as the impact of its use on the thermal behavior of a building.

In order to satisfy the first aim, a multizone air flow network model has been supplied with the facility of sensitivity analysis which gives the user the possibility to study the impact of various parameters on ventilation. These parameters include building (indoor air temperature, opening dimensions) as well as climatic parameters (wind speed and direction, outdoor air temperature). The user is able to specify a variation range for his parameter of interest and the program performs successive simulations keeping the rest of the input data constant. Results are given both in analytical and in graphical form.

Another important educational feature of this tool is the “reverse process”. This feature gives the user the possibility to define a desired target under specific climatic conditions and run a simulation to calculate the parameters that will ensure that the target will be fulfilled. For example, the user might want to know for a specific case he is studying, what would be the most appropriate opening size (parameter) in order to achieve a specific number of air changes per hour (target) in a studied zone. To achieve this, the program selects the most appropriate values for the specified parameters among a very high number of simulation results.

In order to satisfy the second aim, the overall performance of a building will be assessed by thermal models which will be linked to the airflow model.

4. CASE STUDIES WITHIN AIOLOS

In order to study different aspects of natural ventilation a number of case studies will be carried out in different climates, using a variety of naturally ventilated building types. The aim of this activity is to demonstrate the natural ventilation efficiency and the impact of various design parameters (climatological and architectural) on it. For this purpose, the following buildings will be investigated.

- An office building in Athens (Greece)
- A single family residence in Porto (Portugal)
- A studio apartment in La Rochelle (France)
- A family apartment in (Italy)
- A school building in Lyon (France)

- A single family dwelling (Belgium)

This collection of buildings covers a wide range of climatic conditions and a variety of building uses. Indoor air temperature and ventilation measurements will be available for each of them in order to demonstrate their thermal behavior and the impact of natural ventilation on it. Experimental data will be used in order to perform simulations using the AIOLOS software tool. Sensitivity analysis will be carried out in order to demonstrate the efficiency of various natural ventilation configurations and techniques on the overall thermal performance of the building. A thorough description of the building characteristics together with the results from the measurement and simulation campaigns for each case study will be given in the form of brochures. The educational material developed within AIOLOS will also comprise a series of slides based on the case studies.

5. AIOLOS DISSEMINATION ACTIVITIES

Dissemination activities within the framework of AIOLOS programme involve the organization of two seminars: the first is scheduled to be held in Lyon on the 21-22 of November 1996 and the second one in Athens at the end of the programme. The seminars will be addressed to engineers, architects, designers and scientists involved in the field of building physics and will include a variety of lectures and workshops on the following topics: air flow and thermal processes in buildings, air flow modeling, coupling of air flow and thermal models, critical barriers and practical solutions for natural ventilation as well as presentation of case studies in naturally ventilated buildings.

6. CONCLUSIONS

AIOLOS is an ALTENER project on the creation of an educational structure on the use of passive cooling ventilation techniques for buildings. Within the framework of the project an educational package will be developed consisting of a handbook on natural ventilation, a number of brochures and slides from existing naturally ventilated buildings and a computational tool for educational purposes. Dissemination activities involve the organization of two seminars on the subject of natural ventilation. The project has an overall duration of 18 months and is currently in progress.

ACKNOWLEDGEMENTS

The AIOLOS project is partly financed by the Commission of the European Union, DG XVII for Energy. The authors would like to thank the Commission of the European Union for their support and cooperation.

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

Design Guidelines for Ventilation System for Pollution Control in Large, Semi-enclosed Bus Terminus

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SYNOPSIS

In response to complaints about poor air quality in bus termini, the Hong Kong Government is considering imposing legislative control over ventilation system design and operation of bus termini in Hong Kong. However, there are, as yet, no relevant air quality criteria and ventilation system design guidelines for Hong Kong. In this paper, measured air quality data for 5 semi-enclosed bus termini are reviewed. This provides a picture of the prevalent air quality in bus termini. To establish a design guide for ventilation systems for Hong Kong, a number of overseas design guides are examined and a ventilation system design method is proposed. The significance of design parameters, such as bus engine emission rates, utilization of the terminus, ventilation effectiveness, etc., are discussed.

1 Introduction

In Hong Kong, many bus termini are located on the ground floor or basement of large building complexes, adjacent to or below stations of mass-transit systems. The daily bus movements at such a transport interchange may exceed 2,500 (up to 160 during the peak hour), and the total number of commuters may reach 130,000 on a typical weekday. Often, passengers have to queue by the side of bus lanes and are exposed to bus engine exhaust. Notwithstanding that mechanical ventilation is provided for pollution control in semi-enclosed bus termini, there have been complaints about the poor air quality inside some bus termini. Although there is concurrently no statutory control over the air quality inside enclosed or semi-enclosed bus termini in Hong Kong, the Government has taken the initiative [1] to develop design guidelines and practice notes for the design and operation of ventilation systems for vehicular tunnels, car parks and transport interchanges.

Buses with diesel engines are the dominant source of pollutants in a bus terminus. Amongst various constituents of diesel bus exhaust [2-4], oxides of nitrogen, collectively quantified by NO_x (NO & NO_2), or NO_2 in particular, is the most critical pollutant in respect of the adverse impact on health of passengers, and on the visibility inside a bus terminus. Thus, the concentration of NO_x or NO_2 has been adopted as the air quality criteria in some ventilation system design guides in such applications [3,5-8].

2 Measured air quality data for five bus termini in Hong Kong

To ascertain the causes and seriousness of the air pollution problem in bus termini, the Environmental Protection Department (EPD) of the Hong Kong Government conducted air quality measurement in the summer of 1993 in 5 semi-enclosed bus termini (denoted as Sites A to E). This provides a picture of the prevalent concentration levels of pollutants in bus termini. EPD's measurement covered concentrations of SO_2 , NO_x , NO , NO_2 , CO and CO_2 . The equipment used (Table 1) were installed within a mobile laboratory. During the measurements, air samples were drawn-in continuously from a representative sampling points at 2.5 m above ground level, with measurements recorded as 5 minutes time weighted averages (TWAs). The total duration of measurement was about one month per bus terminus. The measured maximum concentrations of the gaseous pollutants in the 5 bus termini are summarized in Table 2. As air samples were not taken within the breathing zone level (i.e. at

about 1.5 m), deviations are expected between the measured air quality and that to which passengers were exposed.

Table 1 Equipment in EPD's Mobile Air Quality Monitoring Laboratory

Pollutant	Method of measurement	Manufacturer and model	Frequency of measurement	Detection limits
Sulphur dioxide	Pulsed fluorescence	Thermo Electron Model 43A (0-1 ppm)	Continuous	0.6 ppb
Oxides of Nitrogen	Chemiluminescence	Thermo Electron Model 42 (0-2 ppm)	Continuous	0.5 ppb
Carbon monoxide	Non-dispersive infrared absorption with gas filter correlation	Thermo Electron Model 48 (0-50 ppm)	Continuous	0.1 ppm
Carbon dioxide	Non-dispersive infrared absorption	Milton Roy Model 3300 (0-5000 ppm)	Continuous	400 ppm

Table 2 Five Minute & Hourly Maximum Concentrations of Gaseous Contaminants in five Bus Termini (shaded if exceeding HKAQO limits)

Contaminant		Site A	Site B	Site C	Site D	Site E	HKAQO ^[1]
SO ₂ (µg/m ³)	5 min	2623	1117	1912	1362	1395	-
	hourly	1025	910	968	299	762	800
NO _x (µg/m ³)	5 min	7891	6015	5767	4024	3752	-
	hourly	6566	4390	2805	2655	985	-
NO (µg/m ³)	5 min	4882	3898	3878	2643	2455	-
	hourly	4012	2862	1754	1632	1000	-
NO ₂ (µg/m ³)	5 min	876	798	588	1775	1233	-
	hourly	433	300	199	436	287	300
CO (mg/m ³)	5 min	32.5	11.9	8.23	24.4	21.15	-
	hourly	24.3	11.1	5.2	6	4.4	30
CO ₂ (mg/m ³)	5 min	2137	1364	2612	9260	6040	-
	hourly	1786	1314	1714	850	963	-

Comparison of the maximum pollutant concentrations (Table 2) with various industrial work place air quality standards (Table 3) shows that none of the measured maximum concentration levels exceed the recommended upper concentration limits for used in occupational health. The figures are also within the air quality limits for underground car parks recently proposed by EPD [8] (Table 3). Concentration figures (hourly maximums) that exceed the respective limits set by the Hong Kong air quality objectives (HKAQO) [1] are shaded in Table 2.

3 Observed causes of poor air quality in bus termini

The dimensions, percentage of confinement (percentage of the perimeter of a bus terminus that is blocked), existing ventilation systems and utilization of the 5 bus termini studied are summarised in Table 4. Amongst them, Sites A and B had the poorest air quality (Table 2). This may be ascribed to the fact that these two sites have the highest percentage of confinement (~ 90%, Table 4) which made natural cross ventilation almost impossible. Thus, pollutant removal and dilution relied almost solely on the effectiveness of the mechanical ventilation systems.

Table 3 Comparison on various Industrial Work Place Standards

Contaminant	EPD[8] Proposed Guideline for Carpark	OSHA ² US	Industrial Workplace Standards		
			ACGIH ³ US	HSE ⁴ UK	OEL ⁵ HK
SO ₂ (µg/m ³)	-	13000 STEL 5000 TWA	13000 STEL 5200 TWA	13000 STEL* 5000 TWA	13000STEL 5200TWA
NO (µg/m ³)	-	- 30000 TWA	Subject to BEI 31000 TWA	45000 STEL 30000 TWA	- 31000STEL
NO ₂ (µg/m ³)	1800STEL 5mins.	1800 STEL -	9400 STEL 5600 TWA	9000 STEL 5000 TWA	9400STEL 5600TWA
CO (mg/m ³)	115STEL 5 mins.	229 Ceiling 40TWA	Subject to BEI 29 TWA	330 STEL 55 TWA	330STEL 55TWA
CO ₂ (mg/m ³)	-	54000 STEL 18000 TWA	54000 STEL 9000 TWA	27000 STEL 9000 TWA	54000STEL 9000TWA

1 Hong Kong Air Quality Objectives: Hourly maximum limits

2 Occupational Safety & Health Administration, Department of Labor. "Code of Federal Regulations 29 Part 1910, Revised as July 1, 1992". Final rule limits.

3 American Conference of Government Industrial Hygienists. "Threshold Limit Values 1992-1993".

4 Health & Safety Executive. "Occupational Exposure Limits 1993".

5 A Reference Note on Occupational Exposure Limits for Chemical Substances in the Work Environment, Labour Department, Hong Kong, 1992.

STEL 15-minute TWA Short Term Exposure limit.
TWA 8-hour time weighted average.

* 10-minute TWA Short Term Exposure Limit.

Table 4 Summary of Dimension and Design Ventilation Rates of the 5 Bus Termini (Figures in brackets are measured values)

Bus Terminus	Appr. area m ²	Appr. Ht. m	Confine ment %	Ventilation system	Total air flow m ³ /s	ACH ¹	l/s per m ²	Bus no. at peak hour	Daily bus no.	Remarks
Site A	109 x 82	5.3	90	Exhaust only at H/L	120 (48.93)	9.12 (3.7)	13.43 (5.47)	36	484	cross vent minimal
Site B	203 x 73	11/9/7.4	89	Supply at H & exhaust at H/L	150 (162.4) supply*	3.36 (3.6)*	10.12 (10.96)*	110	1404	cross vent minimal
Site C	109.7 x 42.2	6.59	46	Exhaust only at H/L	104.92	12.38	22.66	95	1350	good cross vent
Site D	132 x 57.5	7.79	46.2	Supply only at H	142.38	8.67	18.76	167	2505	good cross vent
Site E	138.5 x 90.5	6.2	77	Exhaust only H/L	376.78	17.45	30.06	67	858	cross vent possible

H High level H/L High and low levels * only 50% of the flow rate operated over 80% of the time in a day.

Site investigations were undertaken in the 5 bus termini and ventilation rate measurements were conducted in Sites A and B to identify the cause of poor indoor air quality in these bus termini. The observed causes, which are believed to be rather typical for similar bus termini in Hong Kong, include:

- Utilization was underestimated or has grown beyond the expected limit.
- Fresh air intake and exhaust locations were inappropriate.
- Insufficient space for installation which led to duct size reductions, and thus reduction in air flow rate, and equipment inaccessible for maintenance.

- The ventilation system not properly balanced and/or the flowrate being maintained not verified since the system was installed, or after subsequent modifications.
- Ventilation systems are manually operated or timer controlled, without regard to the actual pollutant concentrations inside the bus terminus.
- Vehicle engines, particularly those of air-conditioned buses (and taxis), remain running when queuing up inside a bus terminus.
- Inadequate maintenance (e.g. in some termini, low level exhaust grilles were damaged by vehicle, and intake grilles and filters were clogged-up by accumulated dust).

4 Ventilation system design for bus terminus

To maintain acceptable environmental conditions inside a semi-enclosed bus terminus requires a properly designed ventilation system. For this, a design guide needs to be established which includes the threshold pollutant concentration and the method to determine the required ventilation rate. In the absence of such a guideline for Hong Kong, several relevant design guides, standards or design examples available from the literature [3, 6-8 & 12] were studied. These methods were applied to Site B to give a comparison on the ventilation system that will result by using each of these methods to design the mechanical ventilation system. The respective design criteria and the ventilation rate calculated are summarised in Table 5. The results of applying the guidelines for all the 5 termini are summarised in Table 6.

For the CIBSE Guide [5] and the design example of Cockram and Bearman [7], the pollutant concentration limit was set at 5 ppm of NO_x and the ventilation requirement was 80 times the exhaust volume. The pollutant concentration criterion adopted in the ASHRAE Handbook [3] and in Brociner's example [6] was NO_2 concentration of 5 ppm and ventilation requirement was 75 times the exhaust volume. There are no exact ventilation requirement specified for bus terminus in the Australian Standard [11]. Criteria for enclosed drive-in-facilities were selected but, the routing and frequency of use of a driveway in a bus terminus may be very different. Considerable deviations amongst the required ventilation rates determined from these design guides exist (Table 5). Also, the design (or measured) ventilation rates for these termini were greater than or close to the ventilation rates determined using these guides (Table 6) but the air quality was still found unacceptable.

The intake air were assumed to be free of any pollutants in all the above guides and design examples. However, in many cases (e.g. Site A, C and E), the roadside air, which is drawn-in via openings or driveways for ventilating a bus terminus, is already contaminated by the heavy traffic (e.g. the roadside NO_x concentration in Kwun Tong was 0.132 ppm [13]). Therefore, the quality of the intake air should be included in the method for determining the required ventilation rate.

Instead of the abovementioned guides, equation (1), derived from first principles on the basis of a simplified perfectly mixed condition [12], is recommended for use in determining the required ventilation rate for a semi-enclosed bus terminus :

$$V = \frac{N(E_1 t_1 + E_2 t_2 + E_3 t_3 + E_4 t_4)(t_1 + t_2 + t_3 + t_4)}{3600(C_m - C_o)\eta_{vent}} \quad (1)$$

where:	V	=	ventilation rate (m ³ /s)
	N	=	max. total no. of buses entering the terminus during the peak hour (no. of buses/hour)
	E ₁	=	emission rate of a bus during deceleration mode (mg/s)
	t ₁	=	total duration of a bus under deceleration mode (s)
	E ₂	=	emission rate of a bus during idling mode (mg/s)
	t ₂	=	total duration of a bus under idling mode (s)
	E ₃	=	emission of a bus during acceleration mode (mg/s)
	t ₃	=	total duration of a bus under acceleration mode (s)
	E ₄	=	emission of a bus during cruising mode (mg/s)
	t ₄	=	total duration of a bus under cruising mode (s)
	C _m	=	allowable maximum concentration (mg/m ³)
	C _o	=	concentration in ambient air (mg/m ³)
	η _{vent}	=	ventilation effectiveness factor

Results of applying this method to all the five sites are included in Table 6.

Table 5 Relevant Standards and Guidelines on Ventilation Design for Bus Terminus and Their Application to Site B Bus Terminus

Standard/ Guide	Particular Area	Criteria	Ventilation Requirement	Ventilation Requirement for Site B				
				For Site B	Supply Rate m ³ /s	Exhaust Rate m ³ /s	Total Vent Rate m ³ /s	Tot. Ve nt Rate *l/s per m ²
ASHRAE [3]	drive-way	to maintain visibility and control odor, limit NO ₂ to 5ppm	75 times vehicle exhaust	75 x 109 l/s per bus x 10.39 buses		84.94		
	partially enclosed platform	pressurization of the platform	85 l/s per m ² of platform area		34		118.94	11.47
New York example [6]	totally enclosed platform	limit NO ₂ to 5ppm pressurization of the platform	10 l/s per m ² of platform area	75 x 109 l/s x 10.39	4	84.94	88.94	8.57
CIBSE [5]	general	below 5ppm NO _x	80 times vehicle exhaust	80 x 73.32 l/s x 10.39 buses		60.94	60.94	5.88
London example [7]	enclosed parking space	below 5ppm NO _x	4.9m ³ /s per engine running	4.9x110/4		134.75	134.75	13
Australian Standard 1668.2 [11]**	enclosed driveways associated with bldgs	-	200 l/s per metre length of each lane	200 x 54 x 8		86.4	86.4	8.33
Equation (1)		limit to 1ppm NO ₂					370	25

* Based on an area of 10,373 m² which excluded the area for the taxi and mini-bus lanes. This area is about 70% of the total area of Site B

** There are no exact ventilation requirement guidelines for bus terminus in this Standard. Criteria for enclosed drive-in-facilities were selected for comparison.

Table 6 Application of design guideline to all five bus termini

	Site A	Site B	Site C	Site D	Site E
Approximate area m ²	8938	10373	4629	7590	12534
Max. bus no. in peak hour	36	110	95	167	67
Minmium flow rate					
Original design value					
supply/exhaust m ³ /s	0/120(49)	140(162)/60	0/104.92	142.38/0	0/376.78
total m ³ /s	120	140#	104.92	142.38	376.78
l/s per m ²	13.43	13.50	22.66	18.76	30.06
ASHARE [3]					
supply/exhaust m ³ /s	46.33 /27.8	34 /84.9	34.94 /73.33	56.1 /129.92	47.1 /51.75
total m ³ /s	74.13	118.94	108.27	186.02	98.85
l/s per m ²	8.29	11.47	23.39	24.51	7.89
New York [6]					
supply/exhaust m ³ /s	5.45/27.8	4/84.94	4.11/73.33	6.6/129.92	5.54/51.75
total m ³ /s	33.25	8.94	77.44	136.52	57.29
l/s per m ²	3.7	8.57	16.73	17.99	4.57
CIBSE [5]					
supply or exhaust m ³ /s	19.67	60.77	51.90	91.24	36.62
total m ³ /s	19.67	60.77	51.90	91.24	36.62
l/s per m ²	2.2	5.86	11.5	12.02	2.92
London [7]					
supply or exhaust m ³ /s	44.1	134.75	116.38	204.58	82.08
total m ³ /s	44.1	134.75	116.38	204.58	82.08
l/s per m ²	4.93	13	25.14	26.95	6.55
AS1668 [11]					
length of driveway m	432	545	411	660	554
supply or exhaust m ³ /s	109	86.4	82.2	132	110.8
total m ³ /s	109	86.4	82.2	132	110.8
l/s per m ²	12.20	8.33	17.76	17.4	8.84
Equation (1)*					
required flowrate m ³ /s	121	370	320	561	225

* $C_m(\text{NO}_2) = 1.8 \text{ mg/m}^3$, $C_o(\text{NO}_2) = 0.0752 \text{ mg/m}^3$, η_{vent} assumed to be 0.5

Figure in () indicate flowrate measured on site in m³/s

50% part load operating most of the time during normal week day.

In respect of the design criteria in air quality (C_m), the Hong Kong air quality objectives (HKAQO) [1] is by far the only established air quality standard in Hong Kong. However, the HKAQO figures (close to those for outdoor in ASHRAE Standard 62-89 [9]) are meant to be a primary air quality guideline and is not applicable to bus termini.

Industrial workplace air quality standards (e.g. NO_2 concentration of 5 ppm as STEL, Table 3) appear to be a more relevant reference and are the basis of the design guides reviewed above. However, industrial workplace standards are, in general, applicable to healthy adults only but users of bus termini include children, the elderly and people of varied health conditions. EPD has proposed to use 1 ppm NO_2 (1.8 mg/m³, 5 minutes STEL) as the criteria for enclosed carparks with diesel vehicles [8]. Although commuters will stay inside a bus terminus for a longer time than they would in carparks, the same is recommended for bus termini.

The ventilation rate required to dilute the pollutants, and hence to maintain an acceptable air quality, is directly proportional to the rates that pollutants are produced in a given space. Therefore, the likely peak rate of pollutant emission needs to be determined. However, diesel

bus engine exhaust data from different sources vary largely. The following are a few examples:

- a) The following bus exhaust rates were given in ASHRAE 1991 [3] (engine size is not specified):

Mode of operation	idling	accelerating	cruising	decelerating
Exhaust volume flow rate l/s	55	225	163	143

The average durations of bus movements observed in Site B were 120s, 20s, 120s and 20s for the idling, accelerating, cruising and decelerating modes respectively. This set of operating time is adopted in calculating ventilation rate using equation (1) (results shown in Table 5 and 6). The average exhaust, according to the figures above is 109 l/s. This value, however, is much greater than that estimated from the following 2 sets of data from UK [6,7].

- b) Homles (1988) [10] gives the following bus exhaust rates:

Engine size/ mode of operation	idling 320 rpm	full load 800 rpm
6 litres	16 l/s	40 l/s
8 litres	21.33 l/s	53.33 l/s
10 litres	26.66 l/s	66.66 l/s

Assuming that the time for idling and full load is at the ratio of 3:1, the average exhaust will be 36.66 l/s for a 10 litres engine. When the exhaust rate under the acceleration mode is accounted for by doubling the give exhaust rate, the total exhaust rate becomes 73.32 l/s (adopted as the emission data for the calculations by CIBSE, see Table 5 and 6).

- c) According to Cockram 1982 [7], the mean exhaust volume flow rate over 15 minutes was estimated to be 61 l/s (adopted as emission data for London example in Table 5 and 6) for a 12 litres engine that runs at 396 rpm for 13 minutes and at 1998 rpm for 2 minutes [7].

As a much lower NO₂ concentration limit (1 ppm c.f. 5 ppm) was adopted, and outdoor air concentration was accounted for, in using equation (1) to determine the required ventilation rates for the 5 bus termini in Hong Kong, the ventilation rates are much higher than those determined from the other guidelines (Tables 5 & 6).

Equation (1) was derived on the basis of a well mixed, single zone model and its use requires detailed information on bus service frequency, bus emission rates and ambient air pollution concentrations. Accurate data however are, as yet, unavailable. Thus, further work is required to :

- obtain a set of update data on bus emission for various engine size and age under different modes of operation;
- ascertain bus termini utilisation (usage factor) and the idling, acceleration, cruising and deceleration time for bus termini of different layout designs;

- establish a database on concentration of pollutants in the ambient air at various locations in Hong Kong;
- experimentally obtain data on ventilation effectiveness for different air distribution layouts or system layouts.

5 Conclusion

There is a need to develop a practical design guideline for ventilation system in semi-enclosed bus termini in Hong Kong. A design method and an air quality criteria have been proposed in this paper. Besides a proper ventilation rate, the system layout, proper installation, testing and commissioning, operation and maintenance are important factors to maintaining acceptable air quality in bus termini.

6 Acknowledgment

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

**17th AIVC Conference, Gothenburg, Sweden,
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Cooling and ventilation of a high-speed ground transportation system

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Synopsis

This paper presents the special needs and difficulties concerning cooling and ventilation of the *SwissMetro* high-speed ground transportation system. *SwissMetro* is based on four complementary technologies: a complete underground infrastructure, a partly evacuated tunnel system to reduce the aerodynamic drag of the vehicle with a maximum speed between 400 and 500 km/h, linear electric motors and a magnetic levitation and guiding system.

Due to high internal and external heat loads permanent cooling and air-conditioning of the vehicle is required. Additional thermal problems may occur in the case of an emergency. The pressure level inside the tunnel has to be raised within a short period of time to guarantee the safety of the passengers and to avoid any health risks. This repressurization of long tunnel sections may cause shock waves and high temperatures inside the tunnel. The aim of the project is to analyse the spatial and temporal temperature evolution in the *SwissMetro* tunnels under different operating conditions and in emergency situations.

Introduction

With the introduction of the "TGV" in France, the new "Shinkansen" in Japan, the "ICE" in Germany and, to a lesser extent, the "Pendolino" trains in Italy, guided transportation systems which are commercially available, have reached their limits in terms of economy, comfort, speed and their potential for further development.

The *SwissMetro* is planned as a high speed transportation system for passengers only and within the first quarter of the 21st century is intended to contribute significantly to the linking of the major Swiss cities in an ultra high speed network. Fig.1 shows the foreseen pilot stretch, the different stages of the whole network and the connection with the European high-speed rail network.

The evolution of a new form of guided transportation for public use is an enormous, multifaceted and costly public utility project. Such a project requires several years of R&D and detailed design efforts to be completed. The nationwide project *SwissMetro* includes a variety of research activities at the Swiss Federal Institute of Technology and in the supporting Swiss industry.

Under the current ecological conditions, it would simply be impossible to construct a high-speed train system in Switzerland along the traditional lines, i.e., the TGV or the ICE. Even the German "Transrapid", which is partly based on a similar technology, would for the same reasons hardly be thinkable in Switzerland. Due to the legal and social restrictions resulting from high public sensitivity to environmental issues, the mountainous nature of the Swiss terrain and the high land values in Switzerland, the *SwissMetro* will have an entirely subterranean, tube-like infrastructure. Fig.2 shows a design study and Fig.3 shows the cross-section of the vehicle and tunnel structure.

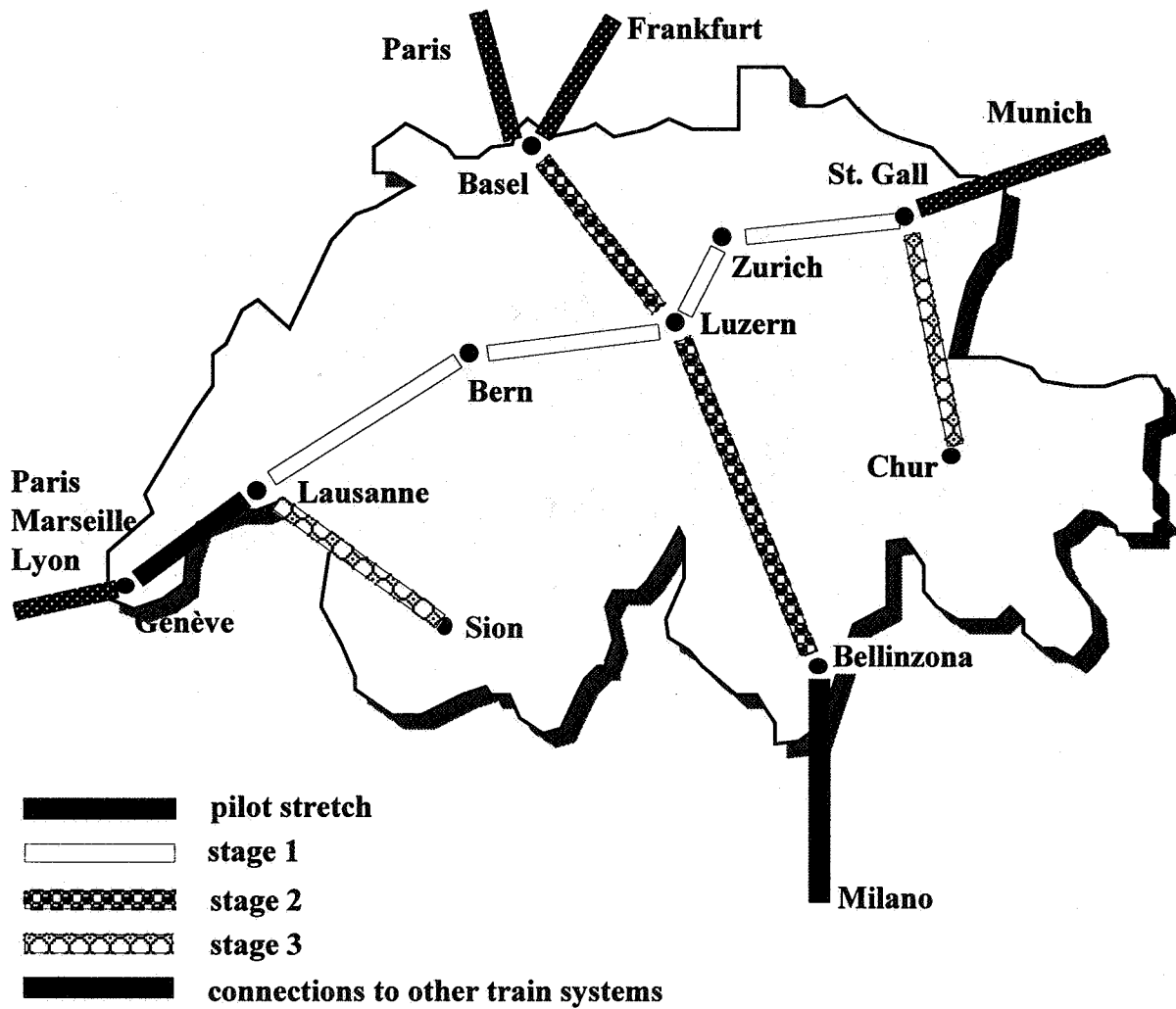


Fig.1: SwissMetro network

SwissMetro is designed to travel at speeds exceeding 400 kph. Subatmospheric operating pressures are thought to be required to reduce the friction and piston effect forces resulting from the use of very small diameter tunnels which themselves are supposed to keep tunneling costs low. The tunnel tubes will therefore be kept under partial vacuum. For SwissMetro, the classical wheel-rail system had to be dismissed, because it is difficult to master that technology at such high speeds. The vehicles will be held on track by a magnetic levitation and guidance system involving no surface contact. The propulsion force is supplied by stationary linear electric motors fitted to the track. This will ensure maintenance-free operation.

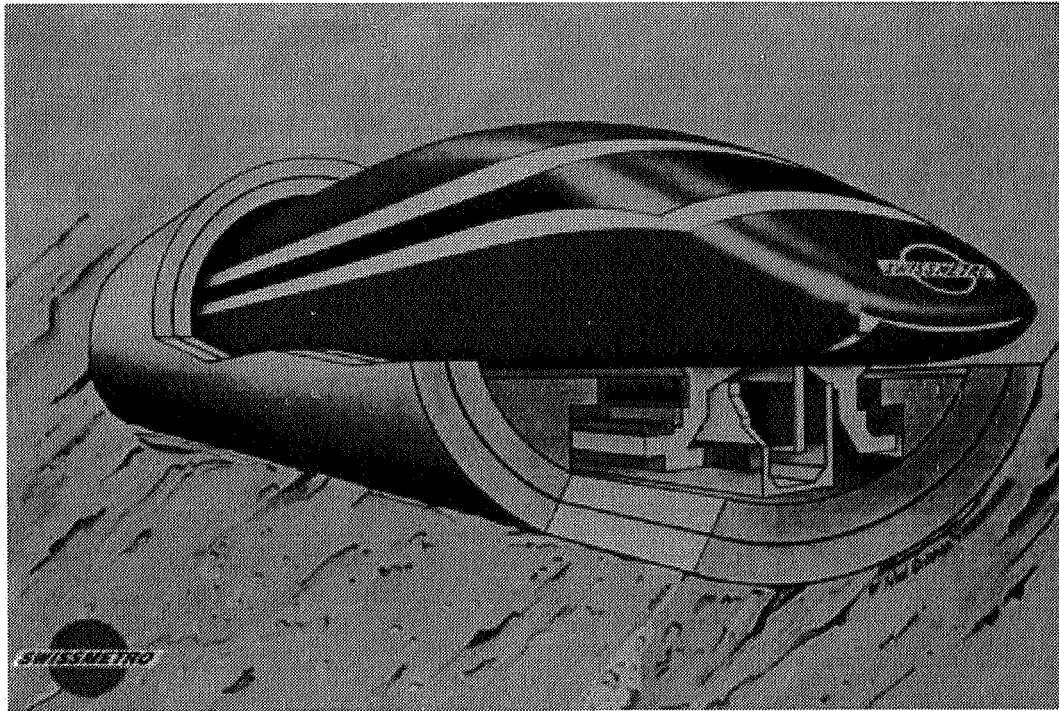


Fig.2: Design study of SwissMetro

Despite the fact that the entire system would be built underground and that very complex technology would have to be used, the costs remain comparable to those of other transport systems. A comparison of energy consumption has shown that *SwissMetro* will consume 35% less energy per person-km than an InterCity train, and some 65% less than the TGV.

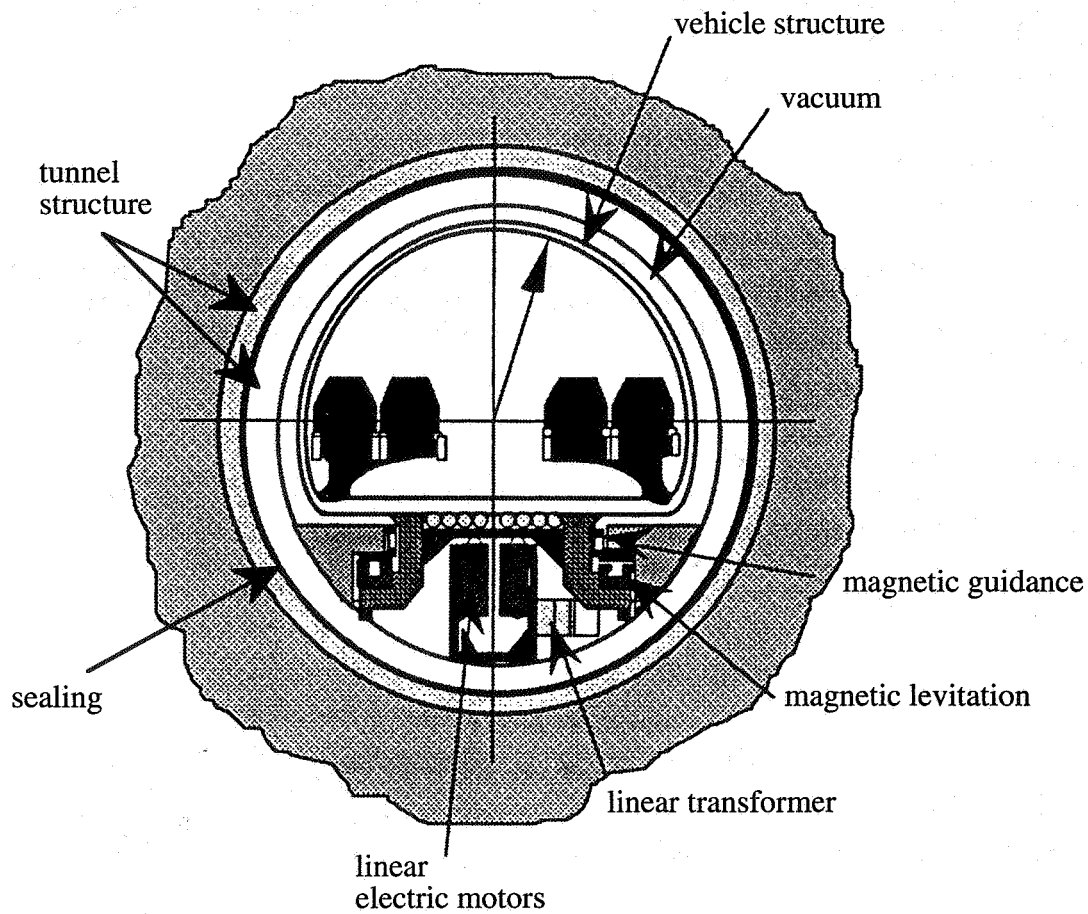


Fig.3: Cross-section of SwissMetro tunnel

Thermal conditions during normal operation

The new and unique technology of *SwissMetro* with the complete underground infrastructure and the partly evacuated tunnel system presents special challenges and difficulties concerning cooling, ventilation and air-conditioning.

Due to the reduced pressure level in the tunnels the vehicle has to be pressurized, similar to a commercial airliner. In this section the thermal balance of the vehicle and a global balance of the tunnel system are analysed.

Thermal balance of the vehicle

In this first step it is assumed that the isolation of the vehicle is very efficient. So the inside and the outside of the vehicle may be regarded separately. The air-conditioning system has to ensure thermal comfort and sufficient air quality, as well as a constant pressure level.

The main heat-sources are:

- the passengers (100 - 150 W per person)
- lighting
- the control system
- the heat-pump

These internal heat loads have to be removed constantly. In the current phase of the project, it is not determined whether the thermal energy is transferred to the rarefied tunnel atmosphere or removed during stops in stations. This requires an additional system to store the thermal energy inside the vehicle.

Global balance of the tunnel system

In contrast to the internal heat sources of the vehicle the external ones are not permanent. The most important ones are:

- Magnetic levitation and guidance
- Linear transformers
- Linear motors
- Electric power supply

The first step of this thermal analysis of the whole system (Monnier, Th.: "Bilan thermique du système Swissmetro") has shown that the tunnel and the surrounding rock will heat up during the operation and steady state temperatures may climb up to 40 °C within nearly 50 years of operation. This effect is today also visible in subway systems with a high vehicle density.

The results still contain some uncertainties, especially the aerodynamics which have a major influence on the dissipated energy of the linear motors, the most important heat source. Another critical factor is the heat storage capacity of the surrounding rock, mainly a question of geology. Also the heat transfer from the track and the tunnel air to the walls in the rarefied gas atmosphere is uncertain. As the study progresses the results have to be refined with the help of more precise input data to design a sufficient cooling and ventilation system for the vehicle, the tunnel with the electrical equipment, and the stations.

Thermodynamics of an emergency situation

Additional thermal problems may occur in the case of an emergency, especially if the vehicle body is severely damaged, a situation causing a dangerous pressure drop inside the passenger cabin.

The human body is very sensitive to any kind of pressure variation. The physiological effects are mainly a function of the pressure drop or increase rate and the lowest pressure levels.

In the case of *SwissMetro* the two most important health risks for passengers are Hypoxia and different forms of decompression sickness, also called Dysbarisms. They have a major influence on the safety aspect of the concept and will play an important role in the licensing by the safety authorities.

Hypoxia can be defined as a lack of sufficient oxygen in the body cells or tissues. It is usually caused by inadequate oxygen supply in the inspired air.

Decompression sicknesses or Dysbarisms are caused by gas expansion within or on the body and may be divided into two groups:

- Trapped gas: During increase and decrease of the pressure level free gas expands or contracts in certain body cavities. A person's inability to pass this gas may lead to abdominal pain or pain in ears and sinuses.
- Evolved gas: These conditions are produced by the low atmospheric pressures in the tunnel, similar to a flight at high altitude. Gases escaping from solution in the blood and other body tissues are responsible for conditions such as chokes, paresthesia and central nervous system problems.

These physiological effects give the time and pressure level limits which have to be observed in an emergency evacuation scenario. To avoid the negative impact of very low pressures on the human body the inside tunnel pressure has to be raised quickly. In case of a minor emergency without perforation of the vehicle, the tunnel does not have to be repressurized.

If the body of the vehicle is ruptured and the pressure inside the cabin has already dropped to the outside tunnel pressure, there is only limited time to repressurize the tunnel. In the current phase of the project it is planned to divide the tunnel into different sections which can be shut off separately. The remaining volume is either repressurized from a reservoir under atmospheric conditions, or from a high pressure reservoir. The main influence factors on the repressurization time are number, arrangement and diameter of the repressurization valves.

The results of Guigas "Etude de la repressurisation des tunnels du SWISSMETRO" have shown that shock-waves and a significant temperature raise during the repressurization may cause serious problems. To analyse the unsteady, compressible flow a 1D-model was used. To verify and refine the results currently a commercial 3D CFD Code is used to calculate the transient temperature and pressure distributions in the tunnel.

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**OPTIMUM VENTILATION AND AIR FLOW
CONTROL IN BUILDINGS**

17th AIVC Conference, Gothenburg, Sweden,

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**Experiments in Natural Ventilation
for Passive Cooling**

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Topics: passive cooling; natural ventilation; modelling; measurement; case studies

Experiments in Natural Ventilation for Passive Cooling

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SYNOPSIS

A naturally ventilated three level office building has been used to study basic stack ventilation configurations and the interaction between ventilation and the subsequent cooling of the building structure in summer. The research was performed in the framework of a European project on passive cooling of buildings and the objective was to validate simple ventilation algorithms and to give an experimental basis to design guidelines for night cooling techniques. The multilevel office allowed the studying of the influence of openings (size and position) on the neutral pressure level (NPL) and on airflow rates. Various cross-ventilation situations have been studied. A single flow path configuration was obtained by closing all windows and doors in the building envelope with the exception of the roof exhaust and one office window as the ventilation air inlet. Air flow patterns were traced with smoke and tracer gas.

In a first set of experiments, where the only driving force is stack pressure, air velocities and the position of the NPL have been measured, and contraction and velocity coefficients as used in the Bernoulli model have been observed. In a second set of experiments, the resulting effective area of a combination of two openings in series was studied.

Air flow rates derived from velocity measurements in the open doorways were found to be in agreement with the flow rates obtained with a constant injection tracer gas technique, with an uncertainty of $\pm 20\%$. Overall agreement was found between the velocity measurements and simplified models based on the Bernoulli equation. In order to cool multiple levels of a building with outside air, the position of the neutral pressure level should be controlled. The restrictions on opening size and position are discussed.

LIST OF SYMBOLS

A	Surface area	[m ²]	z	Height	[m]
C_d	Discharge coefficient	[-]	Δp	Pressure difference	[Pa]
g	Acceleration of gravity	[m/s ²]	ϵ	Jet contraction coefficient	[-]
H	Distance between opening centres	[m]	ζ	Frictional pressure drop coeff.	[-]
L	Leakage flow rate	[kg/s]	ξ	Local pressure drop coefficient	[-]
$\dot{m}$	Mass flow rate	[kg/s]	ρ	Air density	[kg/m ³]
N	Kinetic energy (Coriolis) coefficient	[-]	ϕ	Velocity coefficient	[-]
p	Pressure	[Pa]	SUFFIXES		
T	Temperature	[K, °C]	b	Bottom	t top
u	Air velocity	[m/s]	e	External	i Internal
$\dot{v}$	Volume flow rate	[m ³ /s]	eff	Effective	int Intermediate opening
W	Opening width	[m]			

1. INTRODUCTION

Reality is infinitely complex. Our knowledge of the world is always finite and therefore always incomplete. The marvel is that we function quite well in the world in spite of never fully understanding it. Our tool to deal with incomplete knowledge is modelling. A model is a small finite (incomplete) description of an infinitely complex reality for the purpose of answering particular questions. The complexity of a model depends on the kind of questions we are seeking to answer [1].

Ventilation in buildings is a complex phenomenon. A key question is what variables should be taken into account to answer a particular question. For summer natural ventilation, engineers and architects are interested to know if the ventilation rate is sufficient to extract pollutants or heat from a given space. A dominant parameter to answer this question is the opening position and size. Simple ventilation models in the pre-design phase of a project are often found to be sufficient to determine the effect of these parameters [2,3]. Moreover, in a comparison of detailed and simple models of air infiltration it was found that more complex models do not generally increase the accuracy of the result [4,5].

The well-known Bernoulli model offers valuable information on the air flow path, flow rate and both inlet and outlet velocities [6,7,8]. More recently, detailed ventilation models have been coupled with building thermal models, in order to evaluate the effect of cooling by natural ventilation [9,10], but the effect of cooling by natural ventilation can also be described by using simple models [11,12].

In all these modelling efforts, an important source of uncertainty, affecting similarly both detailed and simple models, is related to the hydraulic resistance of airflow networks. Which value to attribute to discharge coefficients and velocity coefficients? While a large compilation of coefficients can be found in the reference work of Idelchik [13], few data are available which have been measured on real buildings. Indeed, a large number of sources deal with the *coefficient* [2,3,6,8], but not so much with the factors determining its value. In the absence of better information, a general value for the discharge coefficient of a large opening of $C_d = (0.6 \pm 0.2)$ can be used [8].

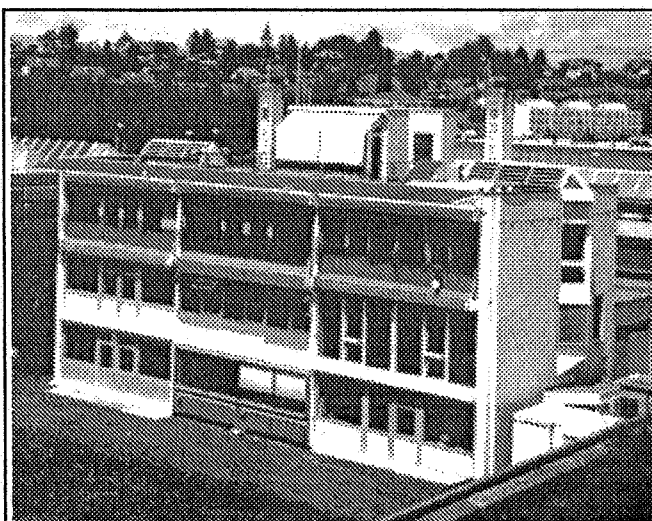


Figure 1. Outside view of the LESO building.

In order to improve our insight into the actual values of flow resistance coefficients in networks describing air flows in buildings, this paper presents new experimental results. The data have been obtained on a naturally ventilated three level office building (Figure 1) which was available for the study of basic stack ventilation configurations. The research was performed in the framework of a European research project on passive cooling of buildings [14,15] and the objective was to validate simple ventilation algorithms and to give an experimental basis to design guidelines for night cooling techniques [16,17,18].

The multilevel office allowed the studying the influence of openings (size and position) on the neutral pressure level and on airflow rates. Single sided and cross-ventilation situations have been studied. A single flow path configuration was obtained by closing all windows and doors in the building envelope with the exception of the roof exhaust and one office window as the ventilation air inlet. Air flow patterns were traced with smoke and tracer gas. Air flow rates derived from velocity measurements in the open doorways were found to be in agreement with the flow rates obtained with a constant injection tracer gas technique, with an uncertainty of $\pm 20\%$. Overall agreement was found between the velocity measurements and simplified models based on the Bernoulli equation.

The simple Bernoulli model is used to determine the influence of the size and position of openings on the natural ventilation of buildings. The position of the openings in relation to the NPL to achieve efficient cooling is also discussed.

In the experiments, air velocities and the position of the NPL have been measured, and contraction and velocity coefficients as used in the Bernoulli model have been observed, where the only driving force is stack pressure. In a second set of experiments, the resulting effective area of a combination of two openings in series was studied.

2. VENTILATION MODELLING

2.1 Stack pressure

Temperature differences between outdoors and indoors cause density differences and therefore pressure differences. When the inside temperature is higher than the outside temperature, the pressure distribution over the building height qualitatively takes the form shown in Figure 2. In this case, incoming air flows through the openings below the NPL and outlet air through the openings above it. The height at which the interior and exterior pressures are equal is called the neutral pressure level (NPL) [3]. z_{npl} is the distance of the NPL from the reference level.

In a closed airtight building the NPL is at midheight of the building. The size and position of the openings determine the location of the NPL (Figure 2 b,c,d). According to the mass conservation principle the air mass flow rate through the openings below the NPL equals the air mass flow rate through the openings above it.

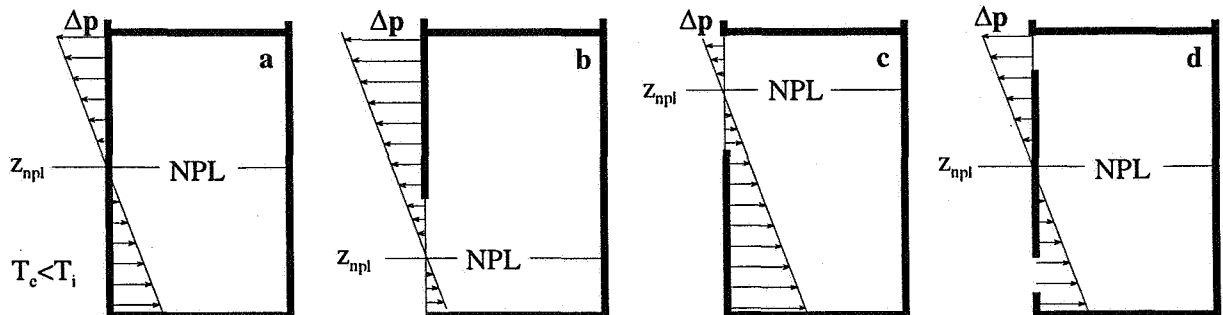


Figure 2. In-out stack pressure difference for different opening configurations.

The indoor outdoor pressure difference in an enclosure where the air is supposed to be ideal gas, frictionless, incompressible and with a homogeneous density is:

$$\Delta P_s(z) = (\rho_e - \rho_i)g(z - z_{npl}) = \rho_i g(z - z_{npl}) \frac{(T_i - T_e)}{T_e} \quad (1)$$

2.2 Bernoulli's equation

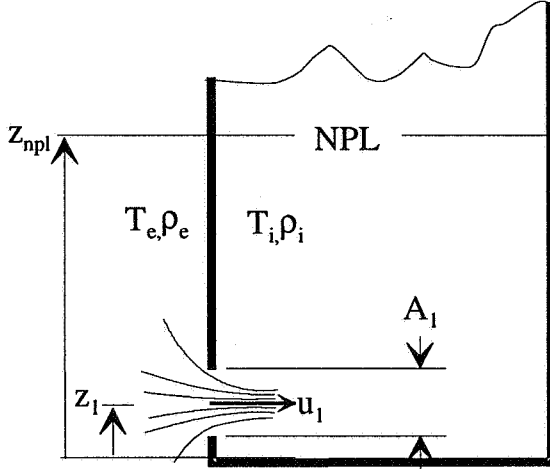


Figure 3: Air flow through an opening due to stack pressure when $T_i > T_e$.

The application of the energy conservation principle [13] between the horizontal cross section of the building at the NPL level and the vertical cross section of the opening A_1 (Figure 3) gives the Bernoulli equation. The surface area of the building cross section compared to the opening surface area is large, so velocity in this section is low. Kinetic energy on the section NPL can be neglected.

The pressure drop between the two sections is the sum of the local pressure drop in the opening and the pressure drop due to friction in the air path from A_1 at NPL. They can be normalised to the air velocity u_1 as:

$$\Delta p_{local} = \frac{1}{2} \xi \rho_e u_1^2, \quad \Delta p_{frictional} = \frac{1}{2} \zeta \rho_e u_1^2 \quad (2, 3)$$

Under these conditions Bernoulli's equation becomes:

$$\Delta p_s(z_1) - \frac{1}{2} \rho_e \xi u_1^2 - \frac{1}{2} \rho_e \zeta u_1^2 - \frac{1}{2} \rho_e N_1 u_1^2 = 0 \quad (4)$$

Air velocity in the opening can be expressed as:

$$u_1 = \varphi \sqrt{2g(z - z_{npl}) \frac{\Delta T}{T_e}} \quad (5)$$

φ is the velocity coefficient:

$$\varphi = \frac{1}{\sqrt{\zeta + \xi + N_1}} \quad (6)$$

N_1 is the kinetic energy coefficient depending on the uniformity of the velocity distribution in the opening section, ξ is the local pressure drop coefficient for the opening and ζ is the friction pressure drop coefficient [13]. In a real building ζ includes also the effects of the pressure drop due to obstacles and direction changes in the air flow path.

φ is determined experimentally and depends on the Reynolds number, on the opening nature as well as on the obstacles in the air flow path. It is difficult to distinguish the effect of each parameter included in φ , but an overall value can be found.

2.3 Neutral Pressure Level

z_{npl} is the distance of the NPL from the reference level. The mass conservation principle leads to equation (7) for an air tight enclosure with two openings, one at the top and one at the bottom.

$$z_{npl} - z_1 = \frac{H}{1 + \left[\left(\frac{A_{bot}}{A_{top}} \right)^2 \left(\frac{T_{in}}{T_{out}} \right) \right]} \quad (7)$$

H is the vertical distance between the centres of gravity of the two openings.

z_1 is the vertical distance between the centre of gravity of the bottom opening and the reference level. For convenience the reference level could be chosen such that $z_1=0$

For enclosures with a single opening, the NPL is situated in the middle of the opening height. Applying the mass conservation principle strictly, the NPL is not exactly in the middle due to the density difference of the inlet and outlet air but the error is less than 0.3% for a ΔT of 10K [8]. For more than two openings the neutral level cannot be calculated by an explicit equation and numerical resolution should be used. The software LESOCOOL [18] offers a tool to determine the NPL for an enclosure with multiple large openings. The algorithm used combines the basic principle of AIDA [7] with the full large opening algorithm [19].

2.4 Jet forming in the opening

The air passing through an opening forms a jet. The air path lines are curved on the opening edge and the air flow does not fill the whole opening section. The effective area of the opening is smaller than the geometrical one due to the contraction of the flow path lines. In section 3.2 the jet forming through an external opening is visualised using smoke (Figure 6). The jet contraction depends on the Reynolds number, on the kind of opening boards, as well as on the ratio of the section area of the opening and the section area of the space through which the air passes. In the case of outside openings this ratio is infinite [13].

The effective area of the opening depends on how the jet fills the opening section. ε is the coefficient of jet contraction (or coefficient of filling the section) and it is defined as:

$$\varepsilon = \frac{A_{effective}}{A_{geometrical}} \quad (8)$$

2.5 Air flow rate

Integration over the opening height gives the airflow rate through the opening

$$\dot{v} = \int_{z_1}^{z_2} W u dz \quad (9)$$

For a single opening this integration gives:

$$\dot{v} = \frac{1}{3} \varepsilon \varphi A \sqrt{gH \frac{\Delta T}{T}} = \frac{1}{3} C_d A \sqrt{gH \frac{\Delta T}{T}} \quad (10)$$

For two openings the air velocity can be considered constant all over the opening height if the opening is far from the NPL. The mean air velocity can be taken equal to the velocity in the centre of the opening. In this case the airflow rate is:

$$\dot{v} = AC_d \sqrt{2g(z - z_{npl}) \frac{\Delta T}{T}} \quad (11)$$

C_d is the discharge coefficient. It is the product of the velocity coefficient φ and the contraction coefficient ε .

$$C_d = \varepsilon \varphi \quad (12)$$

The discharge coefficient can be determined experimentally only if the air flow rate is directly measured. This can be done by tracer gas methods [20]. The velocity coefficient can be determined by measuring the air velocities.

2.6 Two openings in series

For a number of openings in series the mass flow is constant but there is a partial pressure drop over each opening. Because the velocities vary inversely with the opening area (continuity equation) the pressure drop over successive openings varies inversely with the square of the area [6,12].

$$\Delta p = \sum_{i=1}^n \Delta p_i = \sum_{i=1}^n \frac{\rho}{2} \left(\frac{u}{\varphi} \right)_i^2 = \frac{\dot{m}}{2\rho} \sum_{i=1}^n \left(\frac{1}{\varphi \varepsilon A} \right)_i^2 = \frac{\dot{m}}{2\rho} \sum_{i=1}^n \left(\frac{1}{C_d A} \right)_i^2 \quad (13)$$

When two openings on the same height z are placed in series an effective opening area, A_{eff} , can be defined as:

$$\left(\frac{1}{A_{eff}} \right)^2 = \left(\frac{1}{C_{d1} A_1} \right)^2 + \left(\frac{1}{C_{d2} A_2} \right)^2 \quad (14)$$

If $A_2 \gg A_1$ the pressure drop in the second opening can be neglected. If $A_2 > 1.5A_1$ the error of neglecting A_2 is less than 11%, while if $A_2 > 2A_1$, this error is less than 6%.

3. EXPERIMENTS

The naturally ventilated LESO office building (Figure 1) offers many possibilities for real scale experimental work. Its structure is ideal for ventilation experiments. It has three storeys plus a basement and a sunspace as extension of the central staircase through which the building is ventilated (Figure 7). It is 11 m high with openings available on all levels offering several opening combinations.

3.1 Velocity coefficient φ for large openings

For all experiments the building was kept closed. All internal doors towards the staircase were closed as well in order to limit the effect of leakage. The sunspace door at the top of the building was kept open (0.9m×1.98m). The bottom opening on ground level was varied. For

the experiments "b" to "f", an office window of fixed height (1.5m) and variable width (0.95m-0.2m) and for the experiment "a" the entrance door (0.8m×2.3m) were used as the bottom opening.

The experiments were performed at three different dates and the in-out temperature differences varied between 8.6 and 11.8 °C. The wind velocity was less than 0.6 m/s.

T_i is the average of four temperatures measured on every storey. The thermal stratification along the air flow path varies from 2.5 to 5.5 °C. Notice that the effects of the thermal stratification are ignored by the Bernoulli model.

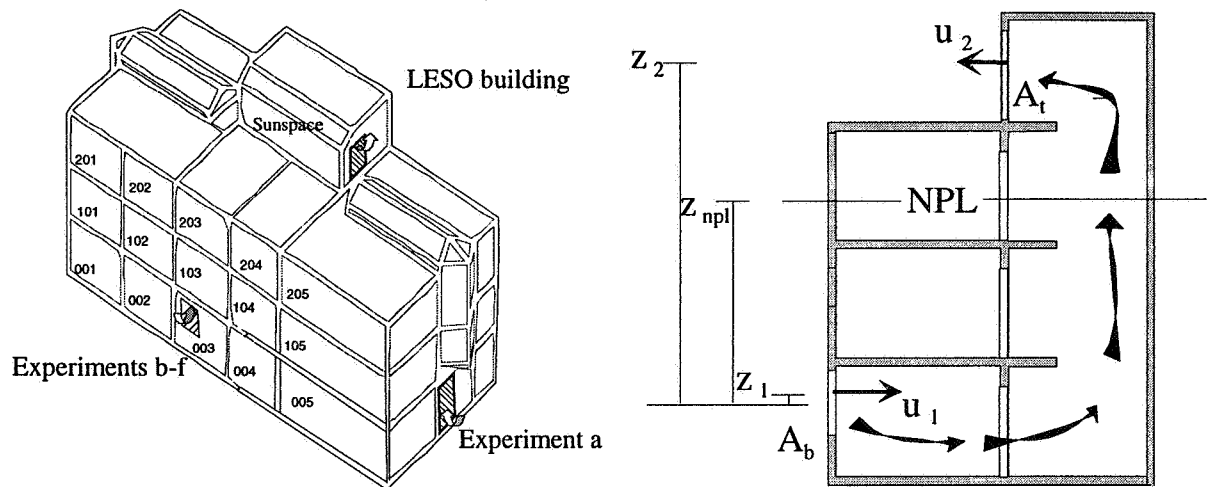


Figure 4: Openings used for the experiments of section 3.1

Figure 5 shows the measured velocity coefficients in the experiments a to f. Within the domain the experiments have been performed, no significant correlation was found between φ and the opening dimension or air velocity. Table 1 gives more details on each experiment.

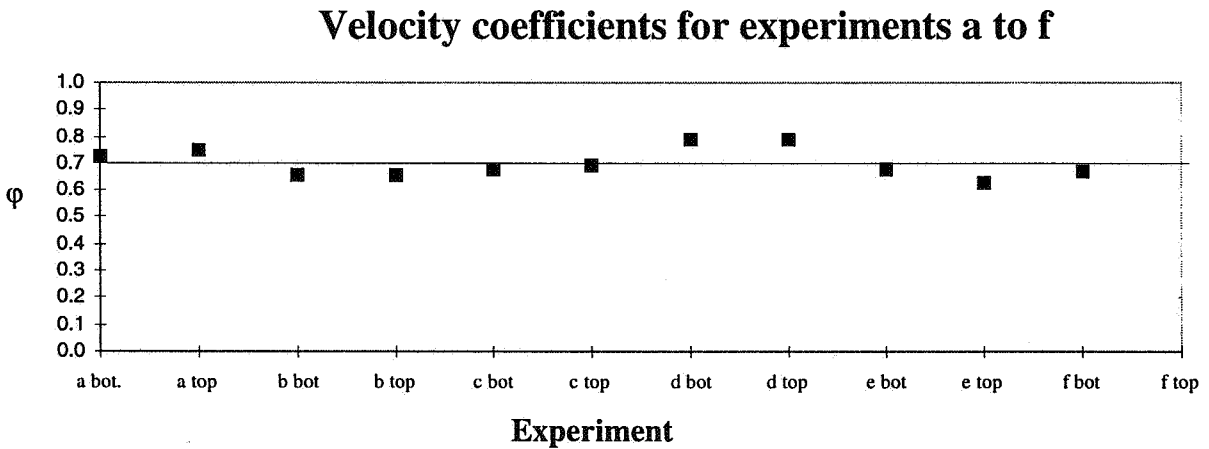


Figure 5. Top and bottom velocity coefficients measured in six experiments. The experiments are sorted according to the bottom opening dimension (Table 1)

Experiment		a	b	c	d	e	f	σ^*
A_b (bottom)	m ²	1.9	1.4	1.4	1.4	0.9	0.3	0.06
A_t (top)	m ²	1.8	1.8	1.8	1.8	1.8	1.8	0.06
T_i (internal temp.)	°C	16.2	22.1	18.9	16.7	22.1	22.1	0.3
T_e (external temp.)	°C	6.0	13.0	9.1	4.9	13.0	13.5	0.3
z_{npl}^* (NPL height)	m	4.4	6.4	6.1	6.2	8.7	10.2	0.3
u_1 (measured velocity)	m/s	1.3	1.3	1.4	1.8	1.6	1.6	0.04
u_2 (measured velocity)	m/s	1.4	1.0	1.1	1.3	0.6		0.04
ϕ_1 (top velocity coefficient)	-	0.73	0.66	0.68	0.79	0.68	0.67	0.09*
ϕ_2 (bott. velocity coefficient)	-	0.75	0.66	0.69	0.79	0.63		0.09*

Table 1. Results of the experiments a to f to determine velocity coefficient ϕ

* The mean value of the calculated velocity coefficient is 0.7. Using the standard deviations indicated in the last column of Table 1 the theoretical standard deviation for ϕ is 0.09. The standard deviation of the twelve experimental values of ϕ is 0.06 showing that the theoretical σ is pessimistic and the differences in the experimental values are due to random errors.

** Equation (7) for the NPL is an application of the mass conservation principle assuming that the building is air tight. In reality there is no absolutely air tight building. The leakage with natural ventilation depends not only on the size of the cracks but also on their location in relation to the NPL. As the second parameter changes for every experiment it is impossible to take into account a fixed leakage. For every experiment the leakage is evaluated applying

$$\varepsilon_1 A_b \rho_e u_1 - \varepsilon_2 A_t \rho_i u_2 + L = 0 \quad (15)$$

L is the leakage mass flow rate.

Taking the experimental values of u_1 and u_2 equation (15) gives the leakage, whereas the theoretical values of u_1 and u_2 give an implicit equation to calculate z_{npl} :

$$C_d A_b \rho_e \sqrt{2gZ_{npl} \frac{T_e - T_i}{T_i}} - C_d A_t \rho_i \sqrt{2g(Z_2 - Z_{npl}) \frac{T_e - T_i}{T_i}} + L = 0 \quad (16)$$

The same experimental procedure was applied to two more buildings satisfying the following conditions:

- Internal restrictions > 2 times the smaller opening
- Temperature stratification < 1 K/m
- Wind < 0.5 m/s
- Building height > 3 times opening height

The measured ϕ was also found equal to 0.7 ± 0.1 showing that this value is not specific to a particular building.

3.2 Contraction coefficient ε

The purpose of this experiment is to visualise the jet forming in the opening. The experimental set up is identical to the experiments of section 3.1. The smoke is released outside the opening 003. A video camera is placed on the opening edge, recording the air-smoke flow. A graph paper placed vertically on the window edge allows an estimation of the contraction and effective area (1 graduation = 1cm). Some digitised images for air velocities from 0.4 to 0.9 m/s are shown in Figure 6.



Figure 6 Digitised video images of the incoming air through a window of a LESO office.

The jet contraction in the opening section oscillates continually between 2 and 4 cm in the range of air velocities 0.3 to 1 m/s independently of it. Considering a contraction of 3 ± 1 cm, the coefficient ϵ for a rectangular opening of dimensions 1×1.5 m is 0.90 ± 0.03

3.3 Openings in series

In the first set of experiments, it was supposed that there was no particular resistance (restriction or considerable obstacles) between the two openings. In this second set of experiments, the windows at first floor are used, varying the area of intermediate opening (office door) in order to test the validity of equation (14).

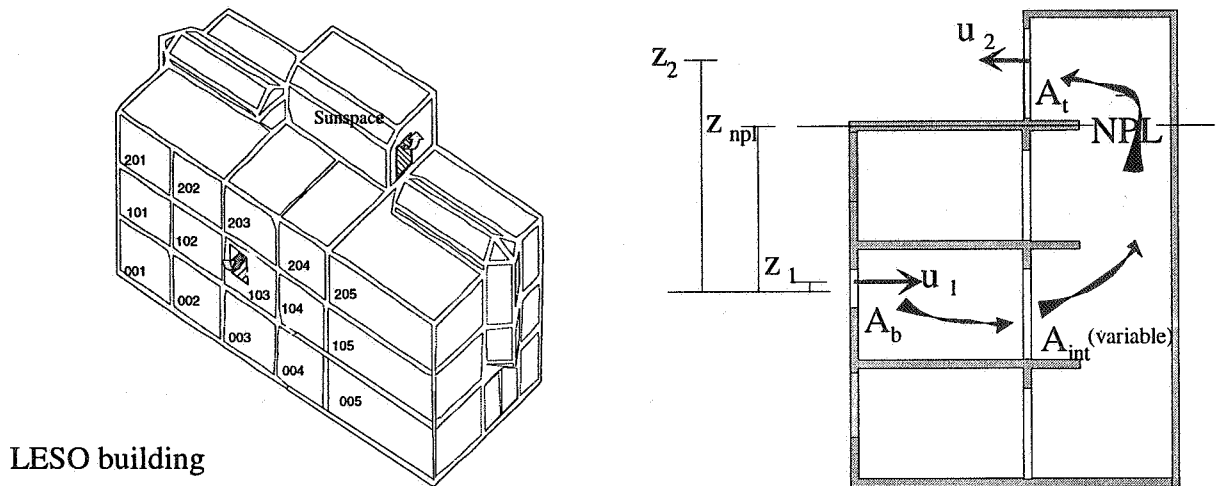


Figure 7: Openings used for the experiments of the section 3.3

The effective surface area A_{eff} could be normalised to A_b as $A_{eff} = \epsilon_b \phi_{eff} A_b$, where ϕ_{eff} is the new velocity coefficient after combining the two openings in series A_b and A_{int}

From equation (14) ϕ_{eff} could be derived as:

$$\frac{1}{\phi_{eff}^2} = \frac{1}{\phi_b^2} + \frac{1}{\phi_{int}^2} \left(\frac{\epsilon_b A_b}{\epsilon_{int} A_{int}} \right)^2 \quad (17)$$

The contraction coefficients have been evaluated by visualising the air path streams with cold smoke. For the outside opening A_b the distance between the jet and the opening edge was $3 \text{ cm} \pm 1$ (Figure 6) on the 4 sides of the perimeter. This gives a contraction coefficient of

0.85 ± 0.05 . For the intermediate opening, A_{int} , the ε_{int} is higher ($\varepsilon_{int}=0.95$) because only two edges are sharp (the sides near the floor and near the wall do not modify the air stream lines).

The results in Table 2 show good agreement between the measured velocity coefficients and those estimated using equation (17). The velocity coefficients for the top opening are the same as those of the first set of experiments.

Experiment		g	h	i	j
A_t (top opening surface area)	m ²	1.8	1.8	1.8	1.8
A_b (bottom opening surface area)	m ²	0.7	0.7	0.7	1.4
A_{int} (intermediate opening surface)	m ²	1.9	0.9	0.72	1.9
u_2 (measured)	m/s	0.72	0.64	0.64	1.00
u_1 (measured)	m/s	1.32	1.10	1.01	0.98
Φ_2 (top velocity coefficient)	-	0.74	0.64	0.62	0.69
Φ_1 (measured bott. Velocity coef.)	-	0.66	0.54	0.50	0.62
Φ_{eff} (calculated from eq. (17))	-	0.66	0.57	0.53	0.58
% difference		-1%	-6%	-5%	6%

Table 2. Measured velocity coefficient compared to the calculated one from equation (17)

3.4 Tracer gas measurements

The experimental set up is the same as in section 3.1 (Figure 4) with a bottom opening of 1.4 m² and a top opening of 1.35 m². In-out ΔT is 3.5°C with no detectable outside wind. In the intermediate door between the office 003 and the staircase a constant flow rate of 4.17 ml/s of SF6 gas is mixed naturally with the incoming air. The concentration of SF6 measured at the top opening gives the air flow rate through the building and therefore the discharge coefficient C_d . The tracer gas apparatus CESAR and tracer gas measuring techniques are explained in ref. [20].

Point velocity measurements at different heights of the openings give the velocity coefficient ϕ . Multiplied with the contraction coefficient ε calculated in section 3.2 they give a second discharge coefficient. Figure 1 shows good agreement of the discharge coefficients evaluated by the two methods. The generally accepted value of $C_d = 0.6 \pm 0.1$ is confirmed once again.

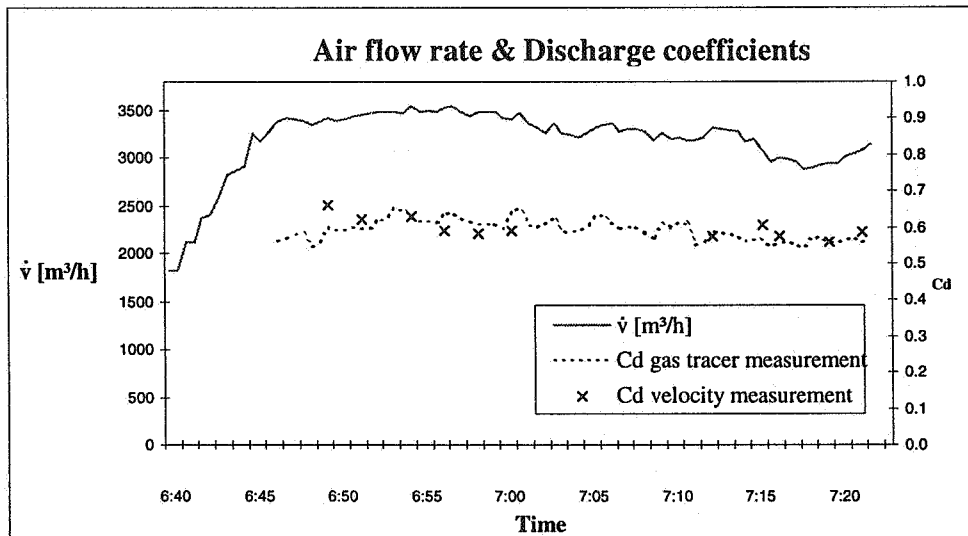


Figure 8. Comparison of the discharge coefficient measured by tracer gas and by velocity measurements.

4. DESIGN GUIDELINES

The NPL is a very important concept for the correct design of natural ventilation. When purely stack driven ventilation is used to cool high buildings, it is impossible to pass fresh air through the openings above the NPL. Opening a window in a high building implies a modification of NPL location and the air path in the whole building can be radically changed with an inversion of the flow direction in some openings.

The top opening should be as large and as high as possible. It should be placed in the low pressure facade of the building so that wind and stack pressure work in the same direction.

If the stack pressure at highest storeys is low, special arrangements may correct the situation (Figure 9). If these arrangements are impossible, mechanical ventilation could be considered for these storeys.

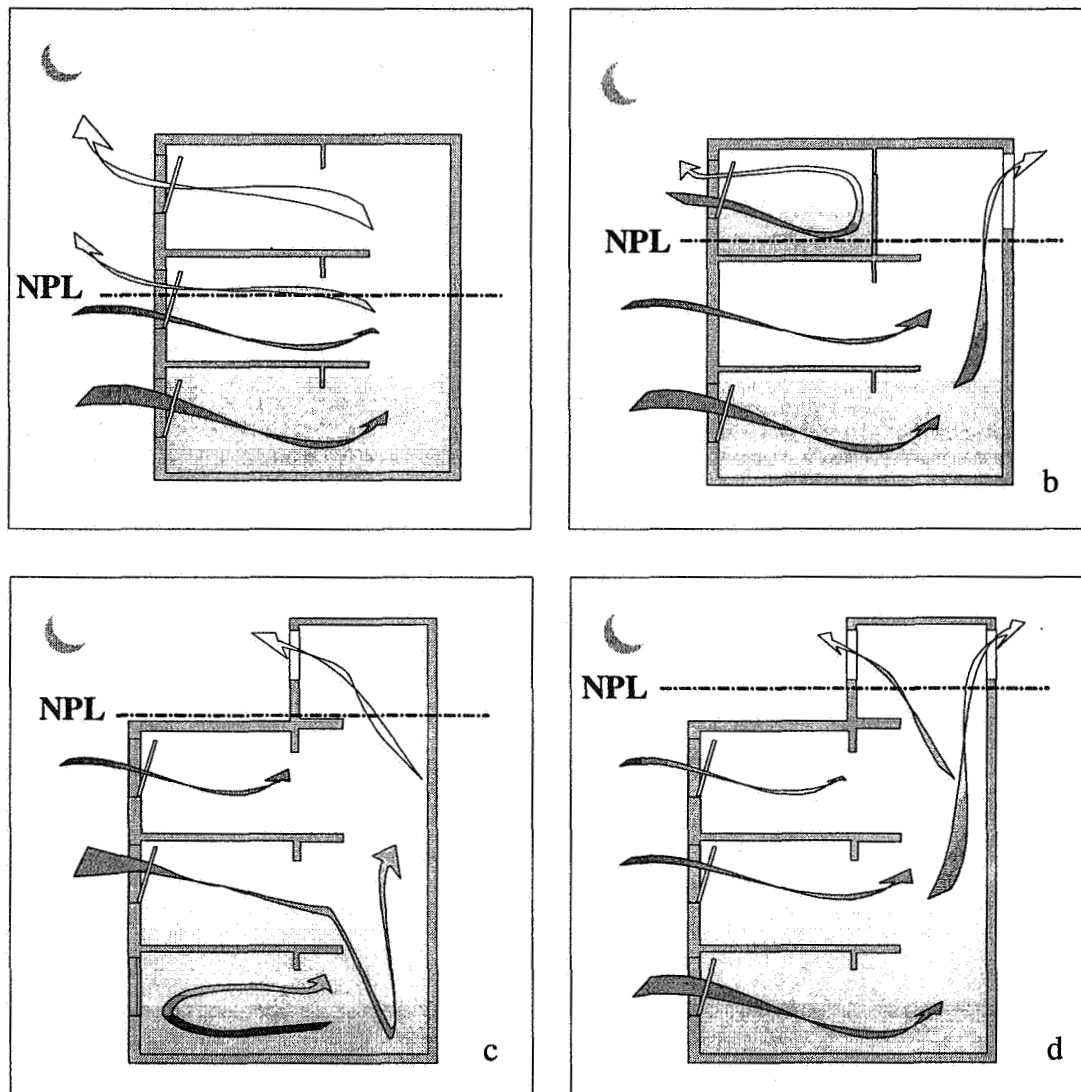


Figure 9. The four drawings show how the opening dimension and position can be manipulated to obtain night cooling of all the storeys of a three storey building.

a): poor cooling of upper level. b) to d): various ways to improve cooling.

The NPL also plays a very important role for the air tightness of the building. Cracks far from the NPL loose more than cracks near to it (see the Δp in Figure 2).

5. CONCLUSIONS

Purely stack driven ventilation can efficiently be used in buildings with more than one storey for night cooling natural ventilation. The outlet opening should be at the highest point of the building and it should have a large opening area. The outlet opening surface area on the top should be calculated so that the NPL be above the highest level to be cooled.

The experiment performed on site in a real building confirm laboratory experiments found in the literature. The air velocity and the tracer gas air flow rate measurements conducted to determine the discharge coefficient give the same results with the same uncertainty. The velocity coefficients $\phi = 0.7 \pm 0.1$ and jet contraction coefficients $\varepsilon = 0.85 \pm 0.1$ found in the experiments are in agreement with the generally accepted value of the discharge coefficient $C_d = \phi \varepsilon = 0.6 \pm 0.1$.

The nature of the phenomenon of natural ventilation makes the validation of models difficult (impossible?) whether they are simple or complex [4, 5, 7]. The precision in the experiments of this paper is of the order of 20% to 25%.

Depending on the problem to be solved, simple models for natural ventilation can give more valuable information to the designer. The imprecision of the model might be a minor issue if the model gives a qualitatively correct answer, especially when account is taken of the uncertainty of input data [4]. Future research should explore the possibilities of using qualitative modelling and qualitative simulation techniques to model natural ventilation.

6. ACKNOWLEDGEMENTS

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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**MODELLING THE INFLUENCE OF OUTDOOR POLLUTANTS ON THE INDOOR
AIR QUALITY IN BUILDINGS WITH AIRFLOW RATE CONTROL**

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MODELLING THE INFLUENCE OF OUTDOOR POLLUTANTS ON THE INDOOR AIR QUALITY IN BUILDINGS WITH AIRFLOW RATE CONTROL

SYNOPSIS

Low emitting building materials have contributed to the reduction of indoor air contaminants, and in many countries gas ranges and gas cookers are rarely used. As a result, in buildings located in urban environments, a considerable part of the contaminants in the indoor air may originate from the outdoor air.

In urban areas buildings are exposed to high concentrations of a large number of contaminants, especially during traffic peak hours. Often, the air intakes are located on the facades, and if there are no filters, the supply air will have the same content of contaminants as the outdoor air adjacent to the buildings. Where if possible, it is advantageous to place the air intakes on the roofs of the buildings (1). The higher the air intakes are placed, the better is the air quality in most cases.

To avoid exposure to high concentrations indoors due to concentration peaks of contaminants in the outdoor air, it is possible to decrease the airflow rate temporarily (2). This could be done by monitoring the concentration of carbon monoxide (CO) adjacent to the air intake. When the outdoor concentration of CO exceeds a preset limit, the fans are simply switched off, or the speed of the fans are decreased to a more suitable level.

LIST OF SYMBOLS

$\dot{V}_{source}$	internal source strength	cm ³ /s
$\dot{V}_{sink}$	internal sink effect	cm ³ /s
$\dot{V}$	airflow rate	m ³ /s
V	volume	m ³
Δt	timestep	s
t	time	s
C_o	outdoor concentration	ppm
C_I^n	indoor concentration at current timestep	ppm
C_I^{n+1}	indoor concentration at next timestep	ppm
C_S	supply air concentration	ppm
C_E	exhaust air concentration	ppm

1 INTRODUCTION

The discussion about the VOC emission from building materials has led to more low emitting materials being used. For many buildings in urban environments this means that the main part of contaminants in the indoor air originates from the outdoor air. However, this is not always

true, for example, office buildings have many such sources indoors, like printers, copy-machines, etc.

During traffic peak hours the outdoor concentrations of contaminants are especially high. If the outdoor air flow is reduced at these times, the result will be lower concentrations of contaminants in the indoor air. This can be achieved by monitoring the concentration of CO in the outdoor air adjacent to the air intake and defining at what concentration level the air flow should be reduced.

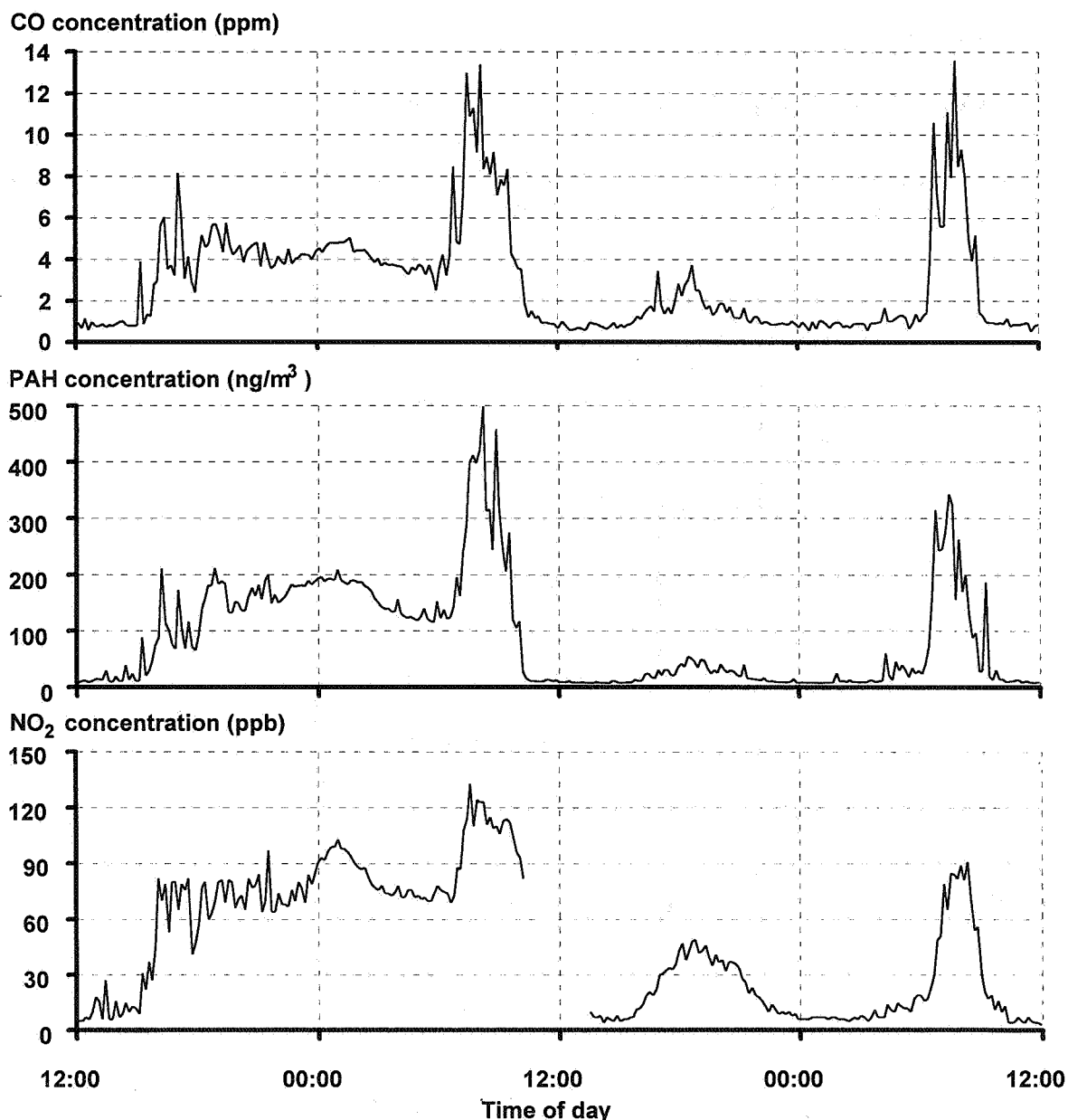


Figure 1 Outdoor concentrations for CO, PAH, and NO₂ over a period of 48 hours measured outside a building specially designed for people with allergy problems in a suburb of Stockholm.

CO is a suitable gas for indicating the amount of traffic related contaminants, and unless the occupants smoke, there are usually no sources indoors of CO. Figure 1 shows an example when the concentrations of carbon monoxide (CO), particulate polycyclic aromatic hydrocarbons (PAH), and nitrogen dioxide (NO₂) in the outdoor air were measured at the same time with three different measuring instruments. The studied compounds show a variation of similar pattern which is often the case for traffic related contaminants in urban environments (3).

2 METHODS

2.1 A model for calculation of indoor concentrations

With knowledge of the concentration of contaminants in the outdoor air and the air change rate in the building, the concentration indoors can be calculated (4-7).

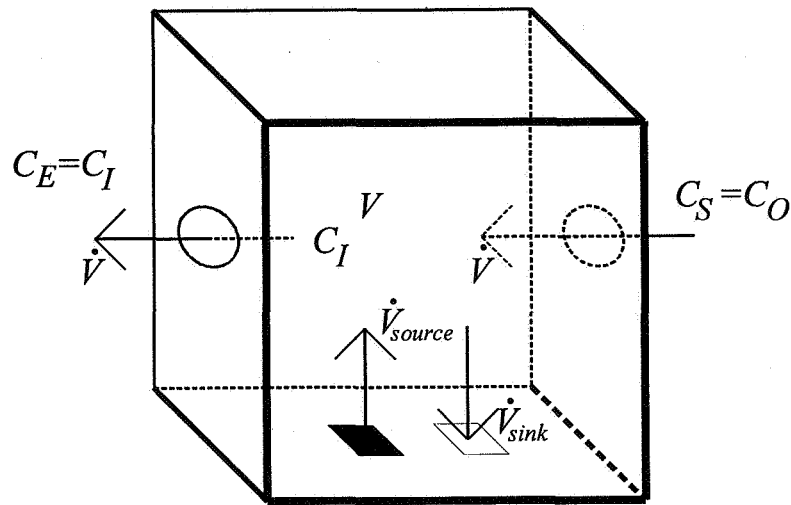


Figure 2 A ventilated room where the air is assumed to be well mixed.

Using the symbols in Figure 2, a balance equation can be established.

$$\dot{V} \cdot C_O + \dot{V}_{source} = \dot{V} \cdot C_I + \dot{V}_{sink} + V \cdot \frac{dC_I}{dt} \quad (1)$$

Equation (1) is valid under the assumption that the air is well mixed. In practice this is not completely true, but the deviation is often rather small. Equation (2) is the analytical solution to equation (1) for constant outdoor concentration and constant airflow, source strength, and sink effect.

$$C_I(t) = C_O + \frac{\dot{V}_{source} - \dot{V}_{sink}}{\dot{V}} + \left(C_I(0) - \frac{\dot{V}_{source} - \dot{V}_{sink}}{\dot{V}} - C_O \right) \cdot e^{-\frac{\dot{V}}{V} \cdot t} \quad (2)$$

where:

$$C_I(0) = C_I(t=0)$$

Equation (2) can be used to calculate the indoor concentration when the mentioned parameters are constant. The numerical solution, equation (3), obtained by the Euler method enables a variation of the parameters when calculating the indoor air concentration.

$$C_I^{n+1} = \frac{\dot{V}_{source} - \dot{V}_{sink} + \dot{V} \cdot (C_O - C_I^n)}{V} \cdot \Delta t + C_I^n \quad (3)$$

The measurements, presented in figure 3, were carried out with continuous monitoring by a Brüel & Kjær 1302 instrument. The measurements were made inside and outside a residential building in a suburb of Gothenburg. An air change rate of 0.5 h^{-1} was measured in the apartment studied.

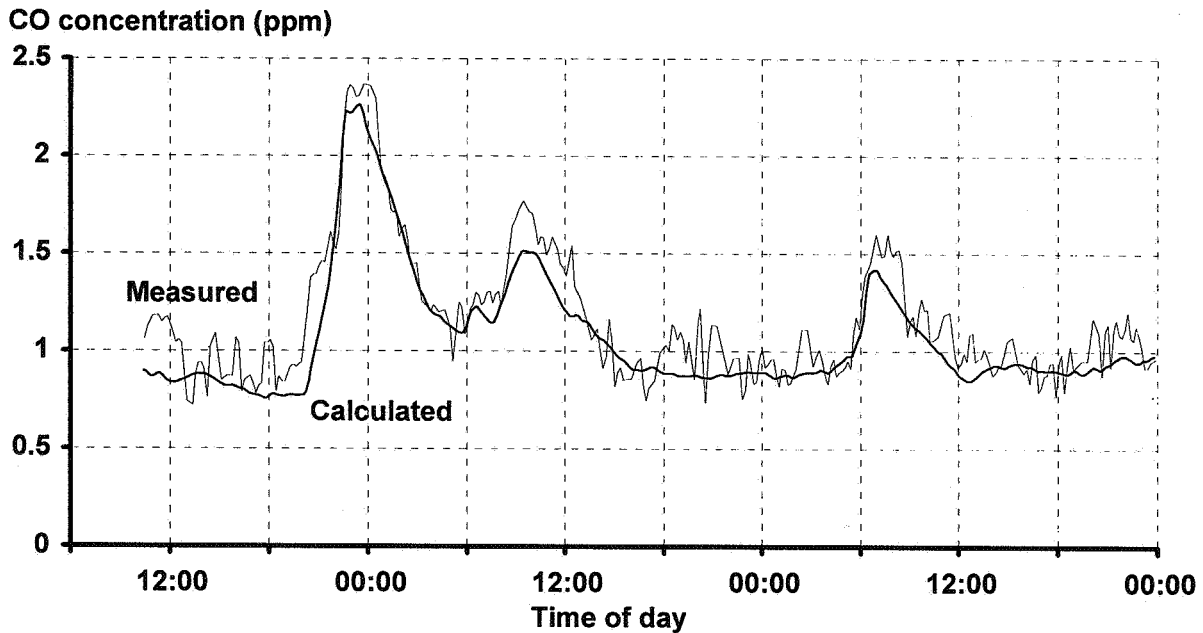


Figure 3 Concentration of carbon monoxide indoors in a residential building, measured values and values calculated with equation (3).

The measurement data on the outdoor concentration of CO was used in equation (3) to achieve the calculated indoor concentration in figure 3. The correlation coefficient is 0.91 between the measured and the calculated concentrations. The accuracy of the calculated

concentration is influenced by the length of the timestep and the accuracy of the measured air change rate used as input data to equation (3).

2.2 Dynamic modelling at variable air change rate

The air change rate in, for instance, residential buildings is typically 0.5 h^{-1} , whereas in office buildings the air change rate is usually 1.0 h^{-1} or higher. When the fans are turned off in a building with exhaust or balanced ventilation, the air change rate will not be reduced to zero, but more likely to about 0.1 h^{-1} , which is used in the example below.

One type of modelling is done by reducing the air change rate from 0.5 h^{-1} or 1.0 h^{-1} to 0.1 h^{-1} in equation (3) when the outdoor concentration of CO exceeds a specified limit. To do this, only monitoring of the outdoor concentration is needed.

Another type of modelling requires that both the outdoor and the indoor concentration of CO are monitored and compared. This helps to decrease the indoor concentration faster after a peak in the outdoor concentration. The indoor concentration can be higher than the outdoor concentration even if the outdoor concentration exceeds the specified limit, and in such cases the fanspeed should not be reduced. This happens when the outdoor concentration has been exceeding the limit substantially for a long time and then decreases rapidly.

3 RESULTS

To illustrate the effect of airflow rate control, measurement data for the CO concentration in the outdoor air during 24 hours is used, see figure 4. The average concentration of CO in the outdoor air during these hours was 1.68 ppm. The air samples of the outdoor air were taken on the balcony of an apartment in a building specially designed for people with allergy problems. The building is situated in a suburb of Stockholm.

In figure 4 the calculated indoor CO concentration is also shown both for constant and controlled air change rates. The calculated average concentration of CO for the indoor air indicates that the calculated concentrations in figure 4 are correct, when the air change rate is constant. The calculated average concentration indoors during the 24 hours in question is 1.65 ppm, with an air change rate of 0.5 h^{-1} . When the air change rate is kept constant at 1.0 h^{-1} , the calculated concentration indoors is 1.69 ppm. In reality, the average indoor concentrations should be the same as the outdoor concentration, but the calculated values show good agreement.

In table 1, for an air change rate of 0.5 h^{-1} , and in table 2, for an air change rate of 1.0 h^{-1} , the average CO concentration in the indoor air can be seen with different specified outdoor concentration limits, and single and double sensor monitoring.

The maximum increase of carbon dioxide (CO_2) during the periods with decreased air change rate is also given in tables 1 and 2. CO_2 is often used as an indicator of indoor air quality (8)

and is used here as an example of a contaminant emitted from a source indoors. During periods with reduced air change rate, the amount of contaminants entering with the supply air will decrease, but at the same time the concentrations of contaminants originating from sources indoors will increase. The source strength of CO₂ is set to 18 l/h, which represents a person at sedentary activity, and the space is given a volume of 36 m³. Assuming an outdoor concentration of CO₂ at 400 ppm this gives an equilibrium concentration of 1400 ppm when the air change rate is 0.5 h⁻¹, and for an air change rate of 1.0 h⁻¹ the equilibrium concentration is 900 ppm.

Table 1 The average CO concentration and the maximum increase in CO₂ concentration over 24 hours with an 18 l/h CO₂ source strength and a volume of 36 m³. The different concentration limits determine when the air change rate is switched from 0.5 h⁻¹ to 0.1 h⁻¹ and vice versa.

CO concentration limit (ppm)	Average CO concentration (ppm)	Decrease in CO concentration	Maximum increase in CO ₂ concentration (ppm)
1.0	1.10	33 %	1732
1.5	1.15	30 %	1028
2.0	1.19	28 %	850
2.5	1.23	25 %	728
3.0	1.25	24 %	727
1.0*	1.07	35 %	1354
1.5*	1.14	31 %	997
2.0*	1.19	28 %	850

* double sensor monitoring, i.e. monitoring of both indoor and outdoor concentration

Table 2 The average CO concentration and the maximum increase in CO₂ concentration over 24 hours with a 18 l/h CO₂ source strength and a volume of 36 m³. The different concentration limits determine when the air change rate is switched from 1.0 h⁻¹ to 0.1 h⁻¹ and vice versa.

CO concentration limit (ppm)	Average CO concentration (ppm)	Decrease in CO concentration	Maximum increase in CO ₂ concentration (ppm)
1.0	1.04	38 %	1798
1.5	1.12	34 %	1141
2.0	1.17	31 %	939
2.5	1.22	28 %	790
3.0	1.24	27 %	790
1.0*	1.01	40 %	1395
1.5*	1.12	34 %	1096
2.0*	1.17	31 %	939

* double sensor monitoring, i.e. monitoring of both indoor and outdoor concentration

The higher the maximum increase in CO₂ concentration, the lower the CO concentration limit is, since the timeperiod with reduced air change rate will be longer. The increase in CO₂ may, in this case, be a minor problem when the concentration peaks are one to two hours long.

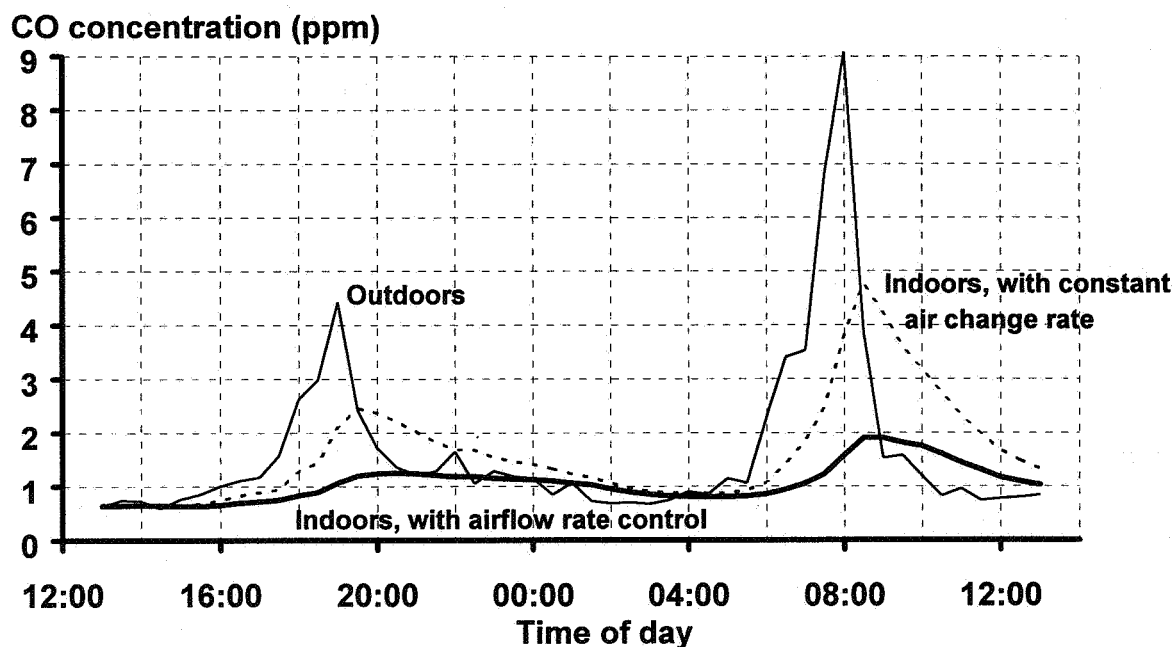


Figure 4 The carbon monoxide concentrations over 24 hours. The outdoor concentration was measured and the indoor concentration was calculated with equation (3) using an air change rate of 0.5 h⁻¹. The indoor concentration with airflow rate control, the thick line, was obtained with a CO limit of 1 ppm and double sensor monitoring.

4 DISCUSSION AND CONCLUSIONS

A tendency can be seen regarding single or double sensor monitoring, i.e. the effect of double sensor monitoring is more evident when the concentration limit for CO in the outdoor air is set low. As can be seen from table 1, the average CO concentration will, in the case of 1 ppm limit, an air change rate of 0.5 h⁻¹, and double sensor monitoring be 35 % less than the average concentration with constant air change rate. In the case of the specified limit being 1 ppm, however, it seems unrealistic regarding the high concentration of CO₂, or concentrations of other contaminants originating from indoor sources.

When the air change rate is 0.5 h⁻¹ and the concentration limit is 1 ppm, the maximum CO₂ concentration is higher than 3000 ppm in the example studied. The concentration will be even higher if the time of high outdoor CO concentration is longer. The longest time above 1 ppm is approximately 4 hours in that example but, for instance, during inversion the CO concentration can be higher than 1 ppm for more than 24 hours.

For the air change rate 0.5 h^{-1} , the average CO_2 concentration over the 24 hours shown in figure 4 is 2010 ppm and in cases when the air change rate is 1.0 h^{-1} the average CO_2 concentration will be 1350 ppm, which might be more acceptable. Whether the contaminants from indoor sources or the contaminants originating from outdoors are most important from a health point of view may vary from case to case, the example presented should be regarded as a demonstration of the method of air quality controlled airflow reduction.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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COMPARISON OF DIFFERENT METHODS OF INCORPORATION OF STOCHASTIC FACTORS INTO DETERMINISTIC MODELS OF INDOOR AIR QUALITY

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SYNOPSIS

The paper will discuss problems connected with incorporation of stochastic factors into deterministic models of indoor air quality. Three different methods:

- quasi dynamic multi-zone modelling with generating of input data time series,
 - multi-zone modelling based on the theory of stochastic differential equations,
 - Monte-Carlo simulation with independent random generating of stochastic parameters,
- will be shortly presented. Described methods are compared on the base of computer simulation of CO₂ concentration in simple two compartment office. The comparison of simulation outcomes shows that the way of stochastic disturbances incorporation into indoor air quality models does not have an important influence on mean value of predicted concentration. At the same time the analysis of standard deviations indicates that the method of disturbances generation and its later incorporation have a great influence on probability distribution of estimated concentrations. The author would like to highlight that differences. The conclusions include also subjective opinion of the author on main advantages and disadvantages of each of proposed methods.

LIST OF SYMBOLS

- C** - contaminant concentration matrix,
C_i - concentration of contaminant in zone *i*,
F - the drift matrix; the terms of the matrix are the constant part of coefficients *K_i*,
G - the diffusion matrix,
K_i - coefficient characterising transport of contaminants between zones or mixing factors, last coefficient in each row represents such mechanisms as production or removal of the pollutant in a zone.
n - number of equations that have to be solved to achieve solution of SDE,
N - number of zones in the building,
t - time,
ξ - the Gaussian white noise variables matrix.

1. INTRODUCTION

The experiences from many measurement programmes indicate that proper explanation of Indoor Air Quality level and dynamics of its variability should include also a description of many disturbances of stochastic nature. There are number of such disturbances but the interactions between weather fluctuations and/or occupants activities and building characteristics (including ventilation and air conditioning systems) may be regarded as the most important examples of these random factors.

As neglecting of these phenomenon may lead to a great inaccuracy there is a need to develop new reliable indoor air quality prediction tools and calculation methods that allow to take randomness of pollutant migration process into account. In last few years several new methods have been proposed by different research centres all over the world. Although, all these methods are based on conventional deterministic multi-zone models of pollutant migration in buildings, they differ in methods of random factors incorporation into the model. The paper will concentrate on the following methods:

- generating of disturbances' time series and than multiple multi-zone simulation,
- description of disturbances as Gaussian distributions and application of SDE theory,
- multiple generation of set of input data and than application of Monte-Carlo simulation.

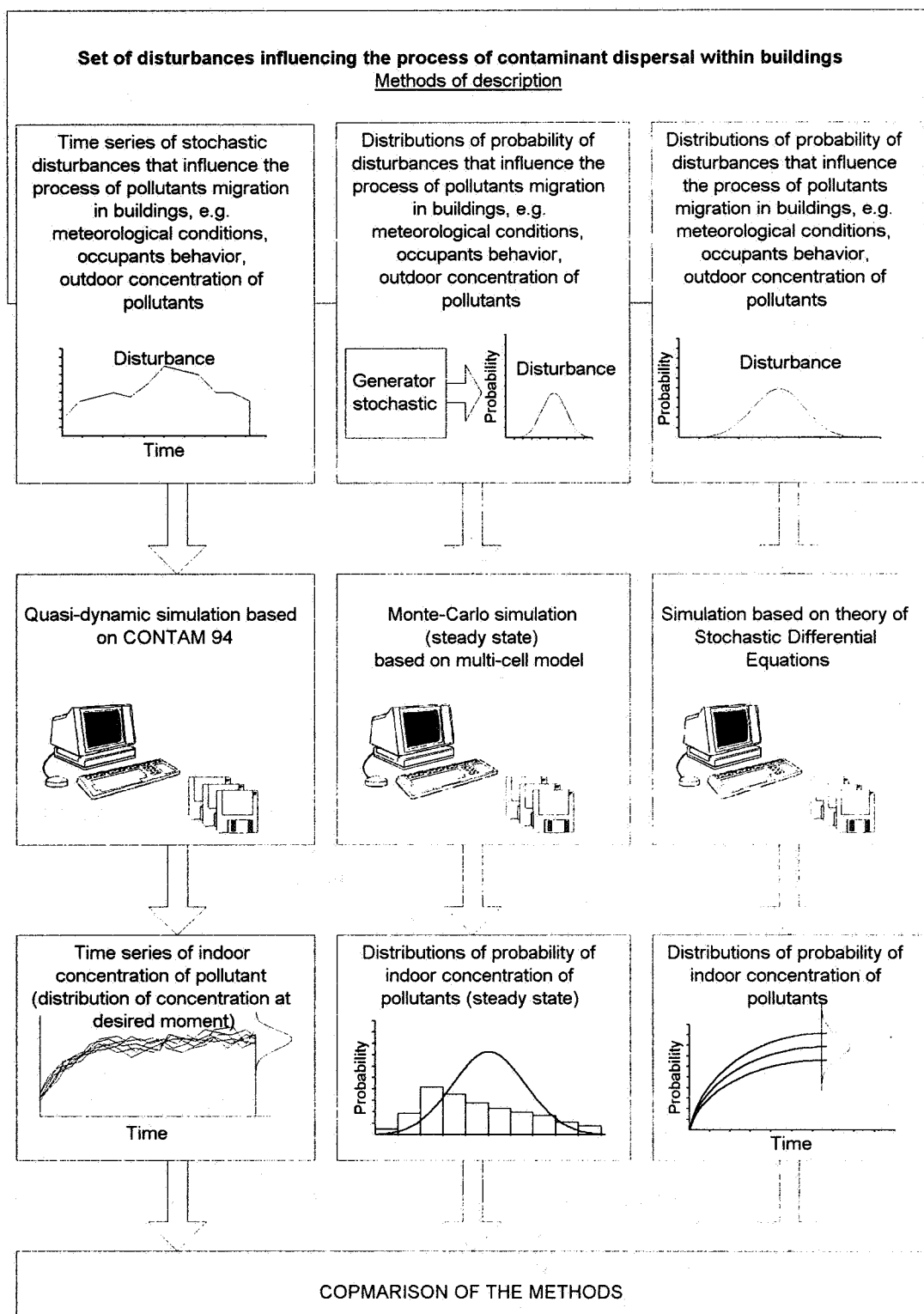


Figure 1. Comparison of different methods of incorporation of stochastic factors into deterministic models of indoor air quality. The scheme of applied methodology.

2. MULTI-ZONE DETERMINISTIC CONTAMINANT DISPERSAL MODELS

The most popular tool for contaminants' dispersal modelling in buildings are models based on deterministic approach. That type of models assumes perfect understanding of cause-to-effect chains in physical processes (different forms of energy and mass transport in buildings). Generally, the process of contamination dispersal is inseparably connected with two other main tasks: thermal analysis and air flow analysis. Often in practical applications thermal and air flow analysis are substituted by simple steady state models or specified fields of temperature and air flows [1]. Usually, for the purpose of modelling, selected building is described as a network, where nodes (called also zones) represent the volumes to which assumption on uniformity of parameters can be made (zones may represent groups of rooms, rooms, portions of rooms, shafts or portions of air handling systems) [7]. That simplification allows to write a set of ordinary differential equations based on contaminant mass balance,

$$\begin{bmatrix} \frac{dC_1}{dt} \\ \frac{dC_2}{dt} \\ \cdot \\ \cdot \\ \cdot \\ \frac{dC_N}{dt} \end{bmatrix} = \begin{bmatrix} K_1 C_1 & + & K_2 C_2 & + & \cdots & + & K_N C_N & + & K_{N+1} \\ K_{N+2} C_1 & + & K_{N+3} C_2 & + & \cdots & + & K_{2N+1} C_N & + & K_{2N+2} \\ \cdot & \cdot & \cdot & \cdot & \cdot & \cdot & \cdot & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot & \cdot & \cdot & \cdot & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot & \cdot & \cdot & \cdot & \cdot & \cdot \\ K_{N^2} C_1 & + & K_{N^2+1} C_2 & + & \cdots & + & K_{N^2+N-1} C_N & + & K_{N^2+N} \end{bmatrix} \quad (1)$$

Investigator is usually looking for discrete solution of formulated set of equations. Number of equations is equal to number of selected zones. Of course if simultaneous air flow or/and thermal analysis is performed number of equations will increase. A lot of computer programmes built on that type of models have been developed.

3. INCORPORATION OF STOCHASTIC FACTORS INTO POLLUTANTS MIGRATION MODELS

3.1. Generating of input data time series for quasi dynamic multi-zone models

The simplest way of incorporation stochastic factors into deterministic models is to substitute constant values of input data by real or simulated time series of the possible states of modelled parameter. The option that allow to prepare some input data as time series became a standard in new generation of contaminant dispersal software. That improvement refers not only to data directly connected with contaminants as:

- emissions and sinks of contaminants,
 - concentrations of contaminants in surroundings,
- but also to data influencing the process of air flows like:
- meteorological factors (atmospheric pressure, temperature, humidity, velocity and direction of wind),
 - occupants behaviour (door and windows opening, switching ventilation systems).

Decreasing of time step brings performed simulation nearer to reality. From the other hand, very small time steps cause considerable increase of computing time.

3.2. Description of stochastic factors as Gaussian white noise for SDE technique

In the late eighties and early nineties scientists from Canada published several papers ([2], [3], [4], [5]) on application of Stochastic Differential Equations (SDE) to the modelling of contaminant dispersal in buildings. Assumption that the K_i factors (in equation 1) may have random nature as well as application of SDE technique cause that equation 1 may be rewritten as

$$\frac{dC}{dt} = F(C) + G(C)\xi \quad (2)$$

Now, solution of set of equations is more complex. Solving the SDE using either *Itô* or *Stratanovich* interpretation will result in a set of possible results. The specific solution $C_i(t)$ depends on assumed probability. Moreover, the number of equations to solve depends on second power of zones number

$$n = \frac{N(N+1)}{2} + N \quad (3)$$

3.3. Generating set of random factors for Monte-Carlo simulation

Another way to incorporate randomness of some processes into deterministic models is multiple generation of set of possible input data from a data space (known parameters of standardised distributions; M-C or known empirical distributions; M-C(de)) and than application of Monte-Carlo simulation. Of course Monte-Carlo simulation is performed on a base of deterministic multi-zone models (steady state or dynamic). One should note that the accuracy of Monte-Carlo method is dependent only on number of runs (after 60-80 improvement of the accuracy is very small) and does not depend on number of input parameters.

4. COMPARISON OF THE METHODS

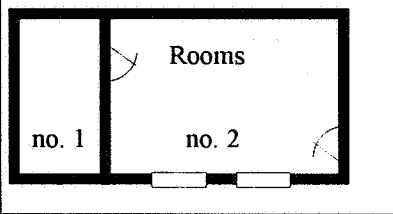
To present, how the selection of method results in differences in simulation outcomes, special computer test of CO₂ concentration in simple two compartment office was performed. The comparison was carried out assuming that the office (characteristics briefly presented in table 1) is ventilated in either natural or mechanical way. Simulation included the randomness of:

- weather fluctuations(conditions of Polish climate in winter),
- concentration of CO₂ in surroundings and
- occupants activities (entering and leaving the office, opening of door and windows, switching ventilation systems on and off).

Of course the set of disturbances influencing the indoor air quality in office was the same for all cases. Differences refers to method of its description and further incorporation into simulation model.

Results achieved by the quasi dynamic multi-zone modelling with generating of input data time series (10 simulations for different set of time series) using CONTAM94 [7] is presented at figure 2. Outcomes from simulation based on SDE is presented at figure 3. Figure 4 presents the example of steady state Monte-Carlo simulation. Figures 2,3,4 present the results from wider analysis of influence of stochastic disturbances on process of pollutant migration inside buildings carried out by Sowa [6].

Table 1. Characteristics of simple two compartment office.

	Room 1 additional room	Room 2 main office
Floor area	15 m ²	50 m ²
Volume	45 m ³	150 m ³
Number of occupants	variable from 0 to no. of occupants present in office	variable from 0 to 10
Openings	door to room 2 (+comp. grill)	entrance door, 2 windows, door to room 1 (+comp. grill)
Ventilation systems (options)		Pollutants
<u>mechanical ventilation</u> supply: r. 1, 450 m³/h (3 h⁻¹) r. 2, 0 m³/h exhaust: r. 1, 315 m³/h r. 2, 135 m³/h recirculation: outdoor air 300 m³/h return air 150 m³/h windows closed	<u>natural ventilation</u> in each room 1 shaft 0.14 x 0.14 m height 1 m estimated characteristics $\Delta P = 1 \text{ Pa}$ $W = 50 \text{ m}^3/\text{h}$ $\Delta P = 4 \text{ Pa}$ $W = 100 \text{ m}^3/\text{h}$ windows open randomly	<u>pollutant: CO₂</u> source: occupants intensity of emission: 20 l/h per person, initial concentration: 780 mg/m³ (~340 ppm) CO₂ outdoor concentration mean 780 mg/m³ SD 20 mg/m³

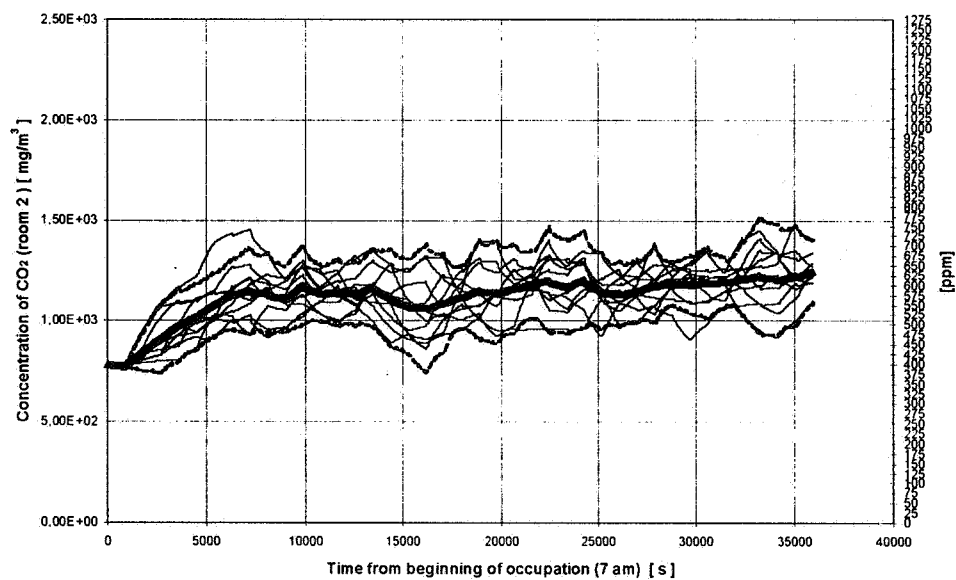


Figure 2. Concentrations of CO₂ in room 2 ventilated mechanically (thin lines - 10 independent runs, thick line - mean value, thick dashed lines - mean value with two standard deviation margin), [6]. Simulation based on generating of disturbances' time series and than multiple simulation using quasi dynamic multi-zone model of air pollution migration in buildings CONTAM94, [7].

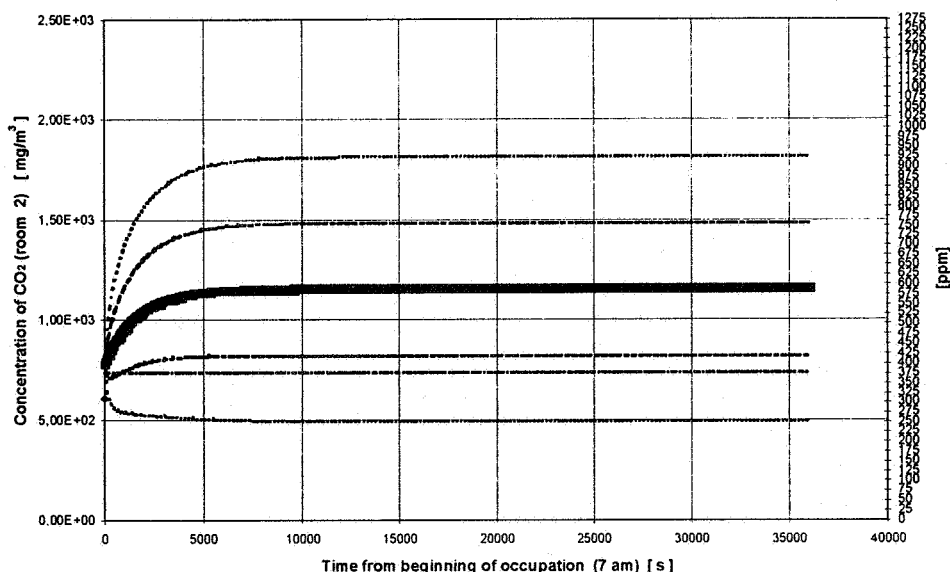


Figure 3. Concentrations of CO₂ in room 2 ventilated mechanically (thick line - expected value, thick dashed lines - expected value with one and two standard deviation margin), [6]. Simulation based on description of stochastic disturbances as Gaussian distributions and then application of stochastic differential equations (SDE) technique.

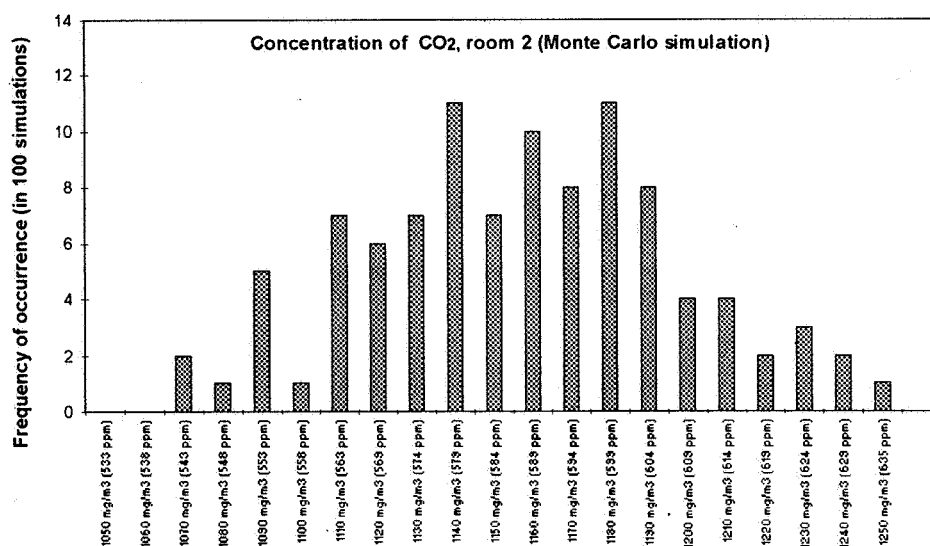


Figure 4. Frequency of CO₂ concentrations in room 2 (mechanical ventilation). Monte-Carlo simulation based on 100 independent generations, [6].

The mean values and standard deviations of simulated concentrations for described simple test office are presented at table 2. CONTAM denotes simulation based on generating of disturbances' time series and then multiple simulation using quasi dynamic multi-zone model of air pollution migration in buildings CONTAM94. Simulation based on description of stochastic disturbances as Gaussian distributions and then application of stochastic differential equations technique was signed as SDE. Monte-Carlo analysis was performed for generation of set of possible input data based on both known parameters of standardised distributions; M-

C, and known empirical distributions; M-C(de). Monte-Carlo simulations were based on 100 independent generations.

Table 2. The comparison of simulation results for proposed methods.

The method	CO ₂ concentration [mg/m ³]							
	Mechanical ventilation				Natural ventilation			
	Room 1		Room 2		Room 1		Room 2	
	Mean	SD	Mean	SD	Mean	SD	Mean	SD
CONTAM	1290.58	181.56	1146.37	95.65	1319.06	228.83	1156.07	147.73
SDE	1306.45	429.07	1154.97	329.70	1290.47	539.20	1133.04	573.00
M-C	1235.99	512.64	1077.28	432.24	1339.95	1257.51	1171.14	1180.49
M-C (de)	1296.03	64.46	1152.20	40.36	1302.60	114.98	1146.24	94.87

4. CONCLUSIONS

The comparison of simulation outcomes shows that the way of stochastic disturbances incorporation into contaminant migration deterministic models does not have an important influence on mean value of predicted concentration (table 2). At the same time the analysis of standard deviations, presented in table 2, indicates that the method of disturbances generation and its later incorporation into contaminant dispersal model results in great differences in distribution of estimated concentrations.

The highest values of standard deviation of concentration occurred in cases when generation of random input data is performed on a base of known parameters of standardised distributions (especially when disturbances are treated as independent factors). This methods seem to overestimate variability of contaminant concentrations. One should notice that using that methods, it is possible to reach unrealistic values of concentrations (e.g. negative concentrations) with quite high probabilities. Monte-Carlo method based on empirical distributions and quasi dynamic modelling based on description of disturbances as interrelated time series provide lower values of standard deviation.

All these differences are much higher for cases when natural ventilation is considered. It is the result of option that allow to open the windows. Windows opening very strongly and very rapidly influence the concentration of pollutants. Moreover in that case description of air volume as one of well known distributions is difficult and questionable.

Mean and SD of CO₂ in Room 1 which is ventilated indirectly, by air from room 2, are higher. That simple test indicates that outcomes of the multi-zone simulations are very sensitive to method of stochastic disturbances description. At the moment, it is quite difficult to definitely say that one of proposed method is simply the best (however it seems to be more safe to recommend Monte-Carlo method based on empirical distributions and quasi dynamic modelling based on description of disturbances as interrelated time series). All methods have their advantages and disadvantages (table 3).

One may expect that in near future that type of modelling will be more popular. Risk analysis calls for probability distributions of concentrations. Incredibly fast development of computer science and technology will probably overcome some of these disadvantages. So, it seem to be quite good time for discussion on the best methods to incorporate randomness of several processes into indoor air modelling.

Table 3. Main advantages and disadvantages of presented methods.

Method	Main advantages	Main disadvantages
Time series and quasi dynamic simulation	• correlation between different parameters may be taken into account, • existing software may be applied (contaminant migration simulators, weather generators, etc.)	• time consuming description of building, • high hardware requirements (simulation of real building with small time step), • demands for huge computer memory to store simulation results,
modelling based on SDE technique,	• modern approach, • results of analysis presented as probability distributions,	• lack of user friendly software, • time consuming calculations, • high hardware requirements, • unrealistic values of concentrations may be reached (e.g. negative concentrations) with quite high probabilities,
random data generating for M-C simulation.	• simplicity of the method, • existing software may be applied (contaminant migration simulators, etc.), • relatively short time of computing,	• treating input data as independent parameters, • unrealistic values of concentrations may be reached (e.g. negative concentrations) with quite high probabilities,

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

A NOVEL VENTILATION FAN WITH CAPABILITY OF REMOVING PARTICULATE POLLUTANTS FROM BUILDINGS

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(Full paper not available at time of print)

Optimum Ventilation and Air Flow Control in Buildings

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Title A NOVEL VENTILATION FAN WITH CAPABILITY OF REMOVING PARTICULATE POLLUTANTS FROM BUILDINGS
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Abstract

Experimental investigation has been carried out on a novel ventilation fan that fulfils the function of gas circulation and de-dusting of gas streams. It is very similar to a conventional, centrifugal ventilation fan except that it utilises a brush disk instead of the blade disk of the latter. The brush, which is thinly coated with water to trap particles and absorb gases, is made of corrosion resistant flexible fibre and is cleaned by a slow, continuous supply of water to its surfaces. As the fibres are in a state of constant oscillation, scale deposits are unable to adhere and the device is very resistant to fouling and blockage. The trapped dust flows out of the fan case as a dilute slurry. This fibre-impeller fan is a cheap, robust, efficient and fouling-resistant unit suitable for application in industrial, commercial and agricultural buildings. Laboratory tests of the performance characteristics of the fibre-impeller fan, in terms of efficiency and particle removal, have been carried out using a 0.4m diameter prototype fan with various fibre diameters and fibre numbers. The test facility and procedures were in accordance with BS848:1:1980. A plastic fibre-impeller disk was mounted in a standard centrifugal fan casing and driven via a variable speed motor. Water was used as the irrigating liquid and aerosol particles were introduced at the inlet of the fan using an aerosol generator. The results showed that the particle removal efficiency (defined as the ratio of the number of particles removed by the fan to the number of particles at fan inlet) increased with particle sizes. In addition, increasing the water flow rate also led to a significantly increase in particle removal efficiency. The particle removal efficiency of 96% was achieved for particles less than 2 μm in diameter, at 800 rpm and with a water flow rate of 4 ml/s. Increasing the fan speed also resulted in a significant increase in particle removal efficiency.

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

(Title)

Residential Mechanical Ventilation Systems: Performance Criteria and Evaluations

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Residential Mechanical Ventilation Systems: Performance Criteria and Evaluations

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Synopsis

The performance of mechanical ventilation systems has been checked in several innovative residential houses in the frame of Swiss research as well as pilot and demonstration projects. This paper gives a list of performance criteria for the ongoing comparison and evaluation of these mechanical ventilation systems. For some criteria, target values are proposed.

In the second part, this paper shortly describes four residential buildings with mechanical ventilation systems where such evaluations were performed and highlights interesting design features and results from the measurements. It then focuses on discrepancies between the design goals and what has been encountered in reality during the evaluation campaign. Problem items were, among others, commissioning, occupant acceptance and window opening behaviour as well as sealing of ducts and heat exchangers.

1 Introduction

At present, the benefits and the market chances of mechanical residential ventilation systems are again passionately discussed in Switzerland. The performance of mechanical ventilation systems has been checked in several innovative residential houses in the frame of Swiss research as well as pilot and demonstration (P+D) programmes. Extended commissioning was performed, energy figures were determined, air flows and the air distribution patterns as well as noise levels were measured and inhabitants were interviewed. The results are presently analysed and documented.

In order to have a better basis for the overall discussion of the evaluation and comparison of the systems, the working group on residential mechanical ventilation systems of the VSHL (Swiss association of HVAC manufacturers) has produced among other documents a working paper on performance evaluation criteria and parameters for residential mechanical ventilation systems with heat recovery and is presently establishing recommended target values for a selected number of these parameter [1]. A summary list of these parameters and the respective target values is given in the following section 2. The list will be reviewed considering the outcome of IEA-ECB Annex 27 [2].

In section 4, four research and P+D houses, three apartment houses and one single family house, are shortly described. All these buildings are very airtight and equipped with balanced ventilation systems with heat recovery and with earth coupling for the air intake. Interesting design features of the mechanical ventilation systems and specific results from the evaluations are highlighted. Experiences gained concerning the earth coupling are summarized in section 3.

2 Performance criteria and target values

Performance criteria to be applied to the demonstration buildings mentioned above are listed below. It is obvious that some of these criteria need further explanations for a proper application. In addition it has to be noted that both criteria and target values have to be defined separately for different building and systems categories.

2.1 Building and systems categories

- Single family house, multifamily house
- New building/retrofit
- Type of heat recovery (air/air plate , a/a heat pump, air/water heat pump)
- Ventilation only / additional warm air heating or hot water production
- Research or pilot & demonstration building/ other

2.2 Performance evaluation criteria and parameters

- Costs**
- Investment costs
 - Operating costs
 - Life cycle costs

Effects on construction and space requirements

- Room needed for system and ducting
- Additional elements such as earth coupling systems or outside air terminals

Power ratings and energies

The general criteria are rational use of energy and use of renewable energy. Specific emphasis is given to the use of electricity and the amount of electrical energy used to gain or recover heat. Energy flows are analysed on the basis of energy flow diagrams, as given in Figure 11. These diagrams are helpful to understand especially more complex systems with heating and domestic hot water supply functions. A second class of criteria applies to the individual components.

System criteria:

- Proportion of renewable energy used
 - Proportion of heat recovery compared to ventilation losses
 - Seasonal or yearly electrical energy consumption of the system
 - 1) Total
 - 2) Specific per floor area
 - Factor heat gain/electric input
 - 1) As a performance related figure for design conditions
 - 2) As an energy related figure for the heating/operation period
- Values ranging from 5 up to 11 have been measured in the evaluation cases.
A target value of 7 is proposed.
Critical for the application of this criteria is the definition of the heating period. Any operation of the system in the transient saison jeopardizes of course this value.

Component criteria:

- Proportion of embodied energy contained in the components and its pay-back time
- Specific electric power consumption of fans. A target value of 0.35 W per m³/h of supplied air is proposed.
- Specific heat exchanger efficiency (related to flow resistance).

Air quality

Air quality is primarily evaluated on the basis of delivered outdoor air flow rates and the reliability for the supply. Extensive auditing as e.g. in the European IAQ-Audit project was generally not performed. Also indoor air pollution sources are not characterized in detail. Two to three system operation modes are generally available with different air flow rates: Basic supply ($n_L \approx 0.3$), normal supply ($n_L \approx 0.5$), enhanced supply ($n_L > 0.5$). Additional criteria considered are:

- Handling of extract air in the kitchen (recirculating system, separate exhaust, ducting through the heat recovery system).
- Proportion of recirculation from the heat exchanger.
- Hygienic performance of earth coupling systems.

This aspects is investigated further in a specific research project conducted at the ETH-IHA in Zurich [3]. Microbiological long term and interval measurements are made in 13 earth coupling systems. Results are expected by the end of this year.

Ventilation efficiency

Ventilation efficiencies are not measured in all cases, but room air distribution patterns and related efficiency values have been measured in the Hausäcker house using tracer gas techniques. For smaller room sizes, air change efficiencies tend to be around 0.5 (fully mixed condition), regardless where inlet and outlet positions are. For larger room sizes and in case of additional warm air heating, efficiency values may vary more.

Thermal comfort

All houses are very well insulated. Therefore, the main concern for thermal comfort is draft. Thus, parameters to be considered are supply air temperature and velocities.

Acoustics and Noise

From the inhabitants perspective, noise produced by the ventilation system is one of the major sources for complaints. This has to be seen under the fact that in Switzerland most dwellings are naturally ventilated and thus any sound from a ventilation system may be judged as annoying. From the evaluations performed so far, it can be concluded that sound generation levels must be evaluated mainly against the background noise and not according to fixed values. Nevertheless, sound levels according to the Swiss standards for system generated noise (30 to 35 dBA) are too high. Acceptable system designs must have significantly lower levels (20 to 25 dBA)

Control, operation, maintenance and reliability

Emphasis is put on criteria like:

- Possibilities for occupants to control flow rates and temperature settings
- Accessibility for cleaning
- Documentation and training of people which are in charge of maintenance

Occupant's perception, acceptability and influence

The occupant's perception of the air quality and the acceptability levels have been determined in many cases using specifically set up questionnaires, prepared by the ETH [4]. A wide spread of different behaviours have been observed, from good acceptance to a total objection including taping of air in- and outlets. Large effects on the system performance have been observed mainly from window opening, because this changes pressures and reduces the amount of available heat for recovery (see Fig. 7).

Design, manufacturing, installation and commissioning of ventilation system

- Manufacturing of components (e.g. leakage of plate heat exchanger)
- Selection and installation of components
- System balancing and commissioning

3 Earth coupling

A respectable amount of buildings are realized in Switzerland with earth coupling for the air intake. This systems normally show factors heat gain/ electric input in the range of 40 to 100. During summer, this system may also provide cooling. Experiences with earth coupling systems can be summarized as follows [5]:

Conditions for profitability:

- Air flow rates according to hygienic requirements
- Requirement for preheating
- No or little additional excavation for the installation of the pipes necessary

Recommendations for the design:

- Airtight, smooth pipes must be used (prevention of radon ingress, low flow resistance)
- Accessibility for inspection and cleaning must be provided
- A bypass is necessary for optimum operation
- Pipe length of approx. 20m is sufficient, air velocity in the pipe $< 3\text{m/sec}$

4 Evaluation cases

4.1 Apartment houses 'Hausäcker'

Building description

The two residential buildings 'Hausäcker' are located in a town 20 km north of Zurich, each three-storey house has 12 apartments with a heated floor area of approximate 1400m². The heating energy consumption per annum and heated floor area is 200 MJ/m²-a [6]. Both houses use district heat which is delivered in the rooms by floor radiators.

Ventilation is provided by a central balanced system, located in the basement of the building. The intake air is preheated in an earth coupling system and then led through a cross flow plate heat exchanger, resulting in an average supply air temperature of 19°C. No additional supply air heating unit is installed.

In the rooms, the supply air is delivered through specifically designed air terminals, placed in the lower part of the door case. The aged air passes through a gap under the door and an overflow terminal in the upper part of the door case and is extracted in the wet rooms and in the kitchen. Due to fire safety and hygiene requirements, the extract air from the kitchen is directly exhausted without passing the heat exchanger.

The performance evaluation

The 'Hausäcker' apartment houses were evaluated according to the following performance criteria:

- Detailed determination of temperatures and energy consumption for the whole heating season
- Heat recovery efficiency
- Ventilation efficiency in a typical room
- Inlet/outlet acoustics
- Interviewing of all occupants according a questionnaire developed by the ETH [3]

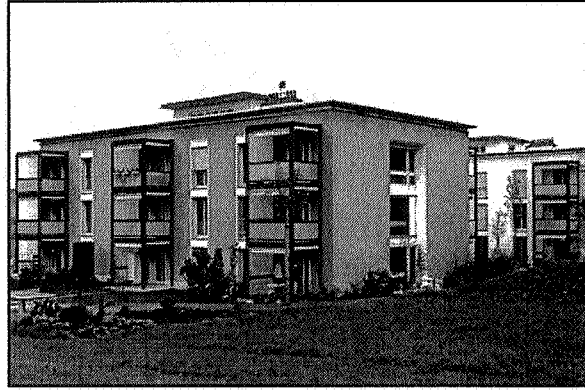


Figure 1: Apartment houses 'Hausäcker'

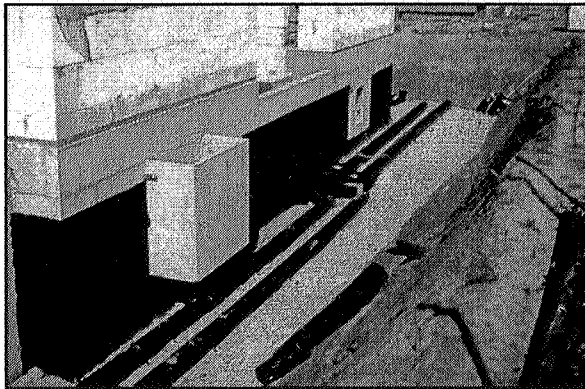


Figure 2: The earth coupling system. The pipes are placed in the regular excavation around the building.

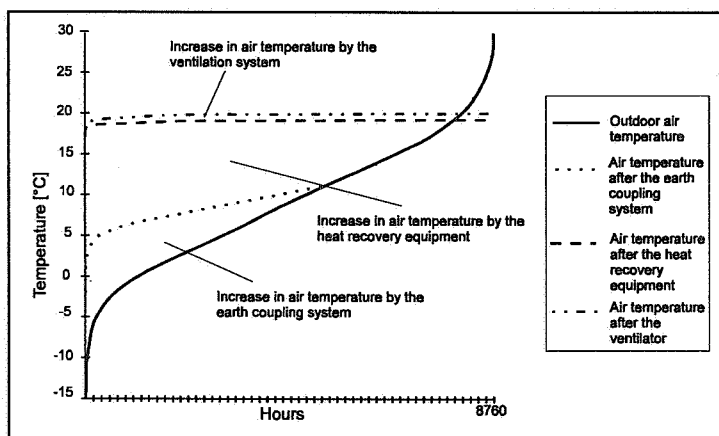


Figure 3: Temperature histograms for outdoor air, air after earth coupling, heat recovery and ventilators respectively.

Results

Visualisation of the air distribution patterns with a smoke generator and tracer gas measurements show a good air distribution in the individual rooms, even in the case of supply air temperature being higher than room air temperature. Although the gap under the door is located just besides the air outlet, no short-circuiting could be observed.

4.2 Housing development 'Hünenberg Eichrütli'

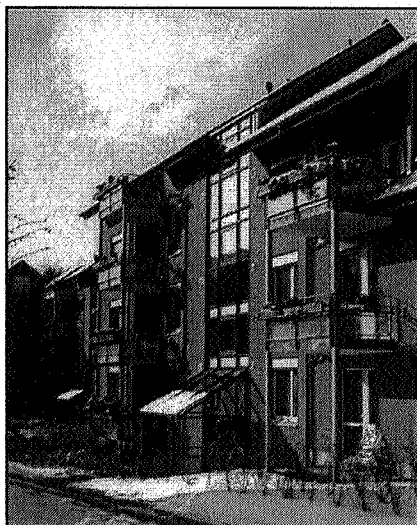


Figure 4: 'Hünenberg Eichrütli'

Building description

The housing development 'Hünenberg Eichrütli' consists of 6 four-storey multifamily houses with total 47 apartments. Three buildings are built together and have one central balanced ventilation system. The energy consumption for heating and domestic hot water is 205 MJ/m²-a.

In contradiction to the apartment houses 'Hausäcker', the earth coupling system in 'Hünenberg Eichrütli' is located under the garage. Also, heat is recovered by a thermal wheel and not by the more common plate heat exchangers used in the other three buildings described in this paper.

A further measure for energy consumption reduction is the air/water heat pump, which is used for the preheating of the domestic hot water.

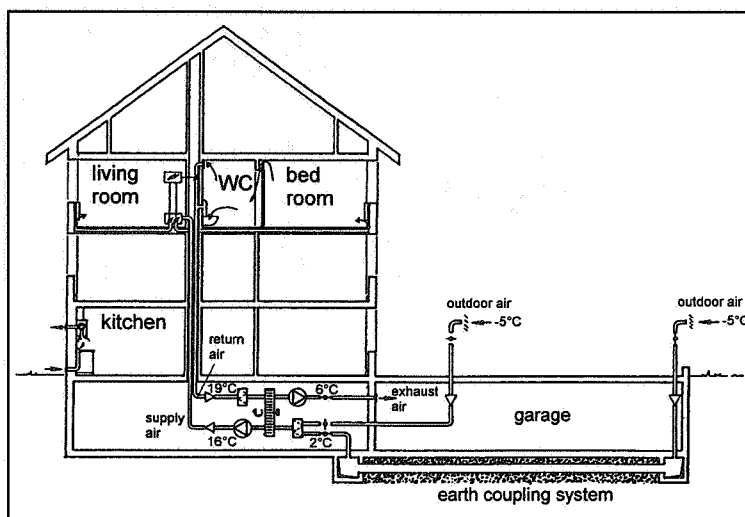


Figure 5: The mechanical ventilation system of 'Hünenberg Eichrütli'

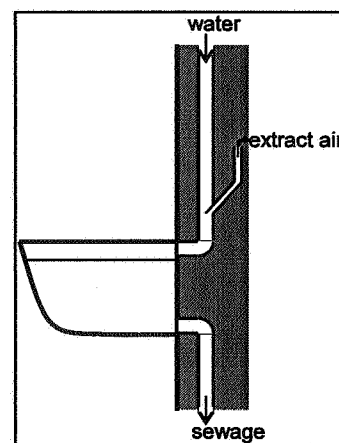


Figure 6: For better source control, the local exhaust system in the toilet is combined with the flushing pipe.

Results of the performance evaluation

Measurements with tracer gas revealed that the natural flow due to airing is much higher than expected (approximately half of the mechanical supply air flow rate, see Figure 7).

Neither could odour transmissions due to leakage in the thermal wheel be detected nor were there any complaints from the occupants in this respect.

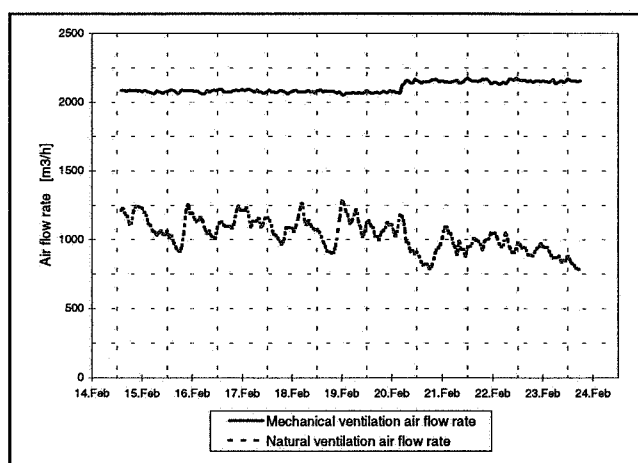


Figure 7: Total outdoor air flow rates in the three-building unit

4.3 Terraced housing development 'Riehen'

Building description

42 apartments and 4 single-family houses form the terraced housing development 'Riehen', located about 5 km north-east of Basel. With a energy consumption for heating and hot water of 200 MJ/m²-a, the development has low energy building qualities [7].

The air passes through the outdoor terminal to the two, 160m long concrete earth tubes and then to the heat recovery unit. Each apartment has its own balanced ventilation system with heat recovery, four of them grouped together in a system room. The four single-family houses have air heating combined with conventional heating, the apartments have just conventional heating. The air outlets are placed at the ceiling, the return air passes via door gaps to the extract air terminals in the wet rooms and in the kitchen. The kitchen hood has a recirculating air system with charcoal filter.

Performance evaluation program and results

In the performance evaluation program, estimations of the embodied energy contents of the ventilation systems have been done. The pay-back time of the embodied energy for the heat recovery equipment is approx. 1.5 years, that of the earth coupling system 9 years. (For comparison 'Hausäcker': Average of approx. 6 years).

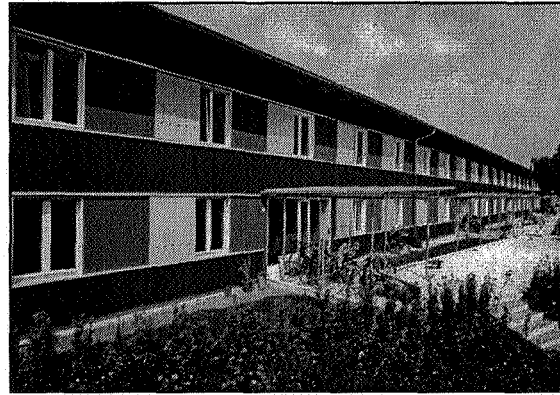


Figure 8: Terraced housing 'Riehen'

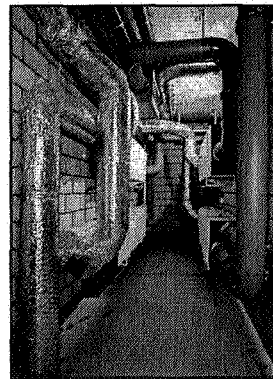


Figure 9: System room with ventilation units for 4 apartments

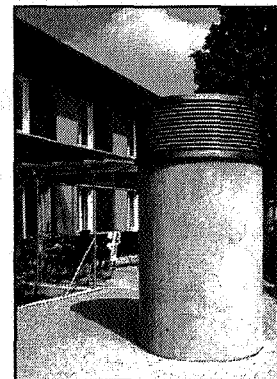


Figure 10: The outdoor air intake terminal

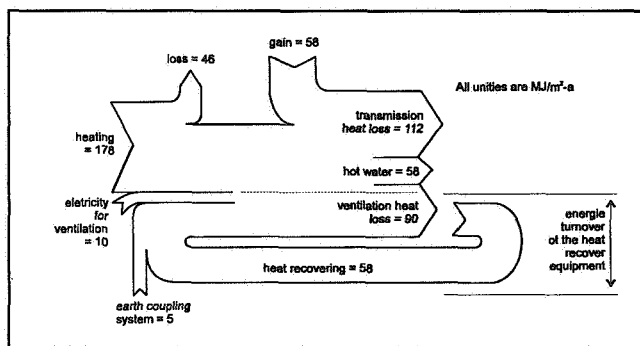


Figure 11: Energy flow diagram of an apartment

The energy flow diagram shows quite distinctively the rather marginal contribution of the earth coupling system. Nevertheless, the real value is higher. Due to the adverse influence of ground pre-heat and heat recovery, only the additional gain is shown.

Although the additional gain of the earth coupling system is rather small (see as well Figure 12), the earth coupling system has advantages: On one hand significant gains especially at days with low outdoor temperatures, on the other hand it prevents the heat recovery equipment from freezing, which would lead to an interruption of the ventilation system operation due to the defrosting cycle.

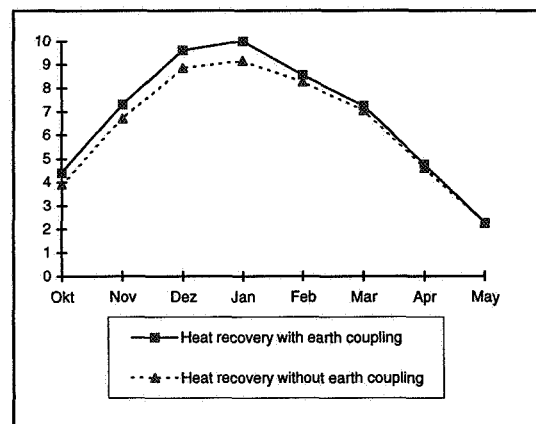


Figure 12: Ratio heat gain / electric input for the ventilation system with heat recovery

4.4 Zero-heating energy houses 'Wädenswil'

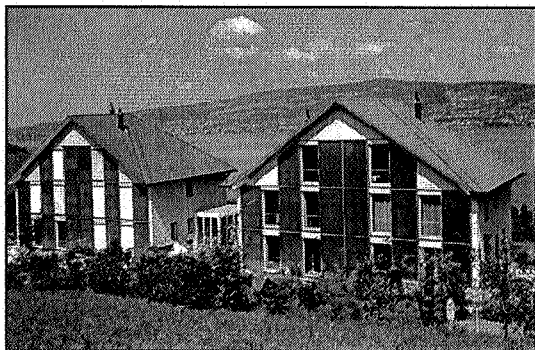


Figure 13: The south façade of the zero-heating houses, in black the transparent insulation of the solar collectors

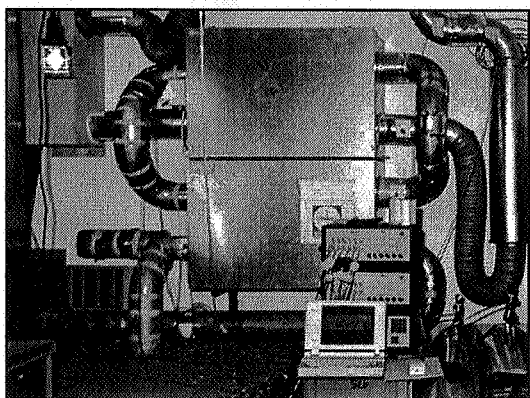


Figure 14: Tracergas measurements at the two cross flow heat exchangers for determination of leakages

Building description

The two zero-heating energy houses constitute together with three low-heating energy houses a housing complex for 10 families, located in Wädenswil, a village at the south border of the lake of Zurich. The three low-heating energy houses have conventional solar collectors of 9 m² with a water storage tank of 3 m³ for hot water. They are heated by a cogeneration plant. The two zero-heating energy houses have large solar collectors of 33 m², which are covered with transparent insulation material. The solar energy is used for the floor heating system and for hot water, a large water heating storage tank of 20 m³ facilitate a seasonal heating storage. Occasional heating with a wood oven in the living room covers the remaining demand in heating energy. The mechanical ventilation system consist of an earth coupling system and two cross-flow heat exchangers.

Measurements and simulations

In order to understand the thermal behaviour of a zero energy house and to work out recommendations for future low energy houses, the Building Equipment Section of the Swiss Federal Laboratories of Material Testing and Research measured most of the energy aspects in one of the two zero-heating energy houses and in the cogeneration plant during 3 years [8,9]. For economical and ecological optimisations, simulation works using the program TRNSYS are in progress.

Results

- Very low energy index, between 17 and 26 MJ/m²-a, for heating and domestic hot water
- During cooking, odour spreading from the kitchen has been observed in zones with fresh air supply. From that, leakages in the heat exchanger were suspected. This has been confirmed by tracergas measurements (injection and sample points for tracer gas measurement see Figure 15).

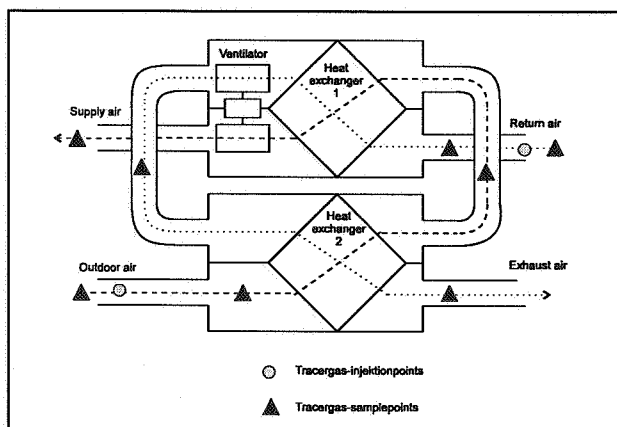


Figure 15: Position of the injection and sample points for the leak detection in the heat exchangers using tracer gas.

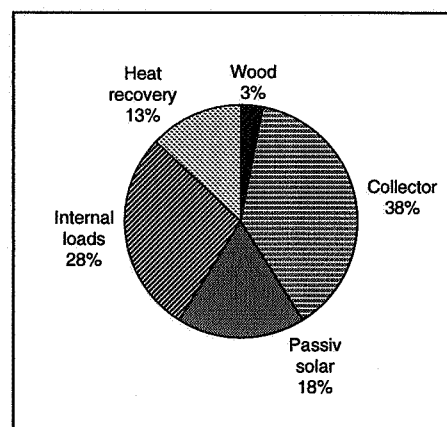


Figure 16: Distribution of heat gains for the 'Wädenswil' house

5 Conclusions

Balanced mechanical ventilation systems with heat recovery are an important element in the design of a low-energy house. Evaluations performed in several in P+D buildings prove the importance of such systems. Nevertheless, many problems, stemming from the design, the installation as well as from occupant acceptability have been encountered. Most of the failures and shortcomings observed can be tracked down to:

- Bad manufacturing of components (e.g. leakage of plate heat exchanger)
- Improper selection and installation of components (e.g. selection of over dimensioned ventilators)
- Bad system flow balancing and inadequate commissioning (resulting in big variations of the flow rates in the individual rooms)
- Too high sound emission at inlet/outlet and sound transmission
- Excessive window airing by and general poor acceptability of occupants

To overcome these problems, the following issues must be carefully considered:

- Careful design, correct manufacturing, proper installation and comprehensive commissioning of the system
- Handout of good and adapted information to the occupants
- Proper and periodically repeated checking and maintenance of the system

6 Acknowledgements

We would like to express our thanks to Peter Hartmann, Thomas Baumgartner and Heinrich Huber who have been responsible for the evaluation campaigns in the 'Hausäcker', the 'Eichrütli' and the 'Riehen' buildings respectively.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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FRENCH VENTILATION SYSTEM PERFORMANCES IN RESIDENTIAL BUILDINGS

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ABSTRACT

The comparison of the performances of ventilation systems must take into account different issues. For indoor air quality, different kinds of pollutant sources have to be defined. To make an evaluation of the results, the best approach is to consider the people exposure. Four generic pollutants are taken into account : rooms components or furniture, human metabolism, cooking activities, passive smoking. As the definition of the unit for each pollutant is free, it is useful for their comparison to press them on a common "normalised" basis. This enables to make a distinction between the simulations based on generic pollutants and their interpretation which could be derived for many kinds of pollutants without having to do additional calculations as far as the considered pollutant follows the conventional patterns. The pressure difference , for example can be related to radon issues as to running of some heat appliances. The room parameters are mainly related to humidity issues as condensation, mould growth or house dust mites. For energy issues, a distinction has to be made between the direct energy use (fans) and the heat needs due to ventilation.

Such a methodology has been defined within the IEA annex 27 project. After a presentation of the different parameters taken into account, we describe the computer code SIREN95 developed at CSTB in order to obtain the required results and we present a sensitivity analysis for the basic ventilation system used in France (mechanical exhaust).

1. - INTRODUCTION

A methodology to compare the performances of ventilation systems has been defined within the IEA annex 27 project.

After a presentation of the different parameters taken into account, we describe the computer code SIREN95 developed at CSTB in order to obtain the required results and we present a sensitivity analysis for the basic ventilation system used in France (mechanical exhaust).

2. - PRESENTATION OF THE STUDIED PARAMETERS

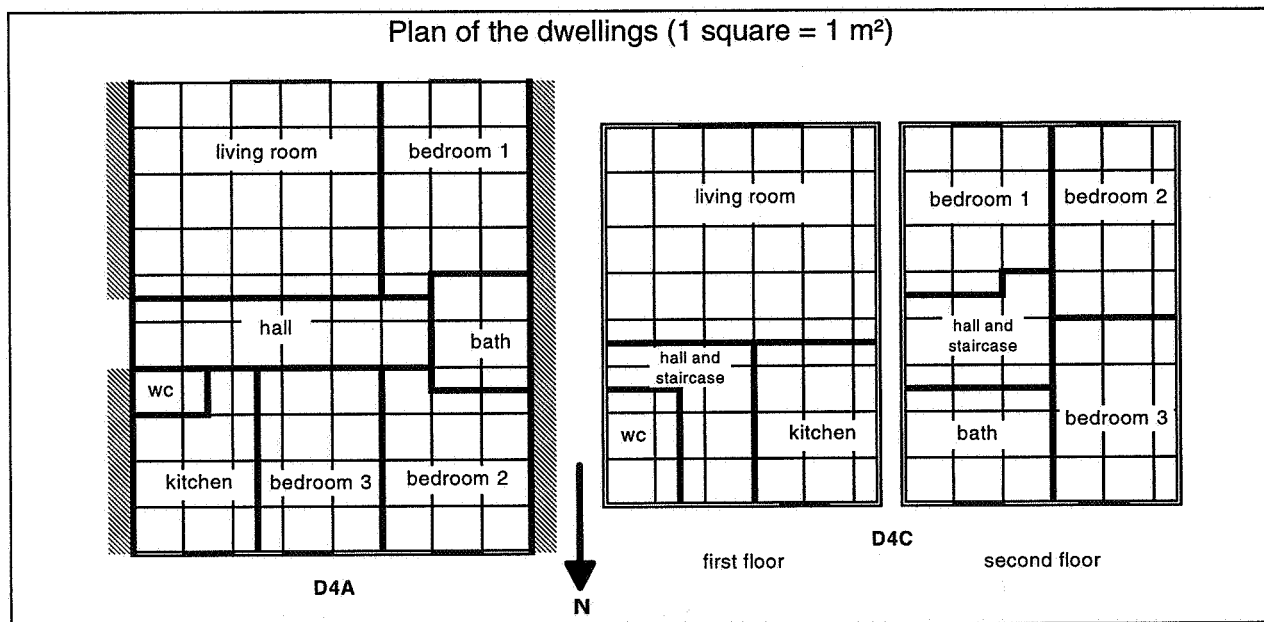
In the IEA annex 27 project ventilation systems are study : 9 dwellings (3 plans and 3 airtightness), 3 occupancy (spacious, average, crowded), 3 climates (cold, mild, warm). [1]

Results are given in term of indoor air quality (pollutant exposure for each inhabitant) and energy.

2.1 - DWELLINGS

Three dwellings are considered :

- D4A : flat located on ground floor in a four storey building.
- D4A : flat located on top floor in a four storey building.
- D4C : single family house (detached)



Leakage values are given in the table hereafter :

	n50 (ach)			
D4A	1	2.5	5	<i>0.46</i>
D4C	2.5	5	10	<i>0.67</i>

*French
std ?*

The values in italic are those given by the French standard for design and dimensioning the mechanical extract ventilation systems (they correspond to 40 m³/h under 20 pascals for the flat and to 75 m³/h under 20 pascals for the detached house) : these values are not take into account in the annex27 assumptions.

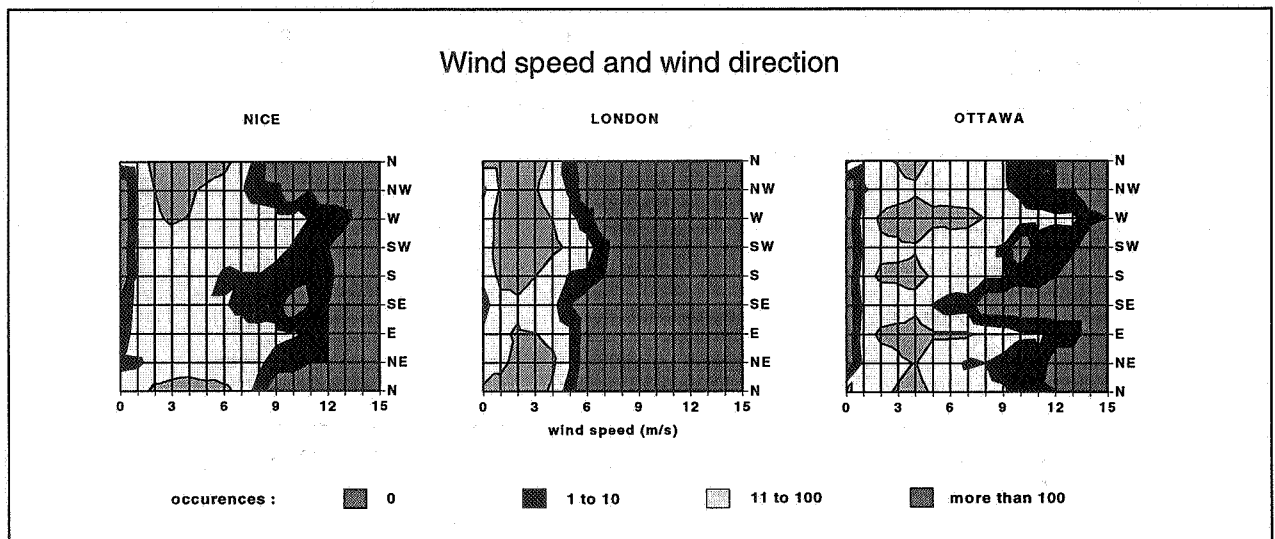
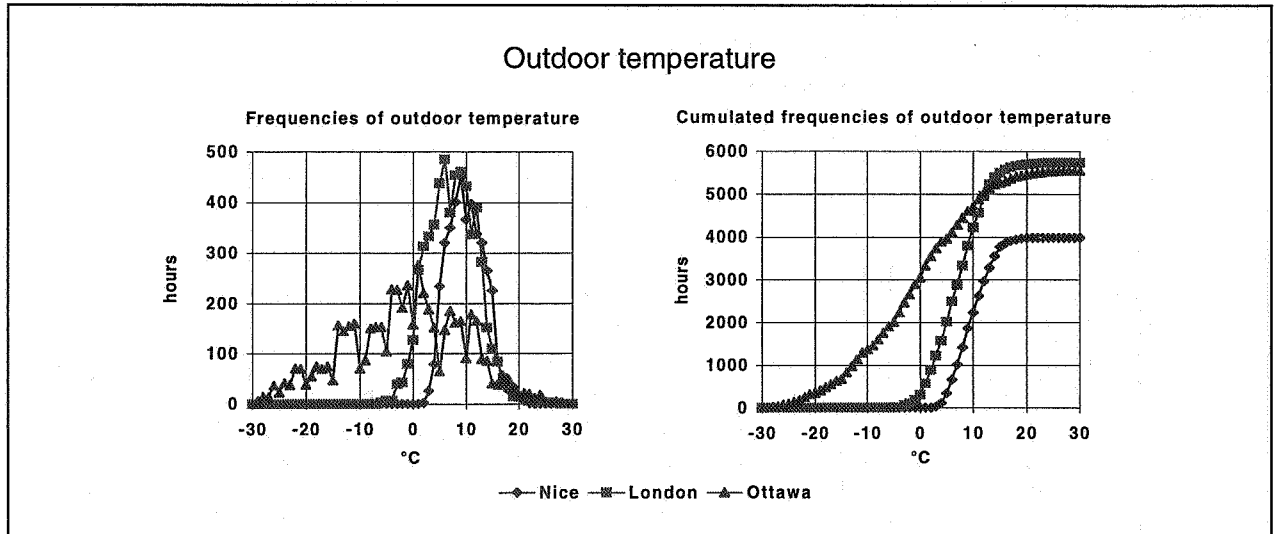
Half of the cracks are located at 0.625 m from the floor and the other half 1.875 m from the floor for the leakage 1 , 2.5 and 5 . For leakage 10 the additional cracks are located at the floor and at the ceiling.

2.2 - CLIMATES

The climates are related to meteorological data from :

- cold : Ottawa (heating season from 2nd of october to 20th of may).
- mild : London (heating season from 24th of september to 20th of may).
- warm : Nice (heating season from 13th of november to 27th of april).

The charts hereafter give frequencies of outdoor temperature and wind speed and direction for this three climates.



2.4 - INDOOR AIR QUALITY FOR PEOPLE

Definition of generic pollutants : For the study of a pollutant source, it can be noticed that the main important point for comparison is the pattern of its production (level versus time and place), whatever this pollutant is . Therefore it is possible to define some generic pollutants which will be defined only by their pattern. For human feeling and health we propose at first to base the comparisons on 5 main generic pollutants :

- Plt1 : this pollutant is based on a constant emission related to the room area. It could be related to pollutant emission by the rooms themselves.
- Plt2 : this pollutant is related to the human metabolism. It is based on the CO₂ production.
- Plt3 : this pollutant is related to the cooking activities. It is proportional to the water evaporated during cooking and could be related to odours production, as to CO or NO_x production in case of gas appliance.
- Plt4 : this pollutant is related to passive smoking . It is based on a constant production of pollutant for the hours and place when and where people are smoking.
- Indoor humidity : this one is here only related to the dryness feeling. It is not a generic pollutant as it can be expressed directly in terms of indoor relative humidity.

Results : It is possible, for each inhabitant, to give the curve of the number of hours above a pollutant level concentration C_i : $N_h(C_i)$. These results can also be given in a condensed form based on the calculation of the cumulated value above a threshold limit C_{imax} . For CO₂ the limits are 0.7 and 1.4 (in 1000 ppm) : the condensed output are expressed in ppm.h above this 2 levels. In this paper we give only results for CO₂ (1400 ppm).

2.3 - OCCUPANCY AND POLLUTANT PRODUCTION

A weekly schedule of the dwelling occupancy has be defined by IEA annex 27 [1].

The CO₂ concentration inside rooms is calculated taking into account pollutants production due to occupation, air renewal due to ventilation system operation and CO₂ outside concentration. The CO₂ outside concentration is assumed as 350 ppm.

The production of pollutants by occupants metabolism is :

		CO ₂ (l/(h*p))	H ₂ O (g/(h*p))
Adult ≥ 15 years	awake	18	55
	sleeping	12	30
Children 10 and 13 years	awake	12	45
	sleeping	8	15
Child 2 years	awake	8	30
	sleeping	4	10

The production of pollutants by occupants activities is :

Cooking g/h by person present at home			Shower g/person	Smoking
breakfast	lunch	dinner		
50	150	300	300	20 U4/h in the living when woman is present between 13 - 24 o'clock

2.5 - ENERGY NEEDS

The energy needs must be split into heat needs and electrical needs for fan.

The heat needs can be calculated knowing the air flows to the outdoor and the temperature difference between outdoor and indoor. The air flows can be separated into 3 parts :

- air exhausted by the ventilation system,
- air exfiltrated through the envelope,
- airing (by opening windows : not studied in this paper).

The average air flow and air change per hour are the direct averages during the heating season of the overall air dwelling air flow. Nevertheless, it is not of direct interest as it is neither related to indoor air quality nor to heating needs. For example with the same average air flow, the heat needs will be increased if the ventilation in winter is higher (passive stack systems) and decreased if the ventilation is lower (humidity controlled systems).

Focusing on a single way to express the heat needs efficiency independently of the climate conditions, it is better to calculate heat needs equivalent air flow rate and air change rate which

are the constant air flow (or air change rate) which would lead to the same heat needs as the ones calculated. It is simply calculated by :

$$Q_{ave} = \frac{\int 0.34 \times Q(t) \times (T_i - T_e(t)) \times dt}{\int 0.34 \times (T_i - T_e(t)) \times dt}$$

The electrical needs will be calculated on the whole year if this corresponds to the system running.

The power is considered to be 40 W for a dwelling : the annual consumption is 350 kWh.

3. - THE COMPUTER CODE *SIREN95*

The computer code *SIREN95* is an evolution *SIREN* ("Simulation du RENouvellement d'air") developed in C.S.T.B [2]. It is used to calculate the air flow throughout the entire heating season (about seven months) in a dwelling. The code uses hourly meteorological data (temperature, relative humidity, wind speed and orientation) ; occupancy and pollutants production (CO₂ and H₂O) are defined with an half an hour step. Each component (air inlets, outlets, cracks, fans, windows, ...) is characterised by its flow rate curve as a function of the pressure difference and also when relevant, of the temperature or relative humidity. Pollutants and humidity concentrations are assumed to be uniform in each room.

In *SIREN95* (unlike *SIREN*) internal pressures are assumed an hydrostatic field ; the inside temperature is considered to be constant in an horizontal plane (only vertical gradient : stack effect is taken into account). The calculation procedure has been changed : now the curves of components can be given by functions (not necessary smooth) or tables : a possible hysteresis is taken into account.

4. - RESULTS FOR THE BASIC SYSTEM USED IN FRANCE

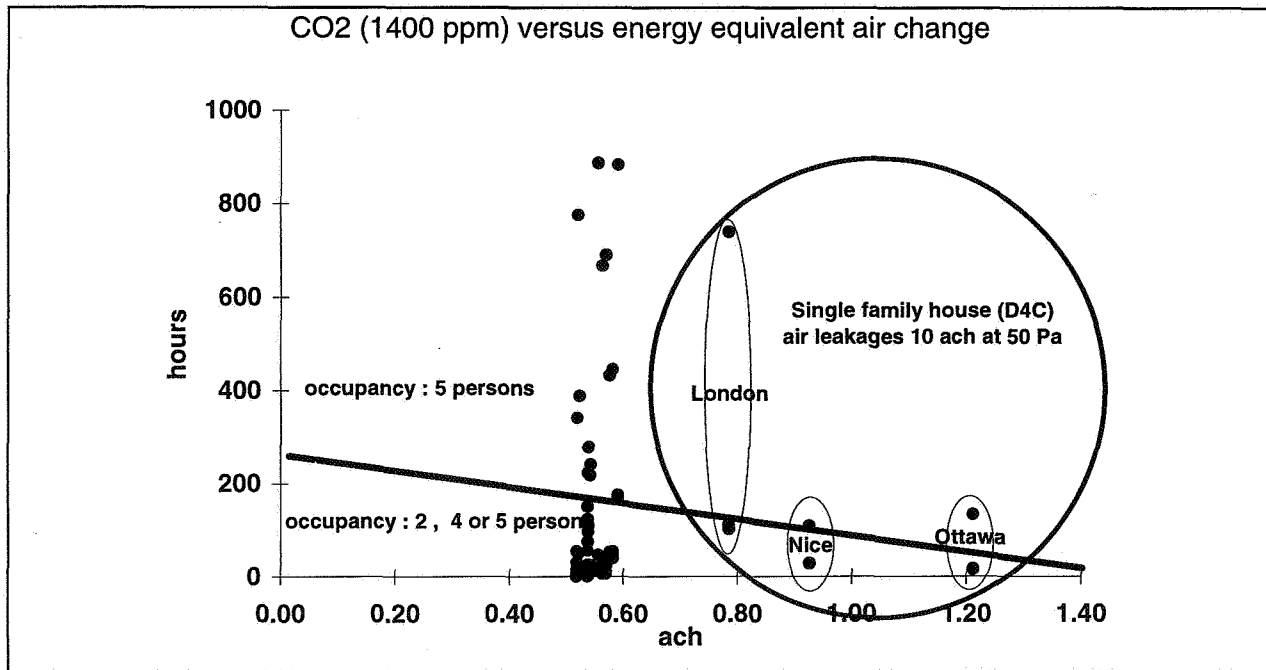
The French mechanical extract system has been described otherwise [3], [4] ; is a permanent extract system, with self regulated inlets in each habitable rooms and outlets in each service rooms : the flows values (for a 4 principal rooms dwelling) are given hereafter :

	living room	bedroom	kitchen	bathroom	toilets
inlets (m3/h under 20 Pa)	45	30	/	/	/
outlet (m3/h)	/	/	45 (120 at cooking)	30	30

Results show that the equivalent air change rate is little dependant of the air leakages, except for the single family house when air leakages are 10 ach under 50 pascals (circle on the chart).

Air quality (CO₂ based) depends on :

- occupancy (all the better since occupancy is low),
- climate (depends on the length of the heating season and on windspeed and wind direction)
- air leakages.



Energy :

	Ottawa	London	Nice
Fans (kWh)	350	350	350
Heat losses (kWh) for 0.55 air change	4470	2690	1470
Total (kWh)	4820	3040	1820
Part of fans consumption (%)	7.3	11.5	19.2

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

**17th AIVC Conference, Gothenburg, Sweden,
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EFFICIENCY CHARACTERISATION OF VARIOUS VENTILATION CONFIGURATIONS

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Synopsis

The study concerns the ventilation of a parallelepiped shape room by means of several systems whose supplying and extracting methods differ, so the different thermic conditions applied to limits. To qualify the efficiency of each of these systems in relation with the various current criteria, we carried out measurements by means of a tracer gas, both with a transitory and a permanent flow.

At the same time, numerical simulations were carried out by means of a CFD code which solves the equations of the fluids mechanic, material and heat transfers associated with flows. These calculations results, after measurements validation, enable to accurately know the air movements in the ventilated room as well as the tracer gas concentrations distribution in the tested configurations.

From these calculation results, was also demonstrated how the complex behaviour of each of these ventilation systems can be characterised owing to the possibility of identifying it to a simple model, using a reduced number of parameters. These typical parameters can in turn be used to implement the system performance evaluation criteria.

These criteria allow to compare the various studied ventilation systems and class them in performance order. It can be seen that this classification can be modified depending on the selected criterion as well as the chosen ventilated area.

The study results equally show the interest of the use of numerical simulation together with experiments, thus extending the range of results in order to generalise conclusions.

1. Introduction

In this paper, a methodology is described to determine the performances of different ventilation systems in a room with a local emission of pollutant. This methodology is based on numerical calculation of air flows and transfers. First of all, this simulation has been applied to a room where measurements were done, in order to validate the calculation.

Then the methodology was applied to four elementary ventilation systems in a room, in order to qualify their efficiency using performance criteria which are defined in this study.

It was shown that the methodology can be used to study and to qualify the efficiency of a defined ventilation strategy in a room.

The comparisons between the different studied configurations show that the performance classification strongly depend on the criteria and on the observation point into the room.

2. Measurements

The measurements used for comparisons with calculations were done in a parallelepipedic room represented in Figure 1.

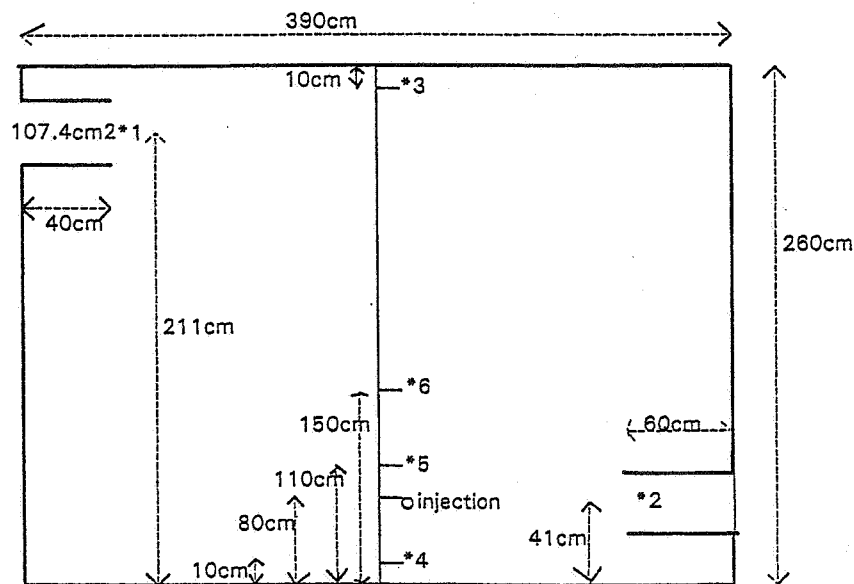


Figure 1 : Test room with concentrations measurement points

In the considered ventilation system, air is supplied on the top of one wall (middle of the wall) and extracted at the bottom of the opposite wall (middle of the wall).

The pollutant is N_2O ; it is generated with constant flow, in the center of the room, at a 0.8 m height.

This type of pollutant has been chosen because it is possible to measure low concentrations and because its density is the same that the CO_2 one, which represents one of the most common pollutant in occupied spaces.

The positions of the measurement points are shown on the same Figure 1 ; four points are on the vertical, at center and different heights ; one control point is in the air inlet, another is in the air outlet.

The experiment is divided in two periods : after establishing the ventilation scheme, the pollutant is generated, continuously, until constant concentrations are reached ; then the pollutant generation is stopped, and the ventilation goes on, until the initial zero concentration is get.

Measurements give the concentrations evolution during the different periods : continuous generation period, transitory increasing and decreasing periods.

3. Numerical simulation

The simulation consists in integrating Navier-Stokes equations in the studied field. The method of the finite volumes is used, with a three-dimensional representation of the space.

The FLUENT CFD is used. The turbulence is taken into account with a two equations model (k-e).

The pollutant diffusion in the air is also represented (molecular and turbulent diffusion) ; the buoyancy effect is also represented, and it induces different forces on the local concentration of pollutant which is denser than the air. The calculation were done in isothermal conditions, with adiabatic walls ; the pollutant is generated at the same temperature as the indoor air.

As in the experiments, the calculations were done for different specific periods : stationary period with continuous generation of pollutant, and transitory period representing the evacuation of N₂O after a pollution period.

The Figures 2 and 3 show the comparisons between measurements and calculations for both periods.

Figure 2 shows the vertical profile of N₂O concentration at the center of the room.

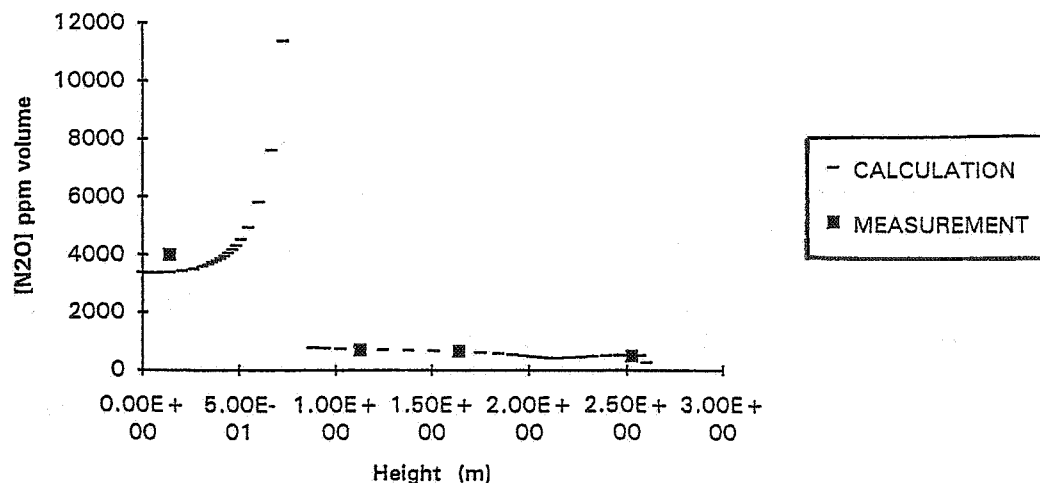


Figure 2 : Profile of concentration in the middle of the room

Figure 3 shows the decrease of the N₂O concentration at different measurement points, during the time, after the end of the generation period.

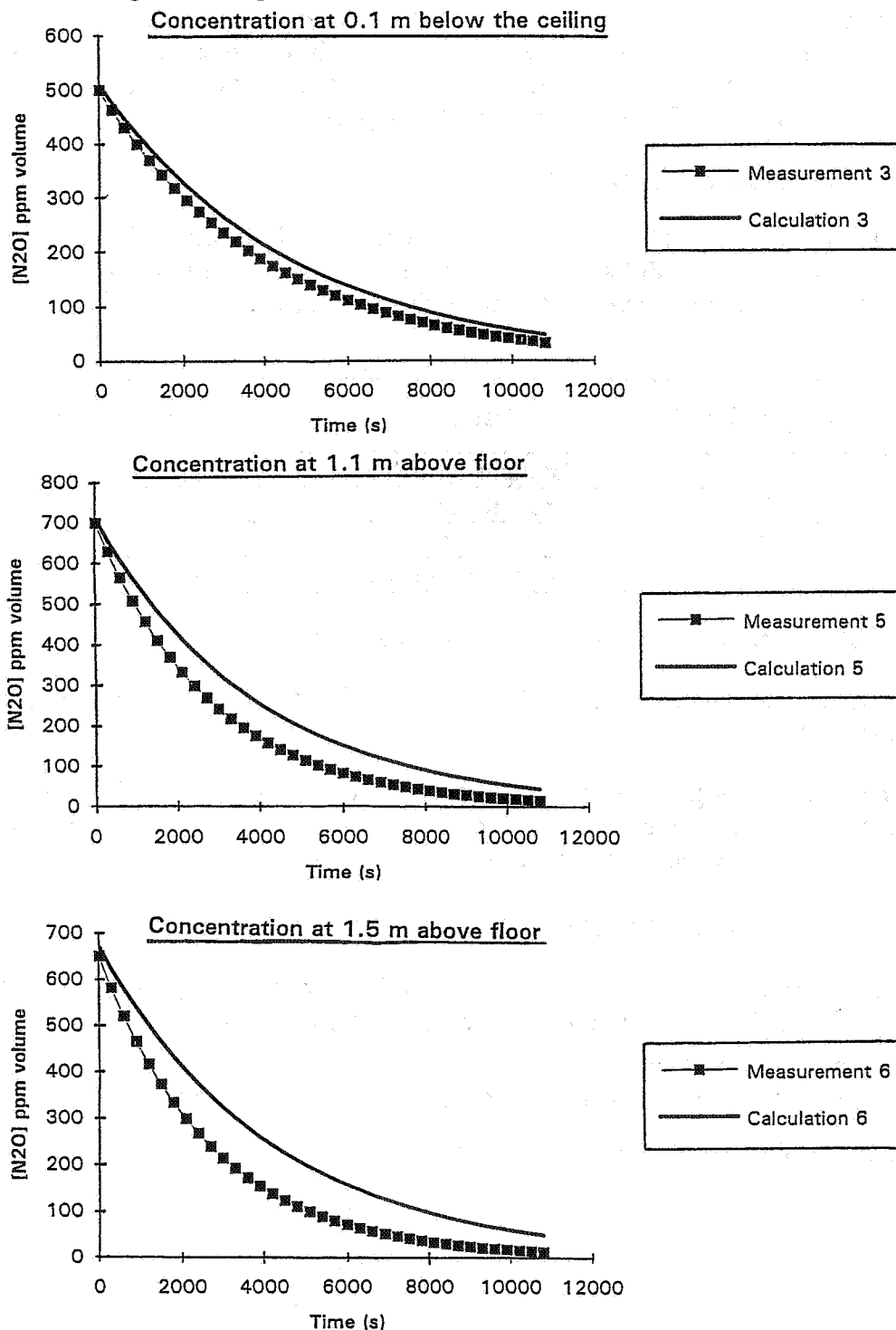


Figure 3 : Evolutions of concentrations at different points comparison between measurements and calculation

Measurements and calculations are congruent ; so, the simulation can be used for a practical study.

4. Exploitation of numerical results

In this part of the study, different configurations are compared, according to their efficiencies, which are established using the criteria coming from the simulation results : results in steady state conditions with continuous pollutant generation, and results in dynamic conditions when the concentration decreases after a pollution phase.

4.1. The studied configurations

On the geometrical base defined in paragraph 2 (Figure 1), four configurations were considered. They differ in the relative positions of supply and exhaust devices. The configurations are described in Table 1. There are all mixing ventilation.

Table 1

Configuration n°	Name	Supply	Exhaust
1	"straight up-up"	up	up
2	"cross up-down"	up	down
3	"cross down-up"	down	up
4	"straight down-down"	down	down

In all the cases, the air change rate is two volumes per hour (so an air flow of 108 Nm³/h).

The pollutant used here is the CO₂ ; it is introduced in the same way as with the measurements.

The calculation gives, among the results, the pollutant concentrations in two particular points situated in the middle of the room at 0.1 m and at 1.1 m above the floor. These results, in the steady state and dynamic conditions are used to qualify each of the ventilation systems.

4.2. Ventilation efficiency in steady state conditions

It is characterised with a criterion called "pollutant removal effectiveness" [2, 4, 5], which represents the capability of the system to evacuate a continuous production of pollutant. In the simple case where the supply air is free of pollutant, it is defined by the ratio between the pollutant concentration at the exhaust $\overline{[CO_2]}$ and the pollutant concentration at the considered point $[CO_2]_{\infty}$ in the ventilated room :

$$\eta_e = \frac{\overline{[CO_2]}}{[CO_2]_{\infty}}$$

where $\overline{[CO_2]}$ is the concentration in steady state conditions at the exhaust, which is simply calculated with the ratio between the pollutant generation flow on the ventilation air flow.

The better the local ventilation effect is, the bigger the pollutant removal effectiveness is.

4.3. Ventilation efficiency in dynamic conditions

As the figure 4 gives an example, the results show that during the period following the pollution phase, the decrease of the concentration in one point is always very close to a linear system response :

$$[CO_2] = [CO_2]_{\infty} \cdot \left[1 - \exp\left(\frac{-t}{T}\right) \right]$$

where t represents the time. This fact allows to characterise the dynamic behaviour of the ventilation system at a constant time.

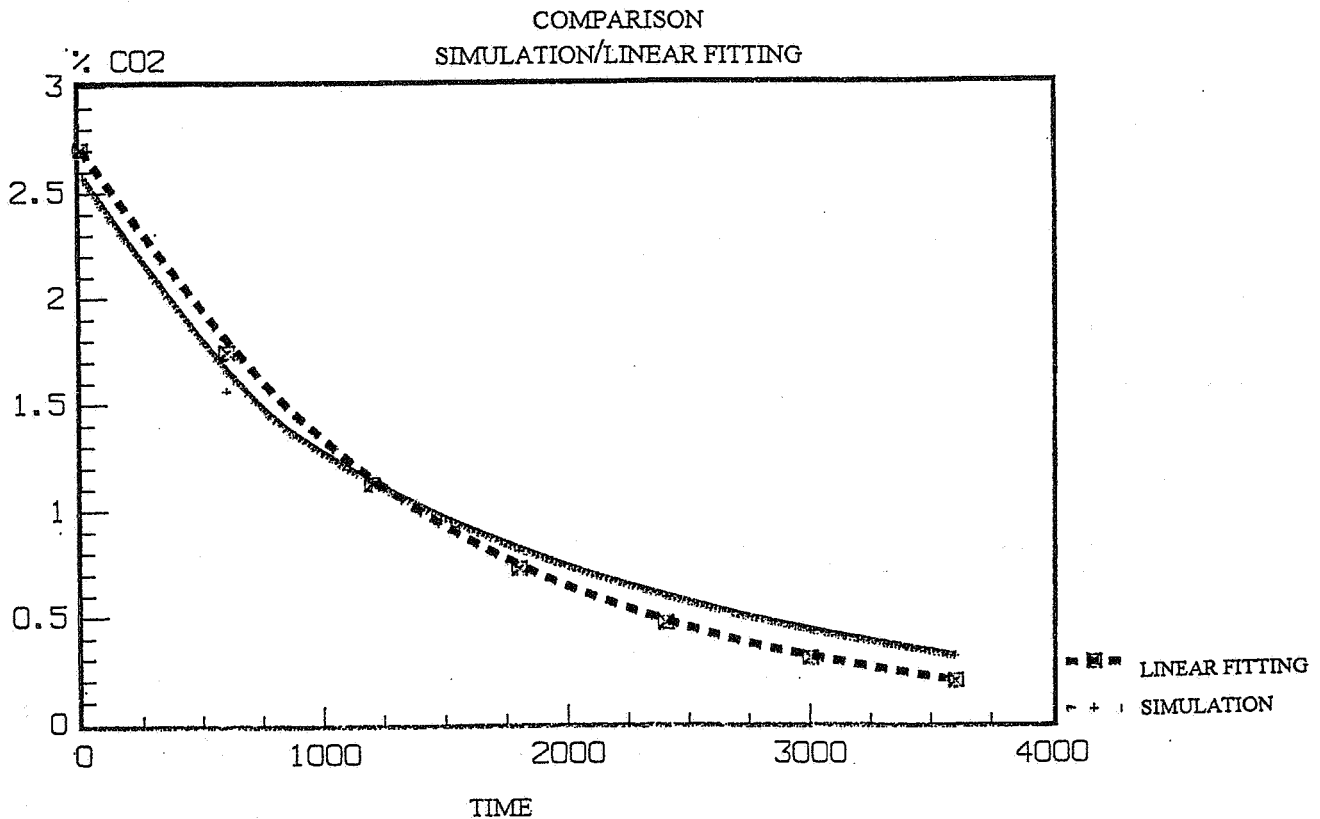


Figure 4 : Decrease of concentration in one point, during the time - Approximation with simple model

Then, this constant time can be used to determine the "air change efficiency". This criterion characterises the efficiency of the ventilation system in quickly replacing the polluted indoor air [3] with new (or fresh) air. This criterion is expressed by the ratio between the average duration for changing air and the previously defined constant time :

$$\rho_r = \frac{\bar{\tau}}{T}$$

The quicker the ventilation is able to evacuate a temporary pollution from indoor air, the bigger the air change efficiency is.

4.4. Application to the studied configurations

The previous defined criteria are calculated for the four studied configurations, and for the two observation points. The results are summed up in Table 2.

Table 2

Configuration	Position (observation point)	Pollutant removal effectiveness	Air change efficiency
1	1	0.704	1.044
	2	0.799	0.847
2	1	0.897	1.318
	2	1.070	1.082
3	1	1.241	1.115
	2	1.006	0.915
4	1	1.192	1.009
	2	0.957	0.834

These values allow to classify the four studied ventilation systems, according to their efficiencies.

If we consider the observation point n°2, the more closest to the occupied zone, then we get the following decreasing classification related to both criteria (Table 3).

Table 3

Classification	Pollutant removal effectiveness	Air change efficiency
1	"cross up-down"	"cross up-down"
2	"cross down up"	"cross down up"
3	"straight down-down"	"straight up up"
4	"straight up up"	"straight down-down"

Then, if we are interested by the pollution at the floor level, we can choose the observation point n°1. The classification is then given in Table 4.

Table 4

Classification	Pollutant removal effectiveness	Air change efficiency
1	"cross down-up"	"cross up-down"
2	"straight down-down"	"cross down up"
3	"cross up-down"	"straight down-down"
4	"straight up-up"	"straight up-up"

As the ventilation system in configuration "cross up-down" seems to be the more efficient in the majority of the cases (all configurations are mixing ventilation), the results show that the relative interest of one system compared to another differs according to the criteria and to the position into the room.

5. Conclusion

This work, made on four ventilation systems, showed the interest in using the numerical simulation which allows to study in an accurate and detailed manner, complex and dynamic phenomena.

We also showed how the detailed results in the concentrations evolution in time given by the calculation, can be reduced, by a simple model identification, to some specific parameters. These parameters can be used, at the end, to apply the evaluation criteria of the systems performances.

These criteria allow to compare the different ventilation systems, but we can see that the performance classification depends on the chosen criterium, and on the region of the space in the considered ventilation room.

Today, comparisons and calculations are made with non-isothermal conditions.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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VENTILATION IN HOUSES WITH DISTRIBUTED HEATING SYSTEMS

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*** LTEE of Hydro-Quebec, Canada**

**** Stricker Associates Inc.**

***** Canada Mortgage & Housing Corporation (CMHC)**

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Ventilation in Houses with Distributed Heating Systems

Denis Parent, LTEE of Hydro-Québec,
Saul Stricker, Stricker Associates Inc.,
Don Fugler, Canada Mortgage and Housing Corporation (CMHC)

Synopsis

The LTEE laboratory of Hydro-Québec, in collaboration with Canada Mortgage and Housing conducted an indoor air quality study involving 30 single family detached houses heated with electric baseboard heaters in the vicinity of Trois Rivières during the 1993-94 heating season. The houses were selected according to the measured air leakage at 50 Pa. so as to have a sample distribution similar to the distribution of air leakage of houses in the province of Quebec. The "source strength" of several air pollutants were calculated from measurements of ambient pollutant levels and total ventilation during a one-week test. In addition, the indoor CO₂ and humidity levels were recorded in eight of the houses continuously during the heating season.

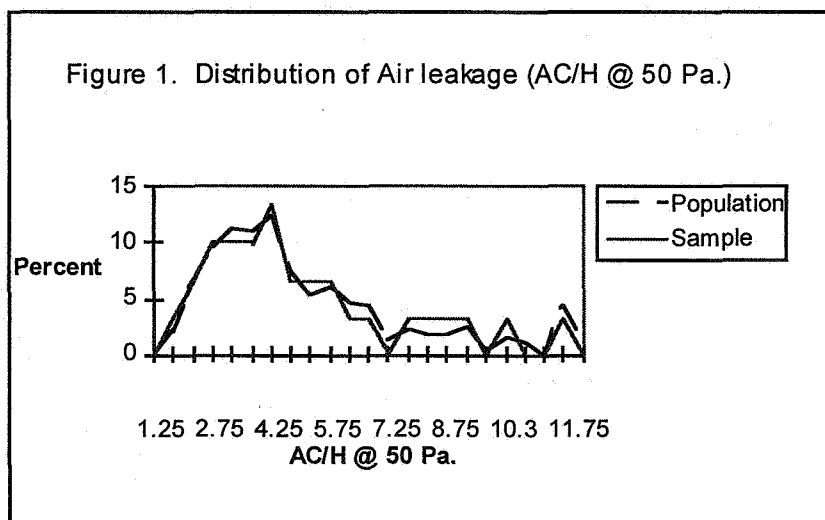
The level of CO₂ in the master bedroom was found to follow fairly closely the CO₂ level in other parts of the building including the basement (within about 200 ppm) except when the bedroom door was closed. With the room occupied and the door closed, CO₂ levels in the bedroom increased steadily during the night until morning, when the door was opened, to levels in excess of 3 500 ppm with one person, and in excess of 4 500 ppm with two persons. Model studies using the measured pollutant source strengths and measured equivalent leakage areas of the buildings indicated that the recommended health guidelines for airborne respirable solid particles (RSP's), CO₂ and formaldehyde are exceeded during periods of low total ventilation, coinciding with mild outdoor temperatures and low wind conditions. It was observed that kitchen and bathroom fans originally installed in some of these houses were not operated by the occupants for sufficiently long times to affect the quality of indoor air.

Various different methods of ventilating some of the houses were tested, including quiet replacement exhaust fans, mixing fans for indoor air, and a fresh air intake and mixing system. The effects of operating various air handling systems were monitored by keeping track of indoor CO₂ and relative humidity in the master bedroom, and occupancy in person-hours per day. Quiet replacement fans noticeably improved indoor air quality when these were operated over 50% to 100 % of the time. An area of remaining concern is the fact that exhaust only systems accentuate the negative pressure in the basement by raising the level of the neutral pressure zone in the building, and may enhance the flow of soil gases into the basement. A system which mixed indoor air between the basement and the main floor also reduced the average level of indoor pollutants. The system was designed to create a pressure difference between the main floor and the basement, causing a slight pressurization of the basement. A system designed to introduce 5 L/s of outdoor air and to mix it with 55 L/s of indoor air for tempering was installed to draw air from the hallway and deliver the mixed air into each of three bedrooms. This system was capable of maintaining CO₂ levels in the master bedroom below 1000 ppm with two occupants in the room and the door closed.

1. Introduction

Hydro-Québec initiated a study of air tightness and ventilation of residences to determine whether building air-tightness alone could be used as the determining criterion for air-tightening the building so as to improve energy efficiency. Using indoor air quality as a measure of building's ventilation performance, a field study was undertaken involving 30 houses heated with electric resistance baseboard heaters in the Trois-Rivières area. The houses were selected on the basis of their air leakage determined via a blower door test at 50 Pa. By design, the sample of 30 houses was selected to have a similar distribution of air leakage as a random sample of several thousand houses in Québec. Figure 1 illustrates the distribution of house leakage determined from a very large random sample of the population (Eval-Iso Programme

/1/), and the distribution of leakage in the thirty house sample of houses selected. The average indoor air contaminant levels during a one-week period was determined in each of the houses, along with the average (total) ventilation during the same period.



Instrumentation and Measurements Made

The physical size and shape of each house was noted along with the leakage characteristics as

determined via the fan pressurization method. During a one week period between December and March, air sampling kits as indicated on Table 1 were installed in each of houses for a period of one week to determine level of the air contaminants of interest:

Table 1. Air Contaminants Monitored and sampling Systems Used

Variable	Sampling System
Average air change rate	PFT tracer method. This method uses 1, 2, or 3 sources and several samplers per house for a 7-day period. The materials and analysis a carried out by Brookhaven National Labs in Long Island.
Indoor air particulates	Pumped occupational hygiene filter cassette for 7-day period (micrograms per cu.m)
Total VOC's:	Pumped special multi-absorption tube for 3-hour period(will identify 6 major components in percent of toluene equivalent)
Formaldehyde:	AQRI passive Dosimeter (tube with a lid) with +/- .02 ppm resolution
Radon:	Canadian Institute for Radiation Safety (CAIRS) pumped filter sampler for 7-day period
Carbon Dioxide and Water Vapour	YES 203 recorder in living room

The average contaminant levels along with the house volume and average ventilation rate for the test week were used to calculate the "source strength" of each airborne contaminant as follows: The source strength S is defined as:

$$S = L \times A \times V \quad \dots(1)$$

where: L = average contaminant level over a one-week period (mg/cu. m.)
A = average air change rate over a one week period (AC/H)
V = house volume (cu. m.)

Using the indicated units, S would be units of mg/hr.

The average air change rate for each house during the test week was determined using the PFT tracer method. The tests in the 30 houses were conducted sequentially in small groups due to the limited number of instruments available. Although the weather conditions during the determination of average air change rates would not have been identical for the group of tests, this is of no concern for the evaluation of pollutant source strength for air contaminants since the controlling factors are essentially independent of weather.

The results of these tests and calculations are tabulated on Table 2. On this table, the natural air change rate is expressed in litres/second. The accuracy of the source strength estimate is affected by the uniformity of diffusion of airborne pollutants throughout the entire building, and by the stability of the source strength of each contaminant over time. While it is known that some of the contaminants such as formaldehyde are released at a fairly steady rate by surfaces and that other contaminants such as carbon dioxide and some of the moisture are produced intermittently during occupancy or during certain activities, the average measurement during a one-week period is expected to capture a good representative sample of the pollutants which naturally occur in a house during a full week cycle.

Table 2. Source Strength of Airborne Contaminants in 30 Québec Houses

House #	Vol	Nat. Air Change	Occu-pancy	RSP	VOC's	Form.	Radon	CO ₂	Moisture.
Units:	cu m	L/s	p-hr/day	mg/hr	mg/hr	mg/hr	µCi/hr	g/hr	g/hr
1*	444	14.8	102	4.42	21	3.54	11	59	259
2*	306	26.4	89	7.21	38	5.37	19	120	488
3	486	14.9	39	1.44	37	6.18	32	64	295
4***	505	19.6	83	1.41	120	8.61	28	107	
5	515	14.3	77	1.96	36	5.19	52	96	358
6*	517	14.4	69	3.83	16	5.28	41	61	342
7	487	13.5	55	1.12	19	3.35	29	36	239
8**	737	26.6	83	3.35	220	4.95	57	104	332
9	478	27.9	82	1.91	30	4.57	40	109	---
10	371	8.2	69	0.50	9	2.56	18	25	---
	513	34.2	74	7.39	86	10.75	25	105	---
12	412	22.9	63	2.06	7	3.55	33	149	---
13	504	81.2	121	7.60	58	7.55	175	139	---
14*	548	22.8	69	6.08	16	5.46	33	92	---
15	429	22.6	40	2.12	41	4.31	16	52	---
16	478	26.6	96	1.72	48	5.76	19	166	---
17	387	39.8	125	2.72	86	7.57	29	89	---
18	397	41.9	32	5.28	60	6.12	60	141	---
19	325	12.6	39	1.18	32	1.90	82	104	---
20	519	31.7	101	1.26	80	1.97	69	121	---
21*	270	15.0	39	0.59	38	4.38	22	47	---
22*	426	39.1	90	2.95	56	7.95	141	199	---
23*	441	22.1	80	3.41	198	2.34	111	64	---
24*	221	23.3	69	2.18	59	3.31	50	102	---
25	748	58.2	69	2.09	63	5.41	84	91	---
26*	444	27.1	24	---	39	3.6	78	44	---
27*	231	24.4	74	3.69	26	4.10	70	91	---
28	456	63.3	90	5.02	46	7.01	46	220	---
29*	418	19.7	59	2.34	21	3.93	313	96	---
30*	455	19.0	99	6.35	68	4.28	232	153	---
Means	449	27.6	73	3.21	55.9	5.03	67.1	102	330

* House where smoking takes place

** House with attached garage

*** Epoxy resin used in basement

Please note that the natural ventilation in AC/H is converted to a rate expressed in L/s (by multiplying by the internal volume of each house) for consistency of units in the evaluation of the source strength. The results are also presented graphically in the form of histograms for each contaminant on Figures 2 to 7.

3. Ventilation model and implied ventilation requirements

The source strength estimates can be used to determine the minimum ventilation required in order to maintain the air contaminant levels within the recommended maximum levels in all houses. Rearranging, equation (1) becomes:

$$\text{Air change rate (A x V)} = S / L \text{ in cu.m /hr(2)}$$

This equation can now be used to calculate the minimum level of ventilation required in order to maintain the indoor pollutant level below the recommended maximum concentration, based on the source strength of the pollutant. By substituting for L with the maximum recommended pollution level for each contaminant in turn and inserting the observed corresponding source strength for each house into equation (2), we obtain the estimated minimum ventilation required in units of (cu. m/hour). By applying the factor (1000/3600), we convert the units of the answer to litres/second. The recommended guidelines for long term and short term exposure levels to the pollutants of interest are presented on Table 3 below. Taking the acceptable long term exposure levels indicated on Table 3, we have calculated the minimum ventilation levels in each of the houses in the study in L/s. The results are tabulated on Table 4

Table 3. Exposure Guidelines

Contaminant	Acceptable Long Term Exposure Range	Acceptable Short-term Exposure Range
Respirable Solid Particles < 2.5 µm mass median aerodynamic diameter	40 µg/cu m*	100 µg/cu m (1 hour)
Total Volatile Organic Compounds	0.2 mg/cu m for no irritation** 3 mg/cu m and above results in complaints from occupants*	
Radon	4 pCi/L (EPA, USA) 800 Bq/cu m*	21.6 pCi/L*
Formaldehyde	60 µg/cu m (0.05 ppm)*	120 µg/cu m (0.10 ppm)*
Carbon Dioxide	3500 ppm *(Health Canada) 1000 ppm (ASHRAE Std 62-1989)	

* Health Canada: "Exposure Guidelines for Residential Indoor Air Quality"

** ASHRAE Handbook of Fundamentals, 1993, p 37.7

The minimum ventilation level required to maintain the levels below the recommended guidelines has been calculated using equation (2) for each pollutant of interest and for each house. The results are presented on Table 4. It is interesting to note that the average amount of ventilation required for formaldehyde, RSP's and carbon dioxide (1000 ppm) are nearly identical at 22 L/s.

Fig. 2 Histogram of RSP Source Strengths

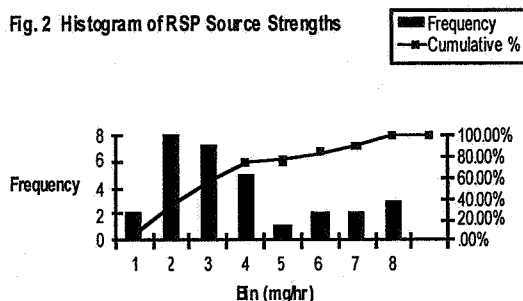


Fig. 3 Histogram of VOC Source Strengths

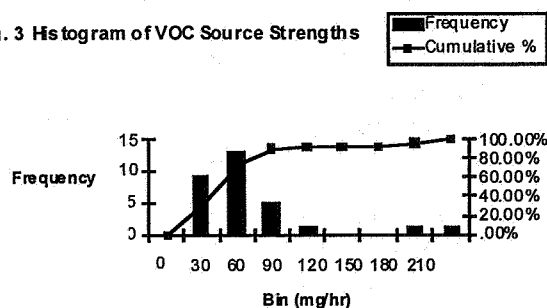


Fig. 4 Histogram of Formaldehyde Source Strengths

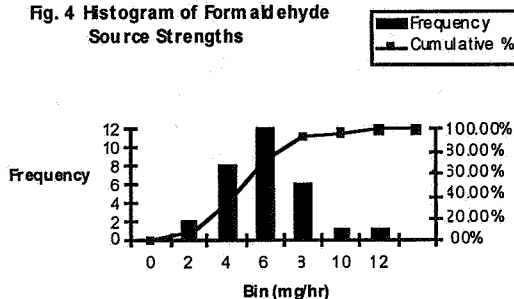


Fig. 5 Histogram of Radon Source

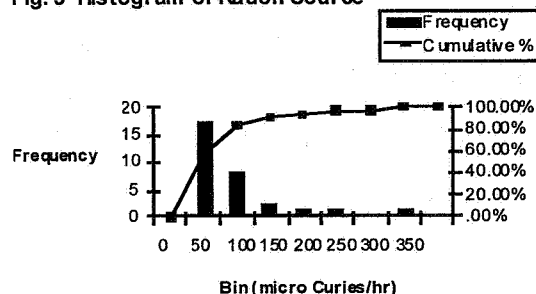


Fig. 6 Histogram of Carbon Dioxide Source Strength

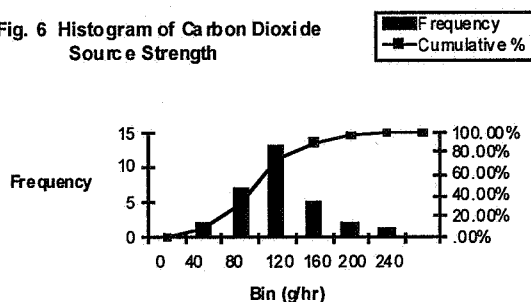
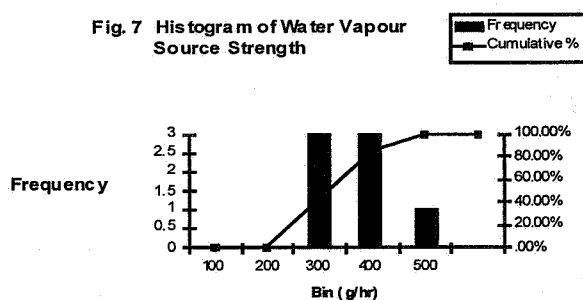


Fig. 7 Histogram of Water Vapour Source Strength



Ventilation Model

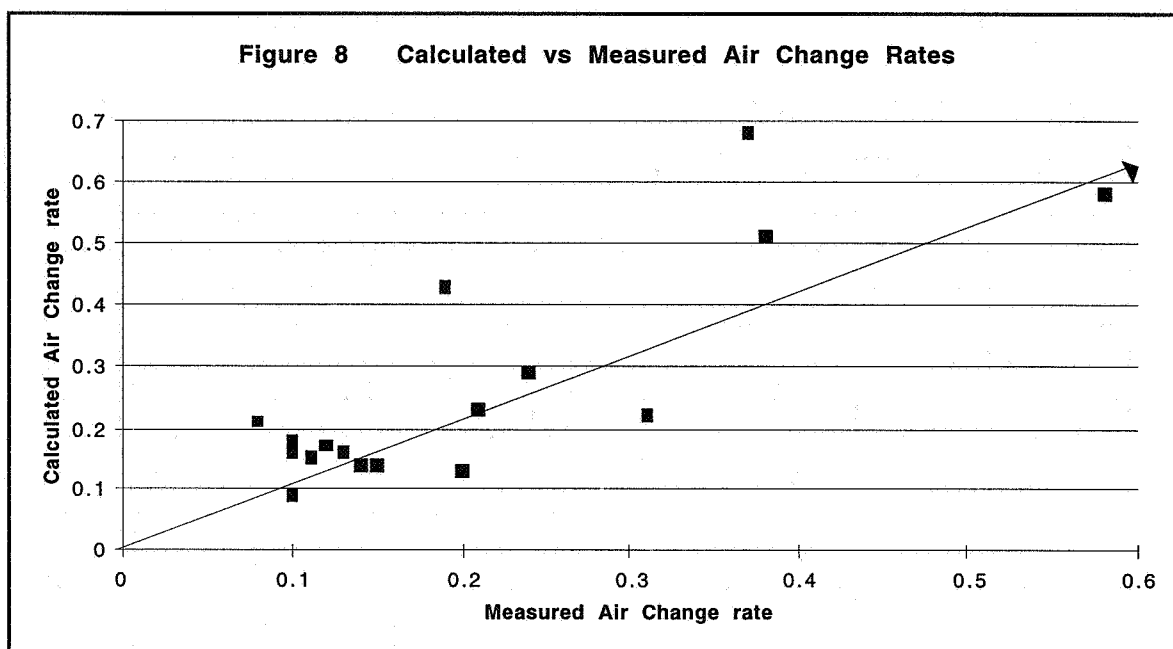
A ventilation model which was developed by CMHC, called AQ1, was used to estimate the natural ventilation during a one-week period in order to compare the results with the ventilation rates actually measured in eighteen of the houses during the 1993-4 winter field test. Once "calibrated" to a house, the ventilation model is capable of predicting the ventilation rate and pollutant levels on an hourly basis or on a monthly average basis from given house leakage characteristics, source strength information and weather data. The model has the capability of taking into account the use of ventilation equipment and wood-burning appliances by keeping track of schedules of equipment use. The method is more fully explained in Reference 2 (Appendix). The reliability of the model was tested by comparing simulation results against measured air change rates in the test houses measured during the one-week test employing the PFT tracer method. These results are presented graphically on Figure 8. The R-squared for the relationship between measured and calculated air change rates was 0.65, and the standard error of estimate was 0.1 air changes per hour.

Table 4. Minimum Ventilation Required to Control Pollutants to within Recommended Levels in Litres/second

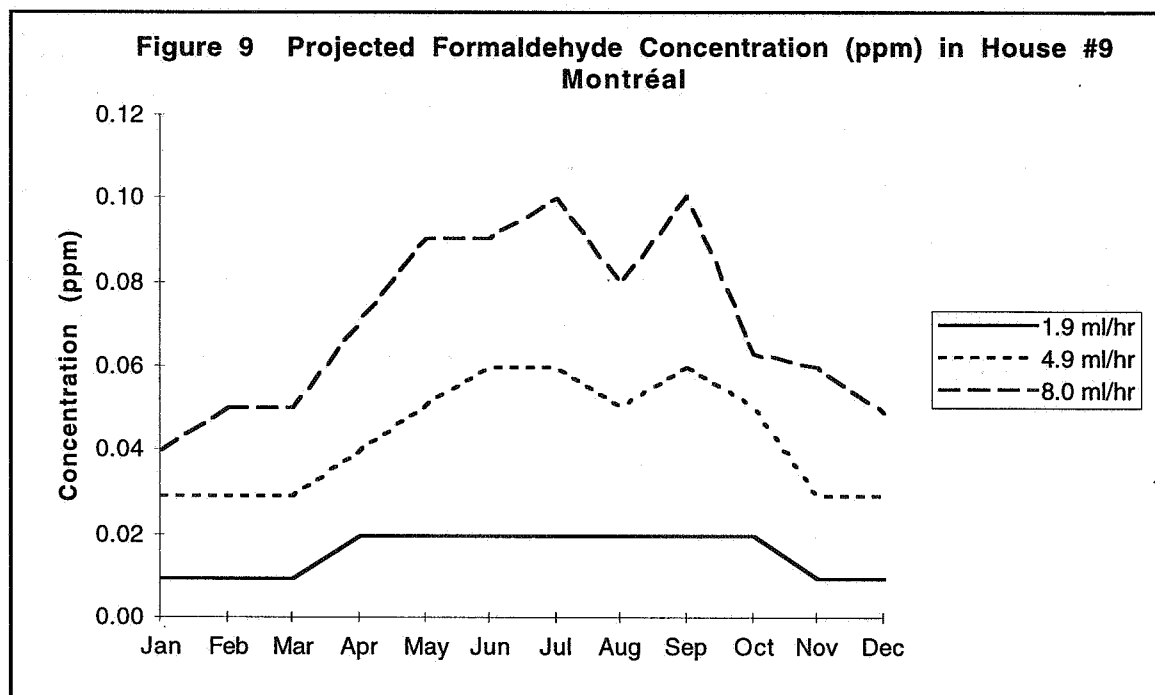
House #	RSP	VOC	Formald.	Radon	CO2 *	CO2 **	Limiting Pollutant
Limiting Level:	40 µg/cu m	3 mg/cu m	60 µg/cu m	4 pCi/L	3500 ppm	1000 ppm	
1	31	1.9	17	0.0	2.8	13	RSP
2	50	3.3	25	0.5	5.2	26	RSP
3	10	3.3	29	0.5	2.8	14	Formald.
4	10	11	40	0.5	4.7	23	Formald.
5	14	3.3	24	0.9	4.2	21	Formald.
6	26	1.4	25	0.5	2.8	13	RSP
7	7.6	1.9	16	0.5	1.4	7.6	Formald.
8	23	20	23	0.9	4.7	23	RSP/Form.
9	13	2.8	21	0.5	4.7	24	CO2 **
10	3.3	0.9	12	0.5	0.9	5.7	Formald.
11	51	8.0	50	0.5	4.7	23	RSP
12	14	0.5	17	0.5	6.6	32	CO2 **
13	53	5.2	35	2.4	6.1	30	RSP
14	42	1.4	26	0.5	4.2	20	RSP
15	15	3.8	20	0.0	2.4	11	Formald.
16	12	4.2	26	0.5	7.6	36	CO2 **
17	19	8.0	35	0.5	3.8	19	Formald.
18	37	5.7	29	0.9	6.1	30	RSP
19	8.0	2.8	9.0	0.9	4.7	22	CO2 **
20	8.5	7.6	9.0	0.9	5.2	26	CO2 **
21	4.2	3.3	20	0.5	1.9	9.9	Formald.
22	20	5.2	37	1.9	9.0	43	CO2 **
23	24	18	11	1.4	2.8	14	RSP
24	15	5.7	15	0.5	4.7	22	CO2 **
25	15	5.7	25	0.9	4.2	19	Formald.
26	---	3.8	17	0.9	1.9	9.4	Formald.
27	26	2.4	19	0.9	4.2	20	RSP
28	35	4.2	33	0.5	9.9	47	CO2 **
29	16	1.9	19	4.2	4.2	21	CO2 **
30	43	6.1	20	3.3	6.6	33	RSP
Mean	22	5.2	23	0.9	4.5	22	
Std Dev	14.6	4.5	9.5	1.0	2.1	10	

* Assuming the current Health Canada guideline of 3,500 ppm

** Assuming the current ASHRAE recommended level of 1,000 ppm



Using three different emission rates for formaldehyde covering the range found in the test houses, and weather data for Montreal, the AQ1 model was used to predict the average level of formaldehyde over an entire year in several houses. For example, using the air change rate of House #9, we show the effect of different levels of source strengths. Please refer to Figure 9. At the lowest formaldehyde source strength of 1.9 ml/h (2.2 mg/h), the formaldehyde level remained below 0.02 ppm, below the Health Canada long term exposure limit of 0.05 ppm (60 µg/cu m). Assuming the medium source strength of 4.9 ml/hr in the same house, the model predicts levels at or over 0.05 ppm from May to September and lower for the colder months. Only during the lowest ventilation periods during July and September and at the highest source strength encountered in the 30 houses (8 ml/hr), was the Canada short term exposure limit of 0.1 ppm reached in this house.



4. Ventilation methods tested

Five different simple ventilation systems were operated and their effectiveness at reducing indoor pollutant levels was evaluated. Simultaneous carbon dioxide recordings in various locations were used along with occupancy records to determine the movement and dilution of the gas over time in each house. The systems could be turned on and off to permit evaluating the changes in indoor conditions. The results are presented very briefly in order of decreasing effectiveness from the points of view of efficacy and economy.

1. Outdoor air supply and mixing with indoor air for tempering, combined with return or supply of air in each bedroom plus a bathroom exhaust - Best system from point of view of comfort and economy.
2. Simultaneous outdoor air supply and exhaust (air exchange) and mixing with indoor air for tempering - air drawn from upper hallway, mixed air released into the basement - Good system, but may have to over-ventilate in order to control bedroom pollutant levels.
3. Two-fan recirculating system installed through the main floor: downflow is greater than upflow fan capacity to cause a slight pressurization of the basement. - System does not provide sufficient outdoor air, but helps diluting pollutants and reducing average levels.
4. Replacement exhaust fan of tolerable noise level; operated at high percent on-times. - Indications are that the effectiveness is about half of a balanced system due to lack of mixing of air within the house and short-circuiting of outdoor air through the fan.
5. Turbine Ventilation system - an open chimney system utilizing a 15 cm diameter insulated duct from the ceiling of the upper hallway, through the attic and roof to the outside, topped with a wind-driven revolving head to facilitate air flow under all wind conditions. This system has the reverse flow characteristics required, with minimum flow during the shoulder seasons and maximum flow during the coldest season. It tends to under-ventilate in mild weather and over-ventilate in cold, windy weather.

5. Movement and distribution of pollutants in a house

In houses without an air distribution system, pollutants such as carbon dioxide released by the occupants redistribute themselves by diffusion and by mixing with air in adjacent open areas. Continuous recording of carbon dioxide in several locations of a house can reveal the amount of redistribution which takes place naturally with doors open and closed. We found the method to be quite sensitive, and was capable of detecting the increase in carbon dioxide production by more people in the house, by the use of candles and by the open fermentation of beer and wine. Some of the more salient observations are summarized in this section.

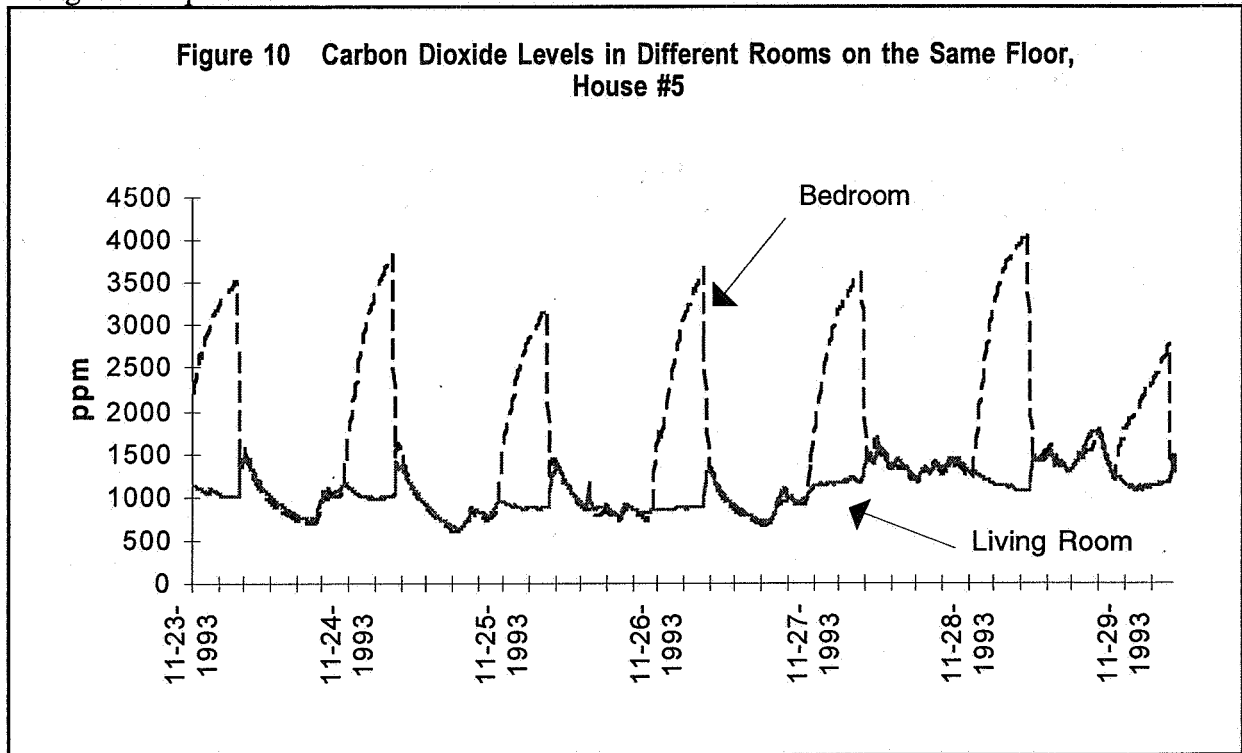
Stratification within a Room

Recordings of carbon dioxide in occupied bedrooms (without an air distribution system) at three levels (floor, 80 cm from the floor and near the ceiling) indicate that there is a small difference in CO₂ levels between the floor and the ceiling while the room was occupied, the readings near the ceiling being higher by about 200 ppm while people were present in the room.

Difference in CO₂ Levels Between Rooms on Same Floor

Simultaneous records of CO₂ levels in the living room and master bedroom in house #3 (on the same level of the building, at table height) which had no air distribution system operating, indicate a difference of 300 to 400 ppm between the occupied and unoccupied room only during the sleeping hours, with the level in the occupied room being higher (bedroom door open). The rest of the time, while the occupants are moving about the house or are away, the room CO₂ levels are within about 50 to 100 ppm.

In House #5, the bedroom door was closed regularly during the night. There was no air distribution system operating in this house. The difference in CO₂ levels between the bedroom and living room increased steadily from the time that the door is closed around 11:00 p.m. until the door is opened around 7:00 a.m., with CO₂ levels rising to 4000 ppm by morning. Once the bedroom door is opened in the morning, it takes approximately 30 minutes for the CO₂ levels in the bedroom and living room to equalize. Please refer to Figure 10. During the day, while the occupants are away and the bedroom door is open, the levels in the bedroom and in the living room equalize.



Difference In CO₂ Levels Between Basement And Main Floor

In houses where new ventilation systems had been installed, the carbon dioxide levels were recorded in the master bedroom, in the living room and in the basement for a period of several days. In House #4, where two fans had been installed through the floor at opposite ends of the house, blowing in opposite directions to mix basement and main floor air, there was an indistinguishable difference in carbon dioxide levels among the bedroom, living room and basement while the equipment was operating (third system described in Section 4)

In House #5, a new ventilation and air mixing system was ducted to each bedroom and to the hallway (first system described in Section 4). In this house, the occupants normally sleep with the bedroom door closed. While the system was operating, there was a small difference of between 100 and 200 ppm in the levels of CO₂ recorded in the bedroom, living room and basement. The occupants noted a marked improvement in air quality compared to the system being off and the bedroom door being closed, when CO₂ levels regularly rose above 4,000 ppm.

In house #8, where a balanced fresh air supply and exhaust system normally operates (second system in Section 4), there are virtually no differences in levels of CO₂ in the basement, bedroom and living room. With the ventilation system off, and the door to the basement in the

normal "closed" position, the average CO₂ level in the basement was always lower than the levels on the main floor. The difference in CO₂ levels between upper level and basement ranged between 300 and 600 ppm. The difference in levels between the bedroom and living room were indistinguishable.

6. Conclusions

1. During the heating season, total ventilation (natural plus mechanical) is inadequate to maintain indoor pollutant levels below the recommended limits during mild weather in the majority of houses tested. During the coldest part of the season, total ventilation is adequate in about half of the houses. The main determining factors are pollutant source strength and building ventilation (operation of fire-places, exhaust equipment and window openings).
2. The main pollutants which determine the minimum ventilation required are formaldehyde, RSP's and CO₂ (when the maximum desired level of CO₂ is 1000 ppm). Since there are no guidelines for maximum TVOC's, no conclusions can be drawn from the observed source strengths of these contaminants.
3. The correlation between the measured air tightness of houses (ACH₅₀) and indoor air quality is very poor.
4. Carbon dioxide levels are directly dependent on the person-hours of occupancy, as well as on the use of candles. The local levels of CO₂ and humidity become elevated when the door of an occupied room is closed in cases where there is no air distribution system.
5. Modeling predicts that the worst indoor air quality will occur in spring through to fall. Air sealing and controlled ventilation together present a significant opportunity for winter peak demand reduction with a small increase in demand the rest of the year.
6. The stratification of carbon dioxide in occupied rooms demonstrates that there is value in mixing air within a building mechanically to improve air quality in the occupied areas.

Acknowledgments

This report is the compilation of an enormous amount of work carried out by the technical staff at LTEE, by the volunteers and their families who allowed us into their homes and faithfully recorded the needed information, and by other staff responsible for the field work and laboratory testing. We are grateful to CMHC for providing partial financial support of this study, and for their technical contribution to the AQ1 model study.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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RESIDENTIAL VENTILATION AND ENERGY CHARACTERISTICS

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SYNOPSIS

The role of ventilation in the housing stock is to provide fresh air and to dilute internally-generated pollutants in order to assure adequate indoor air quality. Energy is required to provide this ventilation service, either directly for moving the air or indirectly for conditioning the outdoor air for thermal comfort. Different kinds of ventilation systems have different energy requirements. Existing dwellings in the United States are ventilated primarily through leaks in the building shell (i.e., infiltration) rather than by mechanical ventilation systems. This report ascertains, from best available data, the energy liability associated with providing the current levels of ventilation and to estimate the energy savings or penalties associated with tightening or loosening the building envelope while still providing ventilation for adequate indoor air quality. Various ASHRAE Standards (e.g., 62, 119, and 136) are used to determine acceptable ventilation levels and energy requirements. Building characteristics, energy use, and building tightness data are combined to estimate both the energy liabilities of ventilation and its dependence on building stock characteristics. The average annual ventilation energy use for a typical dwelling is about 61 GJ (roughly 50% of total space conditioning energy usage); the cost-effective savings potential is about 38 GJ. The national cost savings potential, by tightening the houses to the ASHRAE Standard 119 levels while still providing adequate ventilation through infiltration or mechanical ventilation, is \$2.4 billion. The associated total annual ventilation energy use for the residential stock is about 4.5 EJ (ExaJoules).

LIST OF SYMBOLS

AFUE	gas furnace annual fuel utilization efficiency [-]
COP	air conditioner coefficient of performance [-]
C_p	heat capacity of air [1.022 kJ/kg-°K]
E	annual or seasonal energy load [kJ]
$Elec_{ahu}$	Electrical consumption of air handling unit [% of cooling energy]
$Elec_{comp}$	Electrical consumption of air conditioner compressor [% of cooling energy]
$Elec_{ffan}$	Electrical consumption of furnace fan [% of heating energy]
FH%	percent of heating load met through free heat (solar and internal gains) [%]
HI	inside enthalpy [kJ/kg]
HO	outside enthalpy [kJ/kg]
IDD	infiltration degree days [°C-day]
N	number of hours [h]
NL	normalized leakage area [-]
Q	heat flow/ load[kJ]
T	temperature [°C]
V	ventilation air flow rate [m ³ /s]
ρ	density of air [1.2 kg/m ³]
[h]	indicates hourly value

INTRODUCTION

Infiltration and ventilation in dwellings is conventionally believed to account for 1/3 to 1/2 of the space conditioning energy, although there is not a great deal of measurement data or analysis to substantiate this assumption. As energy conservation improvements to the thermal envelope continue, the fraction of energy consumed by the conditioning of air may increase. Air-tightening programs, while decreasing energy requirements, have the tendency to decrease ventilation and its associated energy penalty at the possible expense of adequate indoor air quality.

In evaluating energy efficiency opportunities, it is important to put into perspective the energy and indoor air quality liabilities associated with residential ventilation. The purpose of this report is to use existing data to estimate these liabilities in the current U.S. housing stock as well as scenarios based on energy conservation and ventilation strategies.

Because of the lack of direct measurements, we cannot approach this as a direct data analysis task. Rather, we approach this objective as a simplified modeling task using the existing sources of data as inputs to the model. The LBL infiltration model and its derivatives will be used as the basis for the calculation.

EVALUATION CRITERIA

In this report we estimate and evaluate, using various ASHRAE standards, ventilation rates, envelope tightness and energy consumption of the stock and some potential alternatives. Various ASHRAE Standards are used to assist us. ASHRAE Standard 119-1988² classifies the envelope tightness of buildings and sets maximum leakage levels based on energy considerations and we use this standard to evaluate the tightness of the housing stock.

ASHRAE Standard 62-1989¹ sets minimum ventilation rates for providing acceptable air quality in all kinds of buildings. For residential buildings the standard specifies 0.35 air changes per hour (ACH), but not less than 7.5 l/s per person. Unfortunately, while the values for residential ventilation are explicit in Standard 62, the interpretation of these values was left vague. The most severe interpretation might be to assume that each room had a minimum of 0.35 air changes at all times; this interpretation would mandate a continuously operating balanced mechanical ventilation system. The most liberal interpretation would only require that the building have the capacity for providing an average of 0.35 ACH; virtually all residential buildings would meet this criterion by having openable windows. The former solution gives no credit to infiltration or natural ventilation, while the latter assumes that occupants are good determinants of indoor air quality and that windows can be opened at any time or weather and in any amount.

Our approach is more moderate: to assume that infiltration contributions can be used to provide ventilation, but that the contribution of natural ventilation will be limited to milder weather conditions and that any whole-house mechanical ventilation system will be sized to meet the 0.35 air change criteria and is run continuously. Using an approach similar to ASHRAE Standard 136-1993³ we can estimate the combined contributions of envelope leakage and other ventilation systems towards meeting Standard 62.

MODELING METHODS AND DATA SOURCES

The modeling methods used in this report have been reported earlier in a preliminary version of this analysis⁴ and are similar to ones used in the general analysis of "blower door" data.³

Variations and additions to the previous modeling methods are described in our corresponding LBNL report.²

Putting available data sources together we can determine for each county the number of houses (from the U.S. Census), the type and sizes of houses (from the Residential Energy Consumption Survey, RECS⁵), the leakage properties (from the LBL Leakage Database¹) and the representative weather conditions. From the analysis of this data, data average and aggregate quantities are developed for the nation as a whole. Our use of the Census and RECS data is identical to that used in our previous analysis.⁴ Based on the RECS data, we have defined 32 different types (or configurations) of houses for use in this analysis: old vs. new (using 1980 as a dividing point); single-story vs. multistory; poor condition vs. good condition; duct systems vs. none; and floor leakage vs. no floor leakage. An expanded U.S. leakage database³ is used in this analysis to develop weighted average leakage values for each climate and house combination (7,680 possible combinations).

Our analysis calculates both heating and cooling loads separately. For heating we use a regional estimation of percent of free heating energy, due to solar and internal gains,⁶ to reduce the heating energy impact. On the cooling side, we only account for cooling load for those fraction of houses having central air conditioning and only when the outdoor temperature and humidity are outside the comfort zone, presuming that ventilative cooling (i.e., natural ventilation) will be used to provide comfort otherwise.

CHARACTERISTICS OF CURRENT STOCK

The housing stock represented by our datasets contains a negligible number of dwellings using whole-house ventilation systems. The task of characterizing the ventilation-related aspects of the stock then becomes one of characterizing the infiltration. We first analyze the leakage data and then use it to estimate ventilation and energy issues.

Envelope Leakage

Using our datasets and the approach discussed in a previous paper³ we can estimate the average normalized leakage (NL) for each county in the U.S., with an average U.S. value of NL=1.2 (std. dev. = 0.34).

Leakage measurements demonstrate a huge variation across house type and age. The statistical distributions are quite wide and do not allow predictions to be made for any single house, but the average values are reasonably representative and can indicate trends. Because the leakage values are the heart of infiltration calculations, this conclusion follows for them as well. While this level of tightness allows for uncontrolled natural ventilation, it corresponds to much higher (looser) levels than that suggested by the ASHRAE tightness standard (119) and contributes to higher, uncontrolled infiltration-related space conditioning loads. Only 15% of the housing stock is tight enough to meet the tightness standard for its given climate. Houses in the milder climates, such as the West Coast, South East and South Central portions of the country, are more apt to meet the tightness standard while houses in the colder climates do not meet the standard.

The databases do not adequately reflect values appropriate for the newest construction which are, in general, much tighter than reflected in the average values. Our comparison of alternative scenarios, however, will have implications for new construction.

Ventilation Rate

The ventilation rate in the stock is dominated by infiltration due to envelope leakage and is calculated from the leakage distribution and the weather using the LBL infiltration model. The concern in this section is only with ventilation rates for providing acceptable indoor air quality and not for energy calculations. Thus we use the effective air change rate which is the constant air change rate that would provide the same pollutant dilution as the actual (time-varying) air change rate.

Although our analysis incorporates the effects of kitchen and bath exhaust fans, these have a negligibly small impact. Our analysis also allows for the use of natural ventilation during mild weather conditions. We estimate the average effective air change rate is 1.09 ACH for the U.S. as a whole and that approximately 95% of current stock meets the intent of ASHRAE Standard 62.

Energy Impacts

The energy impacts associated with the such high infiltration rates are relatively large. We estimate that the heating load attributable to infiltration and ventilation in the current stock is 3.4 EJ and the cooling load is 0.8 EJ. Electrical energy required for parasitics (furnace and air conditioner circulation fans) attributable to infiltration and ventilation is 0.3 EJ. The northern and eastern climates (Mid Atlantic, East North Central, West North Central and South Atlantic) have the highest ventilation-related energy loads, ranging from 0.6 to 0.9 EJ per region. The South Atlantic and West South Central regions (more humid regions) have the highest ventilation cooling-related energy loads, ranging from 0.2 to 0.4 EJ.

Using our air leakage and other databases, we estimate that the national annual cost to provide this much ventilation is \$6 billion/year. The average annual cost per house would thus be \$820/year, with costs ranging from \$50/yr to \$7,000/yr per house. Higher annual costs correspond to areas with colder or more humid climates as well as areas with higher local energy rates.

As mentioned earlier we are assuming a standard set of behavior for all our scenarios: houses are occupied and conditioned full time; therefore, there is no allowance for energy saving strategies such as "set back." This assumption is likely to slightly overstate the energy usage in all our analyses. We also assume that people will use their windows only when it is comfortable outdoors.

ALTERNATIVE SCENARIOS

Although it appears that the vast majority of the U.S. has sufficient residential ventilation, the high cost associated with it suggests that there may be cost effective ways to reduce the infiltration rate and, if necessary, consider mechanical ventilation to meet ASHRAE Standard 62. We shall consider three different scenarios: the "*Base Case*" scenario, the "*ASHRAE*" scenario and the "*Scandinavian*" scenario. For each scenario the most cost-effective means to meet our interpretation of ASHRAE Standard 62 will be found assuming different tightness levels and corresponding infiltration contributions.

The **Base Case** scenario is very similar to the existing stock. But in order to fairly compare other alternatives, the less than 5% under-ventilated stock is modified. In the **ASHRAE** scenario the goal is to also meet ASHRAE airtightness standard 119. The envelope will be tightened as needed to meet Standard 119 and then if required, mechanical ventilation will be

supplied. The **Scandinavian** scenario is similar except that the tightness level will be increased by approximately a factor of two.

We consider two mechanical ventilation systems: simple exhaust and heat recovery ventilation. The simple **exhaust** system assumes that a continuously operating exhaust fan will extract air from the house at all times at a rate of 0.35 air changes per hour. Although various heat recovery strategies such as dynamic insulation or heat pumps are possible, we assume no heat recovery from this system. The Heat Recovery Ventilator (**HRV**) is a balanced air-to-air heat exchanger also sized to provide 0.35 ACH at all times. The HRV recovers some of the energy of the air passing through it, and is modeled with an annual recovery efficiency of 70%. Although other types of mechanical ventilation systems could be considered, these two are the most representative and the only ones we will analyze.

The Base Case

In the Base Case we wanted to find the minimal change that would provide adequate ventilation. As such, we allowed for some *loosening* of the envelope as an option. For the less than 5% of the houses that did not have sufficient ventilation from infiltration, we ran an economic optimization to determine which of our three options (loosen the envelope, exhaust-only ventilation and heat-recovery ventilation) would be more cost effective. Of the stock houses, ventilation systems are necessary in less than four percent of the houses (exhaust fans [1.9%], heat recovery ventilators [1.9%]). Essentially, the base case has no mechanical ventilation. The national average effective air change rates in the base case scenario are essentially the same as that for the stock. The heating and cooling loads increase slightly over that of the stock characterization by loosening the envelope or adding mechanical ventilation. The national annual cost to provide this ventilation is essentially the same as that for the stock.

The “ASHRAE” Scenario

For this scenario we looked at the housing stock and tightened any envelopes necessary to meet ASHRAE Standard 119 and then analyzed the modified stock to determine which houses no longer met ASHRAE Standard 62. Tightening the houses without any mechanical ventilation would reduce the energy cost by almost a factor of four, but some of that gain must be “given back” to provide adequate ventilation. For those 51% of the houses that did not have sufficient ventilation from infiltration we ran an economic optimization to determine which of our two mechanical ventilation options would be more cost effective. (Loosening was not, of course, an option.)

The effective air change rates for the ASHRAE scenario range from 0.35 to 1.18 ACH, with a national average of 0.52 ACH. Census division averages range from 0.48 to 0.59 ACH. The relatively small range is due to the fact that the variation in infiltration has been reduced through tightening and that mechanical ventilation is necessary in more of the housing stock. These air change rates are all higher than the 0.35 ACH minimum due to the fact that we are assuming that the mechanical ventilation system is on continuously. While it is quite likely that the majority of users would not operate these systems at all times, we have used this assumption to avoid overstating the savings associated with the alternative scenarios.

The total energy load for the U.S. for the ASHRAE scenario is about 1.8 EJ. The national annual cost is \$3.6 Billion, a reduction of \$2.4 Billion over that of the base case. The annualized cost of ventilation is \$490/yr for the average house, ranging from \$20/yr to \$2,200/yr per house. The annualized cost reduction achieved is not as large as the energy reduction due to the costs associated with purchasing and operating the mechanical ventilation system. Our

annualized cost calculations take into account these costs but do not incorporate any costs associated with tightening.

Of the 51% of the houses that need mechanical ventilation in the ASHRAE scenario houses, exhaust fans represent 22% and heat recovery ventilators, 29%. The optimal system type varies with house type and fuel costs, but more importantly with climate; the need for mechanical systems is quite minimal on the Pacific Coast but quite significant in the more extreme climates. HRVs are cost-effective in some of the more humid or extreme climates. For the remainder of the country, the general trend is that exhaust fans are used in the frost belt but infiltration alone is used in the sun belt.

The “Scandinavian” Scenario

This scenario is modeled after the northern European shift towards tighter building envelopes and a small amount of operable air inlets. The origin of this trend was in the Swedish standard mandating no more than 3 air changes of envelope leakage at 50 Pascals of depressurization. We have adapted this approach to U.S. climates and our methods, leading to a requirement of a factor of two tighter than the ASHRAE Case. Operable inlets are assumed to be used, when necessary, to bring the leakage to a minimum of $NL=14$ (Standard 119 Class B). As with the ASHRAE case we assume that any mechanical ventilation system is running and that the operable inlets are open.

Ventilation systems are needed in 95% of the houses (exhaust fans [44%], heat recovery ventilators [51%]). The corresponding average air change rates are quite similar to the ASHRAE case, but with smaller regional variation. The optimal system configuration uses 1.6 EJ and has a national annual operating cost of \$4 billion (a reduction of \$2 billion over the stock characterization and over the base case). The annualized cost is approximately \$550/yr for the average house, ranging from \$45/yr to \$1776/yr per house.

The only areas that have a significant amount of infiltration-only systems are the Southern California region and, to some extent, the West Texas / Southern New Mexico region. For the remainder of the country, exhaust-only systems and heat recovery ventilators are favored. Exhaust-only ventilation systems are more predominant than heat recovery ventilators in the Pacific Northwest, Mountain, East South Central and New England regions.

Comparison of Scenarios

The national ventilation energy usage for the various scenarios is summarized in Table 1. Heating, cooling and parasitic energy are essentially the same for the current stock and the base case.

Heating and cooling energy usage decreases from that of the base case for the ASHRAE and Scandinavian scenarios (65% and 72%, respectively) while parasitic energy requirements increase (6% and 29%, respectively). The total ventilation energy usage decreases 2.7 EJ (60%) for the ASHRAE case and 2.9 EJ (65%) for the Scandinavian case.

We can compare the ASHRAE and Scandinavian scenarios to the base case to attempt to determine cost effective levels. Since all of the costs related to the mechanical systems are included, savings represent the income stream available to pay for the required tightening either as a retrofit or in new construction.

TABLE 1. Energy Consumption (EJ)

	Current Stock	Base Case Scenario	ASHRAE Scenario	Scandinavian Scenario
Heating Energy	3.41	3.43	1.15	0.93
Cooling Energy	0.77	0.78	0.34	0.25
Parasitic Energy	0.29	0.31	0.33	0.40
Total Energy	4.47	4.52	1.82	1.58
Free Heating	1.30	1.31	0.64	0.74
Free Cooling	0	0.01	0.19	0.26
Total Free Heating and Cooling	1.30	1.32	0.83	1.00

For the country as a whole the average cost saving is \$290 per house for the ASHRAE case and \$240 per house for the Scandinavian case. Operating cost savings are higher (\$300 to \$450 per house) in the colder northern and northeastern climates as well as in the hot humid climates. Assuming that, on average, house air-tightening costs \$1,000 per house and that the ventilation system operating cost savings are applied to this effort, a typical homeowner could expect a payback of less than five years for the air-tightening efforts needed for either scenario.

It is interesting to note that neither the ASHRAE nor Scandinavian scenarios are always superior. For most of the country the ASHRAE scenario is more cost-effective; in these areas additional tightening beyond that level is not warranted. In the Northern Plains, New England and parts of the hot humid South, the Scandinavian scenario is more cost-effective. Since this analysis does not include the cost of tightening, it is unlikely that the Scandinavian scenario would be practical as a *retrofit* strategy anywhere in the continental U.S.

DISCUSSION AND SUMMARY

The U.S. housing stock currently has a negligible number of houses using whole-house ventilation systems. Infiltration is the dominant ventilation system. Infiltration is often viewed as a poor ventilation mechanism because the flow paths are diffuse and unknown while the driving mechanism is both unstable and variable over the year. While these qualities do little for those who strive for certainty, they do have some advantages. Averaged over any time longer than a day, infiltration provides a lower limit for the ventilation rate even when no ventilation systems operate. Infiltration rates are the highest during the times of the year when window opening is least desirable. Although infiltration may have a relatively low ventilation efficiency, it is at times the optimal system or, more often, a component of an optimal system.

Stock Characteristics

Typical ventilation rates in the stock average slightly over one air change per hour. Because we allow open windows to contribute towards this value, this number is not indicative of the energy impacts of ventilation, only its ability to dilute pollutants. Nevertheless, this number is higher than is often quoted. The representativeness of the leakage data used to make this calculation is not known. While there is no a priori reason to assume the dataset is biased, it is not

impossible for it to be so. The predicted ventilation rates indicate that meeting ASHRAE Standard 62 will not be difficult for most of the stock.

The data implies that the total energy use for residential ventilation is over 4 EJ annually. This number would represent a significantly larger fraction than is normally attributed to residential ventilation. This fact may indicate that some of the assumptions of the analysis should be tested in subsequent efforts. Key factors that could affect the total include air tightness of the stock, temperature preferences and operating strategies; under-conditioning could be a significant contributor. Another key factor to consider is whether or not there is any heat exchange occurring during the infiltration and exfiltration through envelope leaks.

Stock Optimization

As has been discussed in this and other papers, purposely building a house loose in order to provide sufficient natural ventilation by infiltration alone most often results in high energy bills from excessive infiltration. The challenge of building a house to the exact tightness level to balance energy and ventilation through infiltration is an exacting (or exasperating) activity. Likewise, air-tightening an existing house while still providing sufficient natural ventilation is a challenge.

In undertaking this study, we have examined the trade-offs between tightening the building envelope for energy efficiency while adding supplemental mechanical ventilation when necessary to meet ventilation requirements. For most of the U.S. tightening much below the ASHRAE (119) tightness standard does not afford any additional savings, thus implying that from a life-cycle cost perspective there exists an optimal tightness level.

When tightening the envelope to meet the ASHRAE tightness standard, 51% of the houses need some type of supplemental ventilation system. 95% of the houses need supplemental ventilation systems if the houses are tightened to our version of the Scandinavian standard. The trade-offs are found in the energy savings due to lowered, yet still sufficient, ventilation rates with supplemental ventilation equipment.

For the existing stock, these results can be used to evaluate retrofit measures and to develop programs for determining optimal ventilation systems designed to provide adequate ventilation at the lowest cost. The energy savings over that of the existing stock houses with higher ventilation rates (\$240 to \$290/year including the cost of any required ventilation systems), can be applied directly to the tightening and weatherization efforts of a given house resulting in a least-cost effort and minimal financial impact on the homeowner.

Implications for New Construction

Our results have implications for new construction as well as for retrofit efforts, even though our leakage data under-represents the new construction stock. In new construction, the cost of building tighter is principally that of a learning curve, so that the vast majority of our predicted savings can be realized. By treating the base case as a construction option (i.e. design it to leak) rather than as the current state of affairs, we can evaluate new construction options. The optimal level of tightness will vary by region but, overall, the ASHRAE levels do a good job in specifying that level.

Natural ventilation can be used for a significant fraction of the year in the mild parts of the Pacific and Southwest. Thus our economic optimum is not very sensitive to the tightness level in these areas as long as the appropriate ventilation system (if required) is chosen. Typical cur-

rent construction practices are providing tight enough building envelopes for these climates and the only concern may be insufficient ventilation during those parts of the year when natural ventilation is not appropriate.

In the most extreme climates, tightening beyond the level of the ASHRAE Standard may be warranted in order to better utilize the heat recovery of the HRV, but for most of the country this effect is small. Conversely, this implies that there is not a large economic penalty for over-tightening in the more severe climates, where tightening for thermal comfort reasons may be desirable. The striking difference when moving to the Scandinavian scenario is the change in which systems are optimal: there is a sharp drop in the infiltration-only systems in the sunbelt and the rise of HRV systems in the greater Mississippi Valley area and the Northeast with most of the West moving towards exhaust-only fan systems.

Discussion of Errors

The economic conclusions are, of course, sensitive to the price assumptions and specific scenarios we chose. We did not, for example, consider passive ventilation systems, heat-pump heat recovery systems, or dynamic insulation systems; we did not consider high efficiency or variable flow fans, nor did we consider any of the proposed control strategies. Furthermore, the system and fuel price assumptions are unlikely to be universally applicable. Nevertheless, the results indicate clear trends. More specific analyses may be warranted before making localized policy or program recommendations.

Similarly, we have focused on mean values for the technical quantities rather than their distribution. Representative measurements of infiltration and air leakage are known to have large standard deviations, e.g., as big as their mean value, due to inherent inhomogeneities of such samples. Examination of the tails of these broad distributions would require more detailed data than is available nationwide. For mechanically dominated systems, the variation in envelope properties has a less pronounced influence and the distributions become significantly narrower.

Areas for Further Work

The conclusions of this study have clear national implications. The data indicates regional trends, but the specific policies, pricing, and practices of each region are not included in detail. While it appears that leakage retrofit programs may be cost effective over much of the country, the specifics should be incorporated for each locale.

This analysis covers only single-family buildings. It is tempting to say that we would use the same energy intensity for multifamily buildings, which represent only 14% of the U.S. residential floor area, and scale up our values. Future work should attempt to ascertain the accuracy of such an assumption.

As mentioned earlier, an important need is to extend this work more into new construction by improving the database on newly and recently constructed houses. Anecdotal evidence clearly indicates that much of the new construction is already sufficiently tight enough that infiltration and a reasonable amount of natural ventilation will not provide adequate ventilation. The issues in new construction may be not how to make the envelopes initially tighter, but how to provide cost-effective ventilation and how to maintain system integrity.

Because the stock characterization data indicate a much larger load than is normally attributed to residential ventilation, there is a clear need to determine the cause of this discrepancy. One

possibility is bias in the leakage database. This can be resolved by collection and analysis of additional envelope leakage data. Another cause could be the assumption of no significant infiltration heat recovery. Both of these facets should be investigated further.

The validity of the operating assumptions is an area requiring further investigation, but one that affects more than just ventilation. Scheduling of the set-point and the operation of the HVAC system can have a significant impact on not only how much energy is used but how the energy accounting is done. While we avoided some of this issue by using steady-state indoor conditions (and hence no thermal mass effects), further analysis is needed of representative operating assumptions and their impacts.

ACKNOWLEDGEMENTS

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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17-20 September, 1996**

VENTILATION REQUIREMENTS IN NON-DOMESTIC BUILDING AND ENERGY EFFICIENCY

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SYNOPSIS

The research community as well as the design and construction practice is spending a lot of efforts and investments in developing systems which optimise the energy use for achieving certain specified air flow rates. For example, improvements in efficiencies of 10 % in heat recovery systems would be considered as remarkable.

At present, one observes a tremendous difference in the ventilation requirements in various countries as well as at the European level. Variation of a factor 10 of the ventilation rate requirement can be found in the proposal for European standard CEN prENV 1752 depending on the 'cleanness' of the building.

The present paper develops these problems and makes a comparison with the situation in other areas like thermal comfort and lighting requirements.

1. INTRODUCTION

This paper focuses on the required indoor air quality conditions and related ventilation requirements in buildings. The issue of IAQ is compared with thermal comfort and visual comfort. The observed trends shows very important differences.

2. IMPORTANCE OF VENTILATION IN OVERALL ENERGY DEMAND

It is a fact that the transmission losses through opaque and transparent parts of the building envelope have substantially been reduced by improved thermal insulation of these components. High levels of thermal insulation are common technologies in many countries, especially in Scandinavian countries. Further improvements are under development, e.g. transparent insulating materials,....

With respect to the ventilation losses, the situation is far different :

- whereas in the past, providing sufficient ventilation was not a key element in the design considerations, there now is a growing interest in providing good indoor air quality through adequate ventilation of the occupied spaces;
- there have not been spectacular achievements allowing to reduce level the energy requirements for a similar comfort level as has been achieved in the area of thermal insulation.

As a result, one observes a growing part of the ventilation losses in the overall losses of a building. In modern office buildings ventilation losses are of similar or even greater importance than the transmission losses. Therefore, ventilation related energy consumption is very crucial in the context of energy efficiency.

3. COMPARISON OF VARIOUS COMFORT ASPECTS

In the context of energy efficiency improvement, one often focuses the efforts on improving the technologies needed for meeting a certain demand, e.g. a certain thermal comfort level, a certain level of visual comfort. In some cases, the required comfort level influences substantially the energy needs. In this paragraph, an effort is made to compare the situation for thermal comfort, visual comfort and indoor air quality.

3.1 THERMAL COMFORT

As far as thermal comfort in winter conditions is concerned, only very small variation exist between the various standards with respect to the required comfort conditions in buildings (**Error! Reference source not found.**). For the summer situation, there is substantially more discussion, especially about the acceptable temperature levels during warm periods. The outcome of these discussions is important when designing without active cooling. In case of active cooling, it may also lead to a substantial discussion about the cooling needs. The situation is illustrated in figure 2.

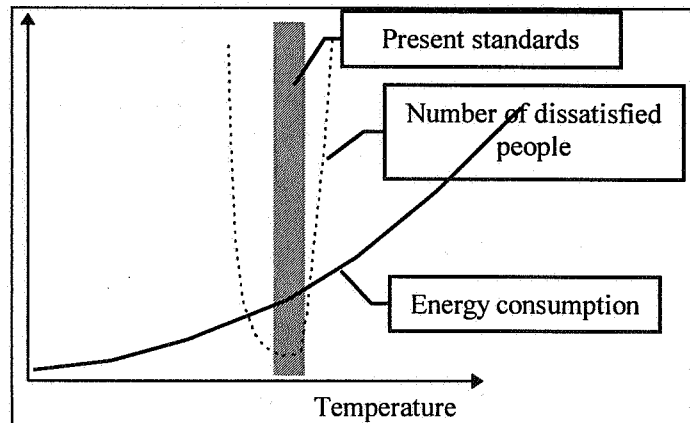


Figure 1- Thermal comfort criteria (winter) and energy

Energy efficiency in the area of thermal comfort in winter conditions should focus on the

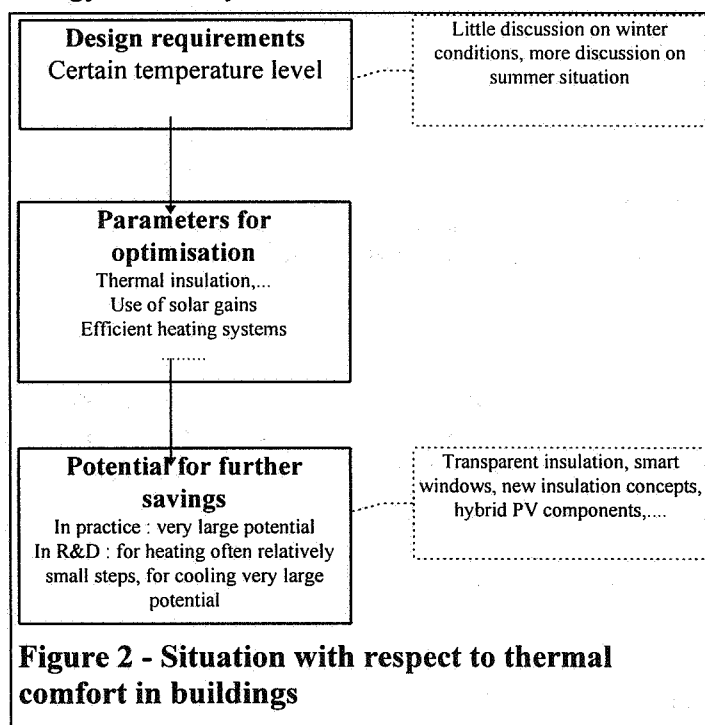


Figure 2 - Situation with respect to thermal comfort in buildings

means for reducing the energy demand: good building design, components with high performances, passive solar design,...

In the case of summer comfort, there is a need for some further research on the comfort needs. But also here, there is a very large potential for improved energy efficiency by a better overall building design, good solar protection,.... In case cooling is needed, innovative low energy cooling systems should surely be promoted. Probability criteria with respect to the thermal comfort in buildings, as e.g. used in the Netherlands, may be a good approach (ref. 3).

3.2 VISUAL COMFORT

With respect to visual comfort, the following elements have to be mentioned :

- in general there is a tendency to decrease the required illuminance levels in comparison with the values used in the seventies and eighties;
- in the case of office buildings, there is also a tendency for a combination of local lighting with general lighting;
- there is an increased interest in natural lighting, allowing to reduce the need for artificial lighting as well as the need and the level of active cooling.

The overall situation is illustrated in figure 3.

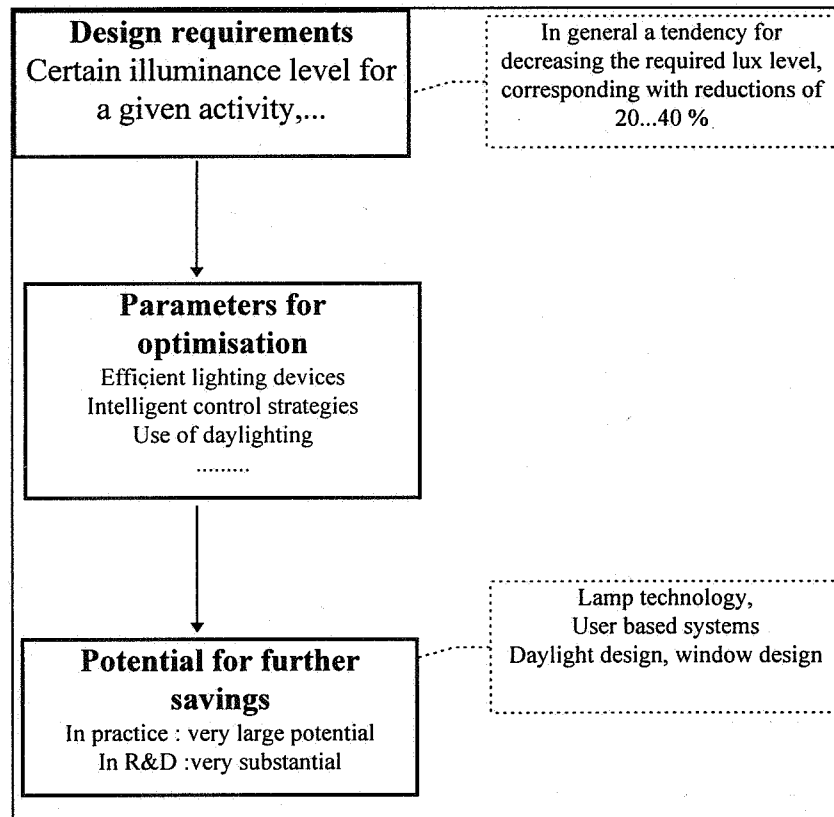


Figure 3 - Situation with respect to visual comfort in buildings

3.3 INDOOR AIR QUALITY

At present, nearly all existing standards require air flow rates which are based on the assumed odour emissions by the occupants. Values in the range of 5..10.... dm³/s/person are quite common. The last years, there is a growing awareness that there are often other important sources of pollution: the building materials, the ventilation system,....

As a result, one observes at present very large differences in the way standards are conceived and, more important, the fact that the variation in requirements becomes extremely large. Some examples are given in section 4.

In any case, the present tendency is towards an increase in the required air flow rates. Such increase is in some cases not of the order of 10...50% but may be a multiplication by 2,3,... Therefore, the impact on the energy demand will be extremely important.

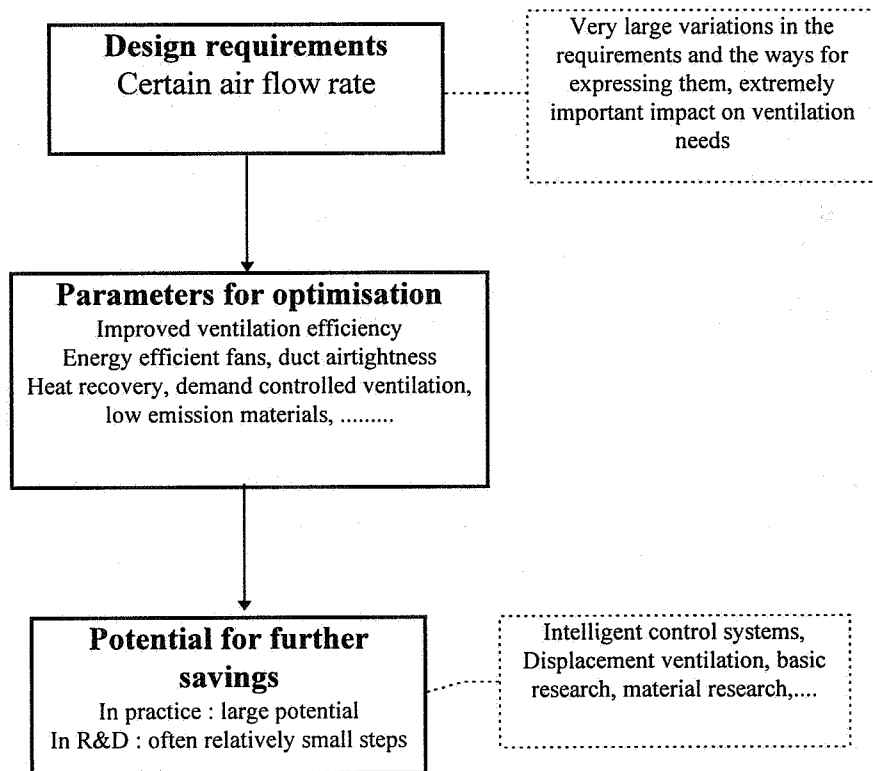


Figure 4 - Situation with respect to indoor air quality and ventilation

Although the ventilation related energy demand is becoming more and more important, it seems that this tendency is not confirmed by the available means for research and development. At present, there is relatively little funding in relation to indoor air quality and ventilation. This may be due to a lack of understanding of the importance of ventilation related energy needs in buildings.

It is important for decision makers to be aware of the fundamental difference between the situation with respect to indoor air quality and e.g. thermal comfort in winter conditions and of the importance of the requirements in the case of ventilation. This is illustrated in figure 4.

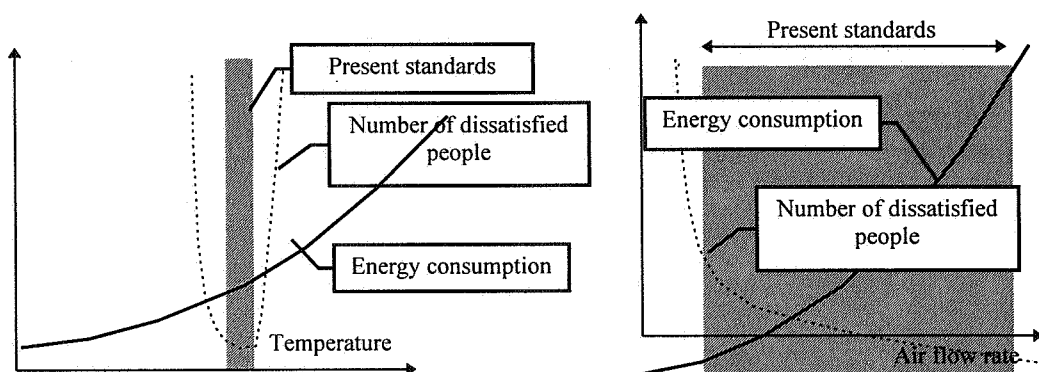


Figure 5 - Comparison between 'thermal comfort versus energy' and 'IAQ versus energy'

3.4 SYNTHESIS

When considering design criteria in the area of thermal, visual and indoor air quality comfort, one observes very large variations in the design criteria in relation to indoor air quality. An average user is in general quite well aware of the differences of the thermal and visual comfort requirement levels on the perception of the space, whereas this is not so much the case with indoor air quality. Given the fact that this variation is very substantial, the impact of the criteria on the energy demand is huge.

4. SOME EXAMPLES OF ONGOING DISCUSSIONS IN THE AREA OF VENTILATION REQUIREMENTS

4.1 CEN

In the context of CEN, Technical Committee 156 'Ventilation in Buildings' is in charge of preparing a standard (CEN prENV 1752) defining the requirements with respect to ventilation in buildings. The issue of indoor air quality is fully included in the discussions but there are substantial differences in the way the participating countries want to deal with it. In any case, there is no major discussion about the fact of considering three levels of air flow rates per person :

- Class A : 10 dm³/s.person
- Class B : 7 dm³/s.person
- Class C : 4 dm³/s.person

The ventilation requirements can substantially increase by including the emission of building materials, HVAC systems,... can substantially increase the ventilation requirements. Table 1 compares requirements on air flow rates for individual offices for the 3 classes and for 2 situations :on the one hand assuming no emission of the building (as is the assumption in many existing standards) and on the other hand assuming an average pollution level as was measured in a range of buildings over the last 5 years. One observes a variation in the requirements with a factor of about 10.

Class	Desirable design conditions	Mean conditions ¹
A	3.3	8.2
B	1.4	5.2
C	0.8	2.6

Figure 6 - Variation in possible requirement levels in the context of CEN standardisation for single offices (dm³/s,m²)

¹These values are based on mean pollution load (from ECA report 1992) and a ventilation efficiency of 0.7 and are under discussion.

4.2 ASHRAE

the ASHRAE Standard 62 'Ventilation for acceptable Indoor Air Quality' , version 1989 is currently under revision.

Some important characteristics of the most recent draft (September 1995):

- the emission of building materials is explicitly included by taking a certain ventilation requirement for the building users (dm^3/s , person) and a certain ventilation requirement for the building ($\text{dm}^3/\text{s}, \text{m}^2$ floor area);
- Smoking is assumed as incompatible with 'acceptable indoor air quality'. In order to allow also rules for rooms where smoking is allowed, an annex S is added in which rules are given for achieving 'Acceptable perceived indoor air quality'. This means that one explicitly assumes that a real acceptable indoor air quality is impossible when smoking is allowed, and that ventilation can only allow to achieve conditions which give the occupants the impression (perception) of an acceptable indoor air quality.

4.3 REGULATIONS IN WALLOON REGION

In the framework of the revision and extension of the thermal insulation requirements for buildings in the Walloon region in Belgium, the government has also decided to require minimum ventilation rates for new buildings. Since this regulation was initiated due to the European CO_2 directive, it is not evident to have very severe requirements with respect to ventilation needs. As a compromise, the class C will be required for all non-domestic buildings, assuming no additional air flow rate for the building emission.

Although these requirements are surely not very high, we believe that the fact of having requirements will allow a very substantial progress in comparison with the present situation. But nevertheless this remark :

'Assuming that the decision makers in the Walloon Region for one or the other reason had been extremely concerned with the IAQ issue, we might have had instead of $0.7 \text{ dm}^3/\text{s}, \text{m}^2$ $7.2 \text{ dm}^3/\text{s}, \text{m}^2$, or a 10 times higher requirement..... Such requirement would have eliminated completely the energy efficiency which can be achieved by good thermal insulation, passive solar design and efficient heating-equipment.'

5. CONCLUSIONS

1. Whereas the thermal comfort requirement in winter time are in a quite small range and not under discussion, there is a tendency for more flexible thermal comfort requirements in summer. More flexible thermal comfort conditions are especially requested by the countries with a relatively hot climate. The extent to what active air-conditioning can be avoided is strongly influenced by the type and level of thermal comfort requirements. As far as visual comfort is concerned, there is a tendency for reducing the illuminance levels.
2. With respect to the ventilation requirements, several countries plan a substantial increase in the required air flow rates
3. The relative importance of the ventilation needs in the overall energy balance is increasing. This is on the one hand due to the continuous reduction of the transmission losses and on the other hand due to the tendency for increasing the ventilation requirements in buildings.
4. The reduction of the emission of pollution from building materials, ventilation systems,... is an important priority for the future.
5. Decision makers (research managers, politicians,...) should become better aware that the human being spends a very large portion of his time indoors, that there are serious concerns about the indoor air quality and that there is a very clear link with the energy efficiency of buildings. They should understand that indoor air quality should receive similar attention as outdoor air quality issues.
6. Decision makers (research managers, politicians,...) should become better aware of the tremendous importance of a correct definition of the IAQ and ventilation needs since the impact on the overall energy demand is enormous. This awareness should then be translated in a substantial increase of the available means for research and development in the area of IAQ and ventilation related issues.
7. ECA is probably one of the best placed organisations for increasing the awareness among decision makers. In that context, a well prepared ECA publication can be of very great importance.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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**IEA ANNEX 27: Evaluation and Demonstration of Domestic Ventilation Systems.
Assessments on noise.**

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Synopsis

The acceptance and appreciation of ventilation systems is mainly determined by the perceived indoor air quality, thermal comfort and noise. Noise in relation to ventilation systems can be divided into three categories:

- outdoor noise (entering the dwelling through ventilation openings, cracks, mechanical supply and exhaust openings etc.);
- noise generated by components of the ventilation system;
- the impact of ventilation systems on sound reduction of partitions (between dwellings, rooms etc.).

Depending on the type of ventilation system, one or more of these aspects are of concern. Noise related to the ventilation system and components, can result in turning off the ventilation system or closing vents etc. This can have a negative influence on ventilation and indoor air quality. In the framework of IEA ANNEX 27 several noise aspects of domestic ventilation systems have been evaluated.

Outdoor noise:

In noise loaded areas the selection and the applicability of different types of ventilation systems are determined by the noise level on the facades. A simplified tool is developed to select ventilation systems as a function of the required noise reduction of the facade, room dimensions and design and construction of the facade.

System noise:

Controlling noise levels caused by ventilation systems is in practice one of the most important factors to contribute to the satisfaction with a ventilation system. Air duct systems in dwellings transport noise generated by fans and aerodynamic noise generated by bends, control valves, grilles etc. ANNEX 27 gives basic formulas to estimate sound power levels of fans and grilles, indication of sizes for silencers and guidelines for design.

Impact on noise reduction of partitions:

The composite sound reduction is the result of different sound channels from one room to the other. One of this sound channels may be the ventilation system (cross-talk). Cross-talk can be brought about through the air duct system, overflow grilles and ventilation openings in partitions, duct transitions etc. ANNEX 27 provides guidelines for sound reductions of partitions and insertion losses for ducts to eliminate the influence of cross-talk.

1. Introduction

Satisfaction with a ventilation system is largely determined by the perceived indoor air quality, thermal comfort (i.e. draft control) and noise aspects. In practice, controlling noise aspects, in particular reducing noise levels, is one of the most important factors that contribute to the satisfaction with a ventilation system. Noise aspects related to ventilation systems can be divided into direct noise and indirect noise. Direct noise is noise generated by the system itself. The system is both the source and the means of transport for noise. Examples are noise generated by fans and by mounting materials of air ducts (structure born noise), and noise generated by control valves and grilles (aerodynamic noise).

Indirect noise includes all noise of which the source is outside the system. In this case the system merely transfers noise which originates outside the system. Examples are traffic noise, noise from industrial plants, catering establishments and aircraft (outdoor noise), and domestic noise (internal noise sources). In general, the following three ventilation principles apply to dwellings:

- natural air supply and natural air exhaust;
- natural air supply and mechanical air exhaust (local and/or central systems);
- mechanical air supply and mechanical air exhaust (balanced ventilation), with or without heat recovery.

Noise related to domestic ventilation systems can be divided into three main areas:

- outdoor noise entering the dwelling through ventilation openings (cracks, grilles, and air supply- and exhaust openings);
- noise generated by the ventilation system of the rooms of the dwelling;
- sound transported within or between dwellings by the ventilation system and/or internal ventilation provisions.

Depending on the type of ventilation system and the strategy, one or more of the three areas indicated in table 1 below are important.

Table 1: Noise importance

	Natural ventilation	Mechanical exhaust	Balanced ventilation
Outdoor noise	x	o	o
System noise	-	x	x
Sound transmission	o	o	x

- irrelevant/not applicable
- o in general of minor importance
- x important

2. Outdoor noise

Requirements and methods of calculation and measurement

Road traffic and other activities like industrial plants can cause a noisy environment. In general, the allowable sound level in rooms is 35 dB(A) (ISO 1996). However, in some countries only 30 dB(A) is allowed (Sweden, Finland). The noise reduction of a facade (G_A) is the difference between the outdoor noise level and the perceived indoor noise level:

$$G_A = L_o - L_i \quad [\text{dB(A)}]$$

Where:

- G_A is the noise reduction of a facade
- L_o is the outdoor noise level
- L_i is the indoor noise level.

The resulting sound reduction is determined by the overall sound reduction of the facade (R_A), taking also into account the noise transfer through ventilation openings and joints and cracks in the construction, and the (acoustic) properties of the room itself:

$$G_A = R_A + 10 \log A/S - 3 \quad [\text{dB(A)}]$$

Where:

- R_A is the overall sound reduction of the facade in dB(A)
- A is the room absorption of the room in m^2 sabine
- S is the total surface area of the facade in m^2
- 3 is the correction for direct sound field to diffuse sound field

Ventilation and infiltration through the building envelope is taken into account in the overall sound reduction R_A . This includes:

- joints and cracks around windows and doors (uncontrolled ventilation);
- ventilation openings in the facades (grilles and windows ajar);
- air ducts in facades and the roof, with or without grilles.

Noise reduction of facades

In noise loaded areas the selection and the applicability of different types of ventilation systems is determined by the noise level on the facade. Dwellings with facades containing windows with single glazing or standard double glazing without weather-stripping achieve a noise reduction of approximately 20 dB(A). With the windows in the ventilation position (ajar), a noise reduction of approximately 15 dB(A) can be achieved. If outdoor noise levels at the facade exceed 50 to 55 dB(A), natural supply systems will require special acoustic measures, particularly with regard to the ventilation system. The transfer of noise through facades takes the following paths:

- closed facade areas (brickwork, panels);
- glass;
- ventilation openings;
- joints and cracks.

Simplified tool

As to determine the most efficient ventilation system in a noisy environment a number of choice matrices are developed, presenting the applicability of ventilation systems with respect to the required noise reduction level, the percentage minor soundproofed parts of the facade and the dimension of the ventilation openings. These matrices are a simplified tool to select the most appropriate ventilation system for a room, if the required noise reduction of the facade, the dimension of the ventilation opening, and the percentage of minor soundproofed parts (such as glass, light weighted panels) in the facade are known. An example of the matrices is shown in figure 1.

The applicability of the ventilation systems is represented in the matrices, as follows:

- system not applicable;
- 0 system applicable if combined with excellent sound proofed measures (high quality acoustic glazing, double weather-stripping);
- + system applicable if combined with normal sound proofed measures (normal acoustic glazing, good single weather-stripping);
- ++ system applicable without extra sound proofed measures (standard double glazing, weather-stripping).

$S_{\text{facade}} = 15 - 25 \text{ m}^2$

Ventilation system		$G_{a,c} = 20 \text{ dB(A)}$														
		Percentage of inferior soundproofing constructions in the facades exosed to traffis noise [%]														
		0					< 50					≥ 50				
		A netto [cm ²]					A netto [cm ²]					A netto [cm ²]				
		0-50	50-100	100-200	200-400	≥ 400	0-50	50-100	100-200	200-400	≥ 400	0-50	50-100	100-200	200-400	≥ 400
Natural supply	Ventilation opening without soundproofing: $R_a = -5 \text{ dB(A)}$	++	++	++	++	-	++	++	+	+	-	++	++	++	0	-
	Soundproofed ventilation opening: $R_a = 0 \text{ dB(A)}$	++	++	++	++	++	++	++	++	++	+	++	++	++	++	++
	Soundproofed ventilation opening: $R_a = 10 \text{ dB(A)}$	++	++	++	++	++	++	++	++	++	++	++	++	++	++	++
	Soundproofed ventilation opening: $R_a = 15 \text{ dB(A)}$	++	++	++	++	++	++	++	++	++	++	++	++	++	++	++
Mechanical supply and exhaust		++	++	++	++	++	++	++	++	++	++	++	++	++	++	++

Ventilation system		$G_{a,c} = 25 \text{ dB(A)}$														
		Percentage of inferior soundproofing constructions in the facades exosed to traffis noise [%]														
		0					< 50					≥ 50				
		A netto [cm ²]					A netto [cm ²]					A netto [cm ²]				
		0-50	50-100	100-200	200-400	≥ 400	0-50	50-100	100-200	200-400	≥ 400	0-50	50-100	100-200	200-400	≥ 400
Natural supply	Ventilation opening without soundproofing: $R_a = -5 \text{ dB(A)}$	++	++	-	-	-	++	-	-	-	-	++	0	-	-	-
	Soundproofed ventilation opening: $R_a = 0 \text{ dB(A)}$	++	++	++	++	-	++	++	+	0/-	-	++	++	+	0	-
	Soundproofed ventilation opening: $R_a = 10 \text{ dB(A)}$	++	++	++	++	++	++	++	++	++	++	++	++	++	++	++
	Soundproofed ventilation opening: $R_a = 15 \text{ dB(A)}$	++	++	++	++	++	++	++	++	++	++	++	++	++	++	++
Mechanical supply and exhaust		++	++	++	++	++	++	++	++	++	++	++	++	++	++	++

Ventilation system		$G_{a,c} = 30 \text{ dB(A)}$														
		Percentage of inferior soundproofing constructions in the facades exosed to traffis noise [%]														
		0					< 50					≥ 50				
		A netto [cm ²]					A netto [cm ²]					A netto [cm ²]				
		0-50	50-100	100-200	200-400	≥ 400	0-50	50-100	100-200	200-400	≥ 400	0-50	50-100	100-200	200-400	≥ 400
Natural supply	Ventilation opening without soundproofing: $R_a = -5 \text{ dB(A)}$	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	Soundproofed ventilation opening: $R_a = 0 \text{ dB(A)}$	++	++	-	-	-	+	-	-	-	-	0	-	-	-	-
	Soundproofed ventilation opening: $R_a = 10 \text{ dB(A)}$	++	++	++	++	++	+	+	+	+	+	+	+	+	0	-
	Soundproofed ventilation opening: $R_a = 15 \text{ dB(A)}$	++	++	++	++	++	+	+	+	+	+	+	+	+	+	0
Mechanical supply and exhaust		++	++	++	++	++	+	+	+	+	+	+	+	+	+	+

Ventilation system		$G_{a,c} = 35 \text{ dB(A)}$														
		Percentage of inferior soundproofing constructions in the facades exosed to traffis noise [%]														
		0					< 50					≥ 50				
		A netto [cm ²]					A netto [cm ²]					A netto [cm ²]				
		0-50	50-100	100-200	200-400	≥ 400	0-50	50-100	100-200	200-400	≥ 400	0-50	50-100	100-200	200-400	≥ 400
Natural supply	Ventilation opening without soundproofing: $R_a = -5 \text{ dB(A)}$	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	Soundproofed ventilation opening: $R_a = 0 \text{ dB(A)}$	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
	Soundproofed ventilation opening: $R_a = 10 \text{ dB(A)}$	++	++	-	-	-	0	0	-	-	-	-	-	-	-	-
	Soundproofed ventilation opening: $R_a = 15 \text{ dB(A)}$	++	++	++	++	++	0	0	0	0	-	0	0	-	-	-
Mechanical supply and exhaust		++	++	++	++	++	0	0	0	0	0	0	0	0	0	0

Figure 1: Choice matrices which present the applicability of ventilation systems in a noisy environment

3. Noise generated by the ventilation system

Requirements and methods of calculation

The air duct system inside dwellings is responsible for the transport of noise generated by the fan and aerodynamic noise generated by bends, control valves and devices. The maximum indoor noise level criteria in most countries, with respect to noise generated by the ventilation system in rooms, is 30 dB(A). The sound power level (L_w) at the optimal operating point of the fan can be approximated by means of the following simplified formula:

$$L_w = L_{ws} + 10 \log q_v + 20 \log \Delta p \quad [\text{dB}]$$

Where:

$$L_{ws} = 1 \pm 4 \text{ dB}$$

q_v is the flow capacity in m^3/h

Δp is the total pressure difference across the fans (Pa)

In general, fans for single family dwellings have a A-weighted sound power level $L_{w(A)}$ of 60 to 65 dB(A).

Sound reducing measures inside air ducts

The noise generated by fans is propagated through the air duct system. If no special noise soundproofing measures are taken, the resulting noise levels can be considerably higher than 30 dB(A). Without special soundproofing measures, internal noise levels of up to 45 dB(A) in rooms can be expected. This requires the use of soundproofing materials inside the air duct system. Soundproofing materials in the supply system should ideally be placed immediately after the unit, but always before the first branching of the duct. Soundproofing provisions for domestic ventilation systems usually consist of soundproofing (flexible) tubes (silencers). Silencers are relatively cheap and easy to integrate in an air duct system. They demand hardly any extra space. A silencer consist of a perforated inner duct, a casing of mineral wool and an outer duct. The ducts may be plastic or metal and are often flexible to a certain extent. The noise attenuation of these materials is poor at low frequencies. Silencers made of thin foil provide better soundproofing at low frequencies, as a result of the fact that low-frequency noise is emitted. This can be a problem in open setups near rooms. Another disadvantage is that the soundproofing effects decrease proportionally to the increase of the internal diameter. As alternative to silencers, double walled steel ducts can be used. These insulated ducts consist of a metal spiral seam inner and outer pipe separated by a layer of sound absorbing material. Soundproofing of these ducts can be achieved by using mineral wool as an insulator and perforating the inner duct. This will render the ducts eminently suitable for combined thermal and soundproofing. As a result of the thinness of the casing, the noise insulating value per linear meter is relatively low, in particular at low frequencies. The perforation level should be 13 to 15 %. The required length is at least 2 m for internal diameters up to 80 mm and 4 m for internal diameters up to 125 mm. These ducts can also be encased within concrete floors.

Simplified guideline for dimensions of silencers

Table 2: Indication of dimension for silencers

	Flow capacity q_v (m ³ /h)	Diameter (mm)		Silencer length (mm)
		internal	external	
ducts directly connected to rooms	$q_v < 100$	100	200	1000
	$100 < q_v < 250$	150	250	1000
ducts directly connected to other spaces	$q_v < 100$	100	200	750
	$100 < q_v < 250$	150	250	750

Acoustic properties of terminal devices

The A-weighted sound power level of grilles can be approximated by:

$$L_{W(A)} = -4 + 70\log v + 30\log \xi + 10\log S$$

Where:

- v is the air velocity across the grille in m/s
 ξ is the airflow resistance factor [-].
 S is the surface area of the grille in m².

Supply grilles must be selected on the basis of the nominal sound power levels to meet the required noise levels. Manufacturers have various ways of indicating the acoustic data. They usually provide a manual to help select the grilles on the basis of their acoustic properties.

In general the maximum allowable sound power levels (L_w) of grilles will be:

- bedrooms: approx. 35 dB(A)
- living rooms: - one grille : approx. 35 dB(A);
 - two grilles : approx. 32 dB(A);
 - three grilles : approx. 30 dB(A).

Aerodynamic noise inside ducts

The following points must be observed in order to prevent aerodynamic noise:

- the maximum allowable air velocity in main ducts is 4 m/s and inside branch ducts to grilles 2 m/s;
- the use of round ducts is preferred;
- sharp bends should be avoided and changes of cross-sectional areas of ducts should be smooth;
- the system must be designed in such a way as to require the minimum number control valves; if the use of control valves cannot be avoided, these should be low-noise valves;
- if possible, control valves must be positioned in front of the silencers;
- air velocity inside the silencers must be such that the aerodynamic noise generated, is 10 dB lower than the ventilation noise immediately after the silencer; aerodynamic noise restricts the velocity of the air passing through the silencers to 4 to 5 m/s.

Fan location

Fans should be placed in a vibration-insulated position on the floor. Vibration insulation may be achieved by placing the four corners of the unit on rubber blocks, with a vulcanized plate, and provided with studs or other mounting facility. It is also possible to place the fan on elastic rigid mineral wool plates.

4. Sound transmission in or between dwellings

Requirements and methods of calculation and measurement

With respect to the sound requirements between two rooms, a distinction is made between rooms within the same dwelling on one hand and between one dwelling and another or between a room in a dwelling and a space outside it on the other hand. More stringent requirements apply to constructions which separate two adjacent dwellings.

There are many different ways in which the sound insulation requirements are expressed. In general for most countries the average sound reduction index R_m in the range between 125 and 2000 Hz for constructions between dwellings must be 50 to 52 dB. The sound insulation characteristic of a wall is usually expressed as the sound reduction index R . The sound reduction index R between two rooms can be evaluated from:

$$R = L_{ps} - L_{pr} + 10 \log S/A \quad [\text{dB}]$$

Where:

- L_{ps} is the sound pressure in source room in dB
 L_{pr} is the sound pressure in receiving room in dB
 S is the surface area separating wall in m^2
 A is the room absorption in m^2 sabine.

The sound reduction index R is the result of the different sound channels from one room to the other. This composite sound reduction index R' value is determined by:

$$R' = -10 \log \sum_{j=1}^n \frac{S_j}{S_{\text{tot}}} 10^{-R_j/10} \quad [\text{dB}]$$

Where:

- S_j is the surface area from one composure of the separating wall in m^2
 S_{tot} is the total area of the separating wall in m^2
 R_j is the sound reduction from one composure of the separating wall in dB

One of the sound channels may be a ventilation system. This phenomenon is also called cross-talk. Cross-talk can be defined as the effect that system components have on the integrity of the sound reduction between two rooms. Cross-talk is of particular concern in balanced ventilation systems and in collective air ducts between dwellings (such as central exhaust systems in residential buildings). It can be brought about in the following ways:

- through the air duct system; this phenomenon can take place both between rooms within the same dwelling and between rooms in two different dwellings;
- through the overflow grilles or openings underneath doors; cross-talk may occur in the case where two rooms have such openings near one another;
- through the duct transitions in walls or floors.

Cross-talk

There must not be a decrease of the sound reduction between dwellings as a result of the transfer of noise through shared ventilation ducts. Both natural (PSV) and collective mechanical ventilation duct systems usually require sound proofing provisions.

Such provisions may consist of:

- a silencer in front of each outlet or between outlets;
- a soundproofed grille at each outlet.

If, in dwellings, rooms are directly linked with one another by the supply duct of a balanced ventilation system cross-talk is a major point of concern. The effects of cross-talk in the system on the average sound reduction index is approximately 14 dB. In this case the overall sound reduction index value decreases and additional measures are necessary.

When doors are shortened at the bottom to facilitate air transport inside a dwelling (balanced ventilation and warm air heating), the following factors should be taken into account:

- doors must be shortened by 3 mm for each 10 m³/h of air supplied to or removed from the room.
- if the doors of two adjacent rooms are directly beside one another, the sound reduction index of the separating wall must be higher than the minimum required value in order to compensate for the transfer of noise through the gap underneath the door.

A major point of concern are collective ventilation ducts (also PSV ducts) specially if they connect rooms in adjacent dwellings. Acoustic measures are in most cases necessary to maintain the required noise reduction. The total minimum insertion loss of a duct between two dwellings is given in table 3.

Table 3. Minimum insertion loss of a duct between two dwellings.

Frequency (Hz)	125	250	500	1000	2000
Insertion loss (dB)	0	0	10	15	15

Improperly constructed duct transitions may create acoustic leaks between the air duct and the separating wall. Soundproof connections between the duct and the separating wall are therefore recommended. These can be achieved by using a sealing glue or an airtight sealant.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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Modelling and assessing ventilation efficiency in an imperfectly mixed ventilated air space

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Synopsis

To ensure indoor air quality an efficient ventilation system should provide fresh air in those parts of a room where it is required. To assess whether the ventilation system fulfils the main objective, different definitions of local ventilation efficiency (the local mean age of air, the local ventilation rate, the local purging flow rate and the local air exchange rate) are reported in literature. Tracer gas techniques (step up method, step down method and pulse method) and CFD-models are mostly used in research to identify the 3-D distribution of these ventilation parameters in a ventilated air space.

In our laboratory the 'local volumetric concentration of fresh air flow rate' [$\text{m}^3/\text{s}.\text{m}^3$] was introduced as a new ventilation parameter. A total amount of 90 step up experiments was performed in a laboratory test installation with a volume of 9 m^3 [$3 \times 2 \times 1.5 \text{ m}$] to model the 3-D distribution of this ventilation parameter in relation to different values of the ventilation rate (120-300 m^3/h) and the heat supply (0-400 Watt) as control inputs. In the main part of the paper some results are given of the modelled 3-D distribution of the 'volumetric concentration of fresh air flow rate' in the test installation. This distribution of fresh air flow rate is assessed in relation to the air flow pattern which is visualised by smoke experiments and quantified by image analysis and the Archimedes number.

List of symbols

V	=	total ventilation rate through the test room [m^3/h]
Q	=	heat supplied by the heating element positioned just beneath the air inlet [Watt]
Q_{in}	=	internal heat production in the test room [Watt]
Q_{wall}	=	heat losses through the walls [Watt]
T_o	=	temperature of the incoming air in the test room [$^{\circ}\text{C}$]
T_i	=	temperature in the 'well mixed zone' [$^{\circ}\text{C}$]
V_c	=	part of the total ventilation rate entering the 'well mixed zone' [m^3/s]
Vol_c	=	volume of the 'well mixed zone' [m^3]
v_c	=	volumetric concentration of fresh air flow rate in the 'well mixed zone' [$\text{m}^3/\text{s}.\text{m}^3$]
Ar_c	=	corrected Archimedes number [-]

1. Introduction

The main objective of a ventilation system is to provide fresh air in those parts of a room where it is required. To assess whether the ventilation system fulfils the main objective, different definitions of local ventilation efficiency are reported in literature [1,2]. The most important are the local mean age of air [s], the local ventilation rate [1/s], the local purging flow rate [m³/s] and the local air exchange rate [1/s]. Different techniques have been developed to identify the 3-D distribution of these ventilation parameters in a ventilated air space. Among these techniques tracer gas experiments (step-up method, step-down method and pulse method) are used in most cases. CFD-models are also used, but these models still have many technical and fundamental problems (immense computational power, validation problem, ...) as reported in [3].

2. Objectives

In our laboratory the 'local volumetric concentration of fresh air flow rate' [m³/s.m³] was introduced as a new ventilation parameter [4]. The main objective of this paper is to show that the modelled 3-D distribution of the 'local volumetric concentration of fresh air flow rate' can be assessed in relation to the measured and quantified air flow pattern in the test installation.

3. Method

In a laboratory test installation with a volume of 9 m³ [3×2×1.5m] 90 step up experiments were performed to model the 3-D distribution of this ventilation parameter in relation to different values of the ventilation rate (120-300 m³/h) and the heat supply (0-400 Watt) as control inputs [5].

3.1. Test installation

The laboratory test installation is presented in figure 1. It is a ventilated room (L=3m, W=1.5m and H=2m) with two control inputs: the ventilation rate V (120-300 m³/h) bringing cold air with temperature $T_o=11^{\circ}\text{C}$ in the test room and the heat Q (0-400 Watt) supplied by the heating element (13 in figure 1) positioned just beneath the air inlet. These control inputs determine the position of the incoming air jet in the ventilated room [6]. The other inputs in the test room are the internal heat production Q_{in} (300 Watt) simulating the occupants and the wall losses Q_{wall} which are minimised because of a second envelope surrounding the test installation. The test room is furthermore provided with 36 sensors positioned in a 3-D grid to measure the spatial and time related distribution of temperature. The sensor configuration is shown in figure 2. A more detailed description of the test installation is given in [5].

3.2. Definition of the 'local volumetric concentration of fresh air flow rate'

According to what has been shown in literature [4,7,8,9] and has been measured in the laboratory test installation [6], the air in the ventilated test room is considered to be not perfectly mixed in which there is a 3-D air flow pattern and related gradients in temperature. Although the air in the test room is not perfectly mixed, it is always possible to consider a 3-D **well mixed zone** around a local sensor position in which there is a better mixing and an acceptable gradient of temperature. The acceptable gradient of temperature in the well mixed zone is defined by the application (0.1°C for a micro-ship oven, 2°C for a greenhouse, 3°C for a livestock building,...). In the test room we furthermore consider an air flow pattern which brings only the part V_c [m³/s] of the total ventilation rate V to the considered well mixed zone. Since the volume Vol_c [m³] of the well mixed zone is unknown, the 'local volumetric concentration of fresh air flow rate' is considered as the local ventilation parameter in the well mixed zone and is defined as $v_c = V_c / Vol_c$ [m³/s.m³]. A graphical representation of this concept is shown in figure 3. It will be illustrated in section 4 that the 'local volumetric concentration of fresh air flow rate v_c [m³/s.m³]' in a well mixed zone is varying as a function of the ventilation rate V [m³/h] and the heat supply Q [Watt] as control inputs and the corresponding air flow pattern. The 'local volumetric concentration of fresh air flow rate v_c [m³/s.m³]' can therefore be considered as a useful local ventilation parameter.

3.3. Identification procedure

1. To identify the 'local volumetric concentration of fresh air flow rate v_c [m³/s.m³]' in the considered well mixed zone in relation to the ventilation rate V [m³/h] and the heat supply Q [Watt], 90 step up experiments were performed in the test installation. During these experiments the ventilation rate V was varied step wise from 80 m³/h (900 sec.) to 120, 140, 160, 180, 200, 220, 240, 260, 280 or 300 m³/h (2700 sec.), resulting in an exponential decay of temperature in the considered well mixed zone. The control heat supply Q was set to 0 or 400 Watt. During each experiment the exponential decay of temperature was measured with a time step of 1 second using the local sensor in the considered well mixed zone. In the tables 1 and 2 an overview is given of the 90 performed step up experiments. In figure 4 one of the step up experiments together with the measured decay of temperature in sensor [431] is presented as an example. The position of this sensor can be found in figure 2.
2. Using the monitored values of temperature in the considered well mixed zone during a step up experiment, it becomes possible to identify the 'local volumetric concentration of fresh air flow rate v_c [m³/s.m³]' in the well mixed zone corresponding to the ventilation rate V [m³/h] after the step and the applied heat supply Q [Watt]. Therefore, equation (1) can be used [10]:

$$v_c = \frac{1}{3600 \int_{900} \frac{T_i(t) - T_{i2}}{T_{i1} - T_{i2}} dt} \quad (1)$$

in which: v_c = local volumetric concentration of fresh air flow rate [m³/s.m³]
 $T_i(t)$ = measured temperature in the well mixed zone at time t [°C]
 T_{i1} = temperature in the well mixed zone at time step 900, just before the step on the ventilation rate [°C]
 T_{i2} = temperature in the well mixed zone at time step 3600 at the end of the experiment when steady-state is achieved [°C]

As an example, equation (1) is applied to the measured temperature decay in sensor [431] as presented in figure 4. The equation returns for this particular experiment a v_c -value of 0.0102 m³/s.m³. This means that the 'local volumetric concentration of fresh air flow rate' in the well mixed zone around sensor [431] is 0.0102 m³/s.m³ when the ventilation rate V in the test room is 160 m³/h and there is no control heat supply Q .

3. By applying equation (1) to the 90 performed step up experiments in the test room, the 'local volumetric concentration of fresh air flow rate v_c [m³/s.m³]' in the considered well mixed zone can be identified in relation to the different values of the ventilation rate V (120, 140, 160, 180, 200, 220, 240, 260, 280 and 300 m³/h) and the control heat supply Q (0 and 400 Watt). In figure 5 an example is given of the identified relationship between the 'local volumetric concentration of fresh air flow rate v_c ' in the well mixed zone around sensor [431] and the two control inputs V and Q . The 90 points on the figure represent the 90 performed step up experiments. The fitting curve is a second order polynomial.
4. The identification procedure has been used to identify the $v_c/V, Q$ -relationships for the 36 sensor positions in the test installation. These relationships are given in [5]. From these relationships the 3-D distribution of the 'local volumetric concentration of fresh air flow rate' can be determined, and this for different values of the ventilation rate V (120-300 m³/h) and the heat supply Q (0-400 Watt) as control inputs. It will be shown in the next section that this 3-D distribution of fresh air flow rate can be related to the air flow pattern in the test room.

4. Results

It was hypothesized in the previous section that the 3-D distribution of the 'local volumetric concentration of fresh air flow rate v_c [m³/s.m³]' can be related to the air flow pattern in the ventilated test room. To test this hypothesis the air flow pattern is first visualised and

quantified for different values of the ventilation rate and the control heat supply. After that, the 3-D distribution of fresh air flow rate is related to the air flow pattern.

4.1. Visualisation and quantification of the air flow pattern

To visualise the air flow pattern TiCl_4 -smoke is injected in the incoming air and the air flow pattern is filmed with a camera. The position of the air flow pattern is thereafter quantified by calculating the co-ordinates of the centreline of the air flow pattern [6]. The position of the air flow pattern can also be quantified by calculating the corresponding Archimedes number Ar_c [11]. According to Randall and Battams [11] three categories of air flow patterns can be distinguished as a function of the Archimedes number Ar_c : a horizontal air flow pattern ($\text{Ar}_c < 30$), an unstable air flow pattern ($30 < \text{Ar}_c < 75$) and a falling air flow pattern ($75 < \text{Ar}_c$). These categories of air flow patterns can also be generated in the laboratory test installation.

Nine experiments were performed in the test room to visualise and quantify the air flow pattern for different values of the ventilation rate V [m^3/h] and the heat supply Q [Watt]. An overview of these experiments is given in table 3. Some of the results are presented in the figures 6 and 7. In figure 6 the centreline of the incoming air jet is visualised for ventilation rates of 88, 158, 249 and 308 m^3/h . The corresponding Archimedes numbers are also calculated and presented in this figure. It can be seen from this figure that the three categories of air flow patterns are generated in the test room and that the criterium of Randall and Battams is confirmed. In figure 7 the 2-D centreline of the incoming air jet is visualised for a control heat supply Q of 0 and 400 Watt.

4.2. Physical interpretation of the $v_c/V, Q$ -relationships

In general it can be concluded from the figures 6 and 7 that a higher ventilation rate or control heat supply results in an upward deflection of the air flow pattern. This influence of the ventilation rate and the control heat supply on the position of the air flow pattern is reflected in the $v_c/V, Q$ -relationships of the 36 sensor positions in the test room. The $v_c/V, Q$ -relationship of sensor position [431] is interpreted here as an example. This relationship is given in figure 5. The position of sensor [431] can be found in the figures 2, 6 and 7. It can be seen from the $v_c/V, Q$ -relationship in figure 5 that a higher ventilation rate V [m^3/h] yields a higher v_c -value in the well mixed zone around sensor [431]. This can be explained, because a higher ventilation rate brings more fresh air in the whole ventilated room and results in an upward deflection of the air flow pattern so that more fresh air moves towards the considered well mixed zone around sensor [431]. It can also be seen that a heat supply Q of 400 Watt yields a higher v_c -value in the well mixed zone around the considered sensor. This is again due to the air flow pattern which is found upward because of the heat supply Q .

4.3. Physical interpretation of the 3-D distribution of the 'volumetric concentration of fresh air flow rate v_c '

From the 36 identified $v_c/V, Q$ -relationships in the test installation the 3-D distribution of the 'local volumetric concentration of fresh air flow rate v_c [m³/s.m³]' can be determined by linear interpolation, and this for different values of the ventilation rate (120-300 m³/h) and the heat supply (0-400 Watt). In figure 8 the distribution of the 'local volumetric concentration of fresh air flow rate v_c [m³/s.m³]' is visualised in the front and the rear plane of sensors when the ventilation rate V is 180 m³/h and the heat supply Q is 400 Watt. From these figures it can be seen that most of the fresh air flows in the upper regions of the test installation. This is due to the air flow pattern which is found upward because of the buoyancy force on the air jet caused by the control heat supply Q .

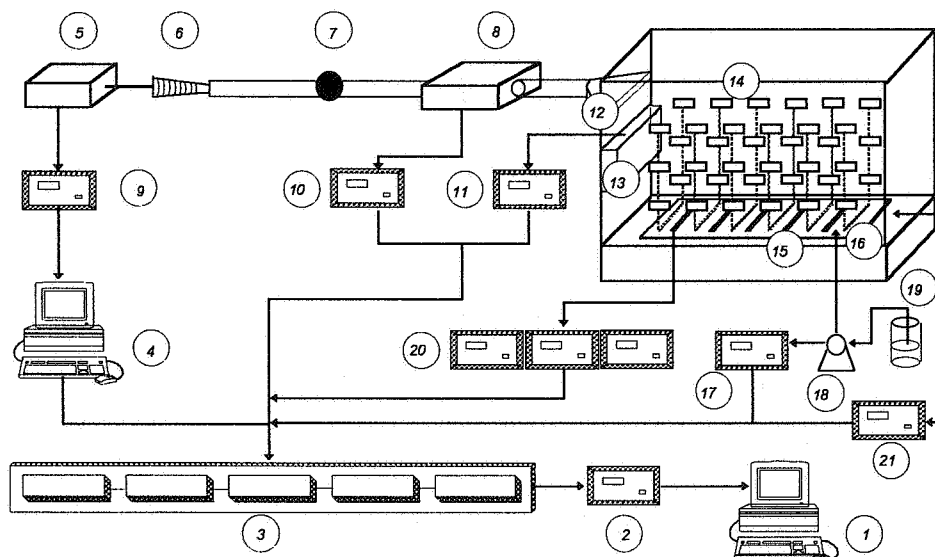
5. Conclusions

The 'local volumetric concentration of fresh air flow rate v_c [m³/s.m³]' has been introduced as a new ventilation parameter. A total amount of 90 step up experiments was performed in a laboratory test installation [3×2×1.5m] to model the 3-D distribution of this ventilation parameter in relation to different values of the ventilation rate (120-300 m³/h) and the heat supply (0-400 Watt) as control inputs. From the results it can be confirmed that the 3-D distribution of the 'volumetric concentration of fresh air flow rate' can be assessed in relation to the air flow pattern which has been visualised by smoke experiments and quantified by image analysis and the Archimedes number Ar_c .

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1. Minicomputer (monitor, floppy disc, to store and visualise the measured data). 2. Parallel-interface for digital and analogue signals. 3. Scan- and measurement unit. 4. Minicomputer (to control and measure the produced air flow rate). 5. Stepmotor to control the position of the cone, used as diaphragm. 6. Cone, used as diaphragm, to produce the desired air flow rate. 7. Centrifugal fan, to generate a ventilating rate. 8. Cooling installation to control the inlet temperature. 9. Differential pressure transducer to measure pressure difference between the test chamber and the envelope. 10. Control- and measurement unit of the cooling installation. 11. Control- and measurement unit of the heating element. 12. Air inlet (slot inlet). 13. Heating element. 14. Three dimensional grid of temperature and humidity sensors. 15. Aluminium semi conductor heat sinks to provide internal heat production. 16. Undeep water reservoir with a streamer containing hot water to generate the internal moisture production. 17. Unit to control and measure the amount of water supplied to the undeep water reservoir. 18. Water pump. 19. Water supply reservoir. 20. Power supplies for internal heat production. 21. Pressure difference measurement used to control the outlet fan.

Figure 1: Schematic representation of the laboratory test installation (L=3m, W=1.5m, H=2m).

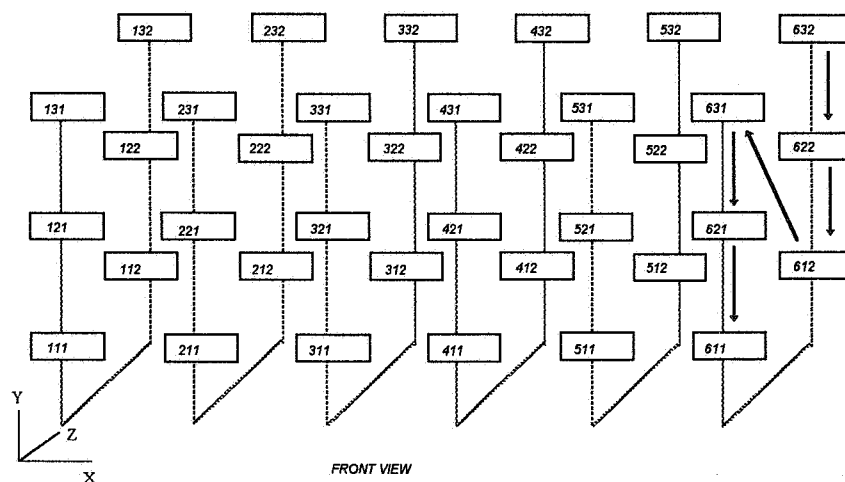


Figure 2: Sensor configuration in the test installation

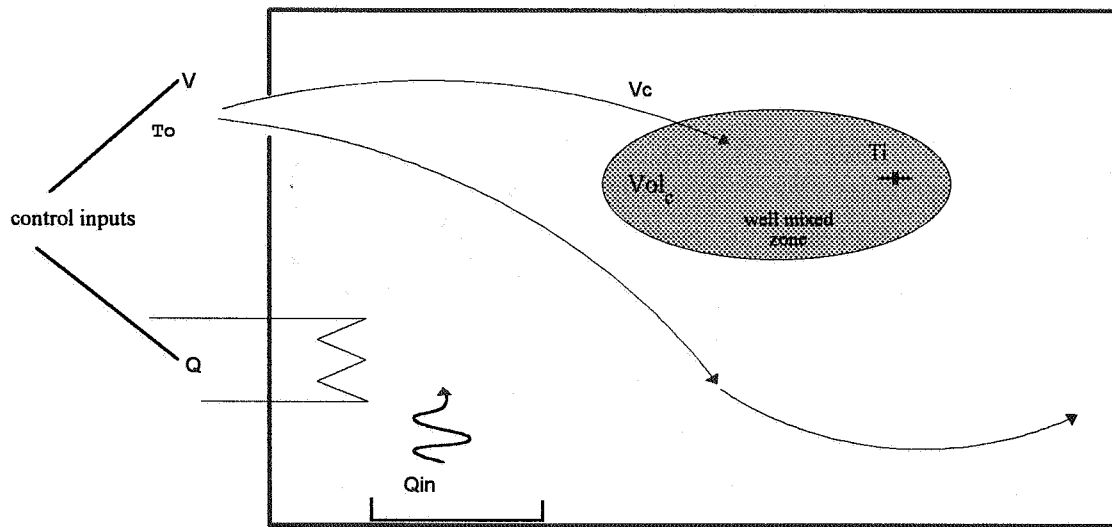


Figure 3: Graphical representation of the concept: the 'local volumetric concentration of fresh air flow rate $v_c = V_c / Vol_c$ '

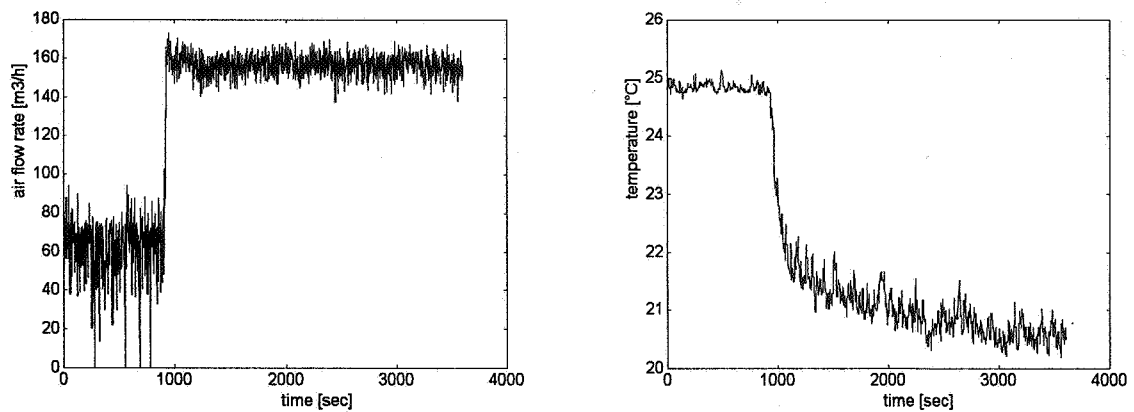


Figure 4 : The ventilation rate V [m3/h] during a step up experiment. The heat supply Q is set to 0 Watt (left). The measured decay of temperature in sensor [431] during the step up experiment (right).

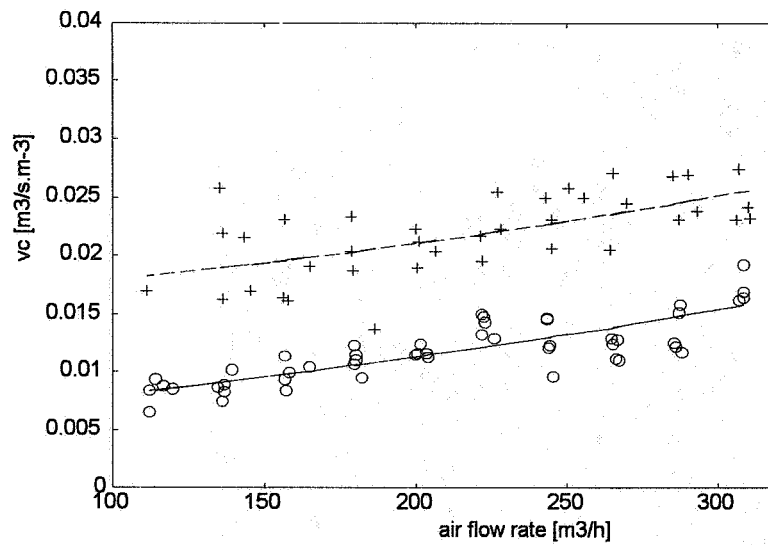


Figure 5: The $v_c/V, Q$ -relationship for sensor position [431].
('+' = 400 Watt, 'o' = 0 Watt)

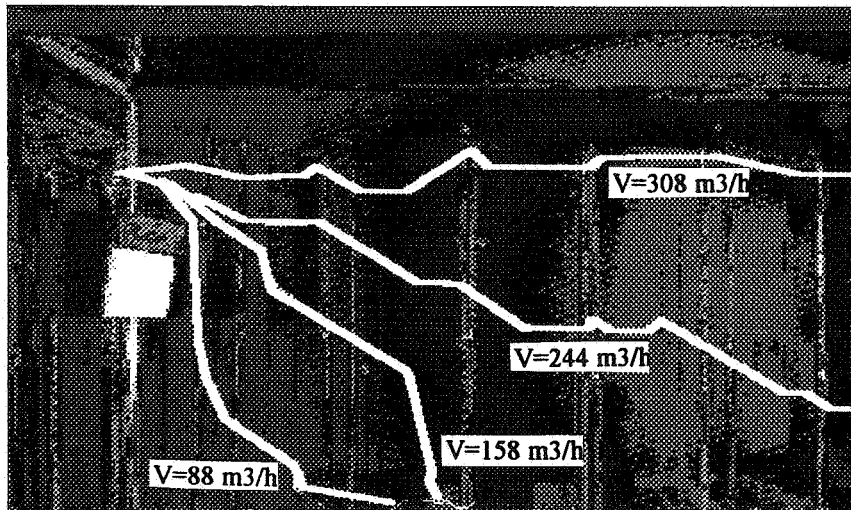


Figure 6: The centreline of the incoming air jet for different ventilation rates and corresponding Archimedes numbers:

- (1) $V=88 \text{ m}^3/\text{h}$, $Ar_c=393$, falling air flow pattern
- (2) $V=158 \text{ m}^3/\text{h}$, $Ar_c=111$, falling air flow pattern
- (3) $V=244 \text{ m}^3/\text{h}$, $Ar_c=42$, unstable air flow pattern
- (4) $V=308 \text{ m}^3/\text{h}$, $Ar_c=24$, horizontal air flow pattern

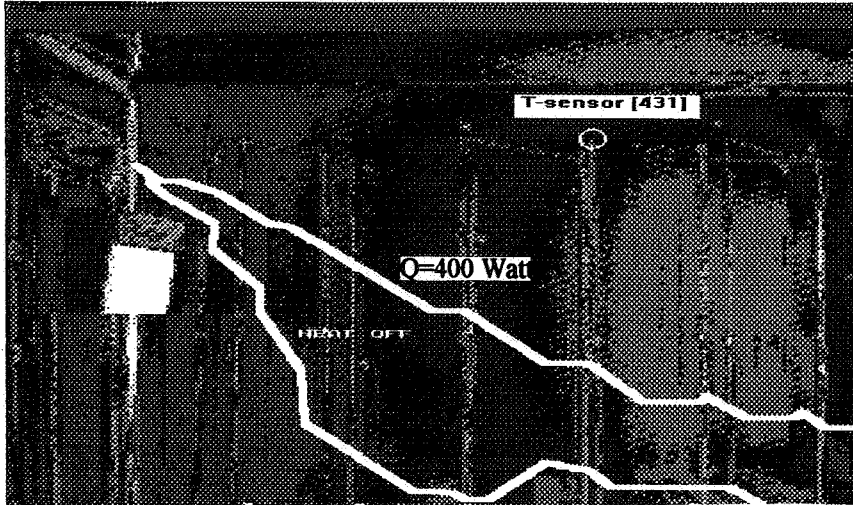


Figure 7: The centreline of the incoming air jet for a ventilation rate $V=148 \text{ m}^3/\text{h}$ with heat supply $Q=400 \text{ Watt}$ and without heat supply Q .

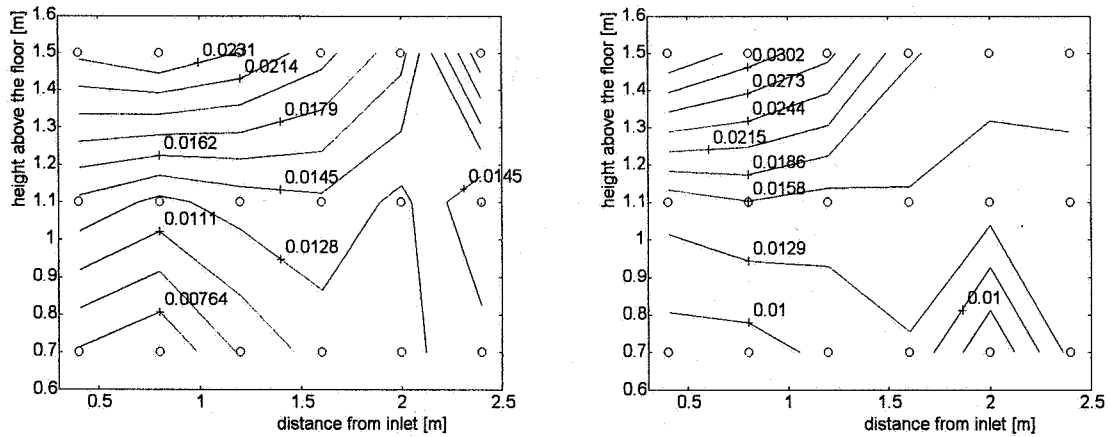


Figure 8: The distribution of the 'volumetric concentration flow of fresh air $v_c \text{ [m}^3/\text{s.m}^3\text{'}$ in the front plane of sensors (left figure) and the rear plane of sensors (right figure) for a ventilation rate V of $180 \text{ m}^3/\text{h}$ and a control heat supply Q of 400 Watt . The lines represent iso- v_c , the 'o' represent the 18 sensors in a plane.

file name	V _{begin} [m3/h]	V _{end} [m3/h]	Q [Watt]	To [°C]
me0510ta	63.78	112.32	0	11.55
me1210ta	54.52	112.06	0	11.22
me2210ta	59.27	136.57	0	11.83
me0511ta	62.34	117.06	0	11.99
me1611ta	71.47	119.87	0	11.67
me0510tb	59.83	136.65	0	11.77
me1310ta	59.11	134.92	0	11.35
me2210tb	60.26	136.43	0	11.97
me0511tb	62.52	114.12	0	12.22
me1611tb	66.82	139.34	0	11.89
me0610ta	57.03	157.20	0	11.80
me1310tb	61.24	156.44	0	11.52
me2210tc	60.74	158.31	0	11.96
me0611tb	58.18	156.79	0	12.44
me1911ta	78.10	165.06	0	11.69
me0610tb	64.12	180.32	0	12.33
me1310tc	53.80	179.57	0	11.65
me2610ta	62.67	179.84	0	12.03
me0911ta	57.07	180.01	0	12.13
me1911tb	64.59	182.41	0	11.67
me0610tc	58.38	200.80	0	12.37
me1410ta	28.72	200.15	0	11.60
me2610tb	72.12	204.14	0	12.09
me0911tb	58.68	201.43	0	12.43
me2011ta	63.64	203.81	0	11.80
me0710ta	61.39	221.98	0	12.02
me1410tb	58.86	222.59	0	11.71
me2810ta	57.97	223.25	0	12.10
me1011ta	58.23	222.14	0	12.15
me2011tb	67.11	226.00	0	11.96
me0710tb	54.72	243.51	0	12.39
me1410tc	58.59	244.40	0	11.79
me2910ta	56.76	243.76	0	12.14
me1011tb	53.47	243.25	0	12.32
me2211ta	70.15	245.45	0	11.85
me0810ta	59.86	264.63	0	12.03
me1510ta	61.12	267.41	0	12.00
me2910tb	56.52	265.41	0	12.40
me1011tc	59.44	267.14	0	12.08
me2211tb	67.92	266.25	0	12.04
me0810tb	54.83	285.92	0	12.36
me1510tb	54.15	288.57	0	12.19
me2910tc	59.36	286.42	0	12.59
me1111ta	66.83	287.76	0	12.18
me2211tc	66.82	287.18	0	12.21
me0810tc	55.71	307.28	0	12.50
me1510tc	51.22	308.91	0	12.26
me2910td	62.75	308.70	0	12.70
me1111tb	58.92	308.82	0	12.30
me2311ta	51.22	308.91	0	11.79

Table 1: 50 step up experiments with variable ventilation rate (Q=0 Watt)

file name	V begin [m3/h]	V end [m3/h]	Q [Watt]	To [°C]
meqv16t1	76.02	155.96	400	12.17
meqv14t1	76.87	135.17	400	12.33
meqv18t1	63.07	178.43	400	11.99
meqv20t1	55.26	199.90	400	12.04
meqv22t1	48.88	221.40	400	11.69
meqv24t1	68.03	244.93	400	11.88
meqv26t1	63.13	255.80	400	12.01
meqv28t1	68.76	287.43	400	11.94
meqv30t1	54.39	306.00	400	11.87
meqv12t1	64.46	111.47	400	11.42
meqv15t2	55.01	145.28	400	11.24
meqv20t2	51.79	200.78	400	11.57
meqv26t2	76.91	269.90	400	11.77
meqv30t2	76.45	310.78	400	11.57
meqv28t2	76.80	290.44	400	11.64
meqv24t2	79.95	250.33	400	11.41
meqv22t2	75.16	227.92	400	11.44
meqv18t2	75.64	186.40	400	11.33
meqv16t2	79.13	165.08	400	10.85
meqv14t2	79.87	143.60	400	11.30
meqv20t3	76.14	206.48	400	11.63
meav22t3	77.30	227.16	400	11.78
meqv24t3	50.47	244.84	400	11.38
meqv26t3	48.63	265.49	400	11.72
meqv28t3	77.43	293.41	400	11.76
meqv30t3	60.09	310.47	400	11.85
meqv18t3	62.98	178.40	400	11.42
meqv16t3	59.28	156.86	400	11.57
meqv14t3	59.05	136.12	400	11.56
meqv30t4	58.55	307.14	400	11.83
meqv28t4	60.95	285.32	400	11.81
meqv26t4	57.75	264.34	400	11.98
meqv24t4	65.26	243.09	400	11.50
meqv22t4	57.26	221.80	400	11.72
meqv20t4	63.11	200.86	400	11.67
meqv18t4	59.11	179.05	400	11.60
meqv16t4	58.21	157.59	400	11.66
meqv14t4	61.17	136.21	400	11.44
meqv12t4	64.46	111.47	400	11.72

Table 2: 40 step up experiments with variable ventilation rate (Q=400 Watt)

experiment	ventilation rate (m ³ /h)	To (°C)	heat Q (W)	Ar _c -number	smoke
smoke 5	88	10.98	0	393	TiCl ₄
smoke 6	124	11.05	0	212	TiCl ₄
smoke 7	158	11.00	0	111	TiCl ₄
smoke 8	201	11.11	0	65	TiCl ₄
smoke 11	244	11.49	0	42	TiCl ₄
smoke 13	286	11.51	0	29	TiCl ₄
smoke 14	308	11.52	0	24	TiCl ₄
smoke 15	101	10.9	0	298	TiCl ₄
smoke 16	101	10.9	500	-	TiCl ₄

Table 3: Overview of the experiments to visualise and quantify the air flow pattern

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

PRACTICAL EXPERIENCES WITH IR CONTROLLED SUPPLY TERMINALS IN DWELLINGS AND OFFICES

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1. INTRODUCTION

1.1 DEMAND CONTROLLED VENTILATION

Ventilation is necessary to provide a good indoor air quality to occupants in office buildings but is however a major energy consumer. In that manner, ventilation in itself can contribute to much more than 50% of the energy consumption for heating in well insulated office buildings. Likewise, the general trend in standards to augment ventilation requirements would still increase its energy costs. Thus, it seems obvious that an intelligent control of ventilation in office building allows to obtain substantial reductions of energy consumption.

To a certain extent this is also true for dwellings even if in general ventilation represents a smaller contribution to the energy consumption for heating than in office buildings. In this connection, it should be noted that the increasing requirements regarding insulation of dwellings has for effect to augment this proportion. Demand controlled ventilation in dwellings appears therefore also as an interesting way to achieve energy consumption reduction.

1.2 EXPERIMENTS PERFORMED

This paper presents experiments that were performed in two different environment but are both related to IR control ventilation.

1.2.1 IR CONTROL VENTILATION IN BEDROOMS

The biggest challenge for ventilation is to achieve the best acceptable indoor air quality at the right cost. Two options are already available in dwellings : heat recovery or modulation of the flows according to the needs.

In "normal" conditions, the simple extraction humidity controlled system (one sensor in each room, habitable or technical) is a good economical solution which has proved reliability for more than 10 years.

If the outside conditions need to be improved (noise, pollution, temperature, ...), or if the occupants have particular sensibility to some outdoor pollutants (allergy, asthma, ...), it can be necessary to use a more sophisticated system which blows the air after filtration, preheating, ...

It must be stressed that any treatment of the air has a cost, either through direct energy consumption (preheating, pressure loss, ...) either through maintenance of the systems.

Hence, it seems quite obvious to modulate the flows rather than to give a constant value which is not always necessary but which always needs to be treated.

The main issue is then to find the more accurate detector for the particular need in each room. The purpose of the experiment in bedrooms described here was to verify that it was possible to detect people through a motion detector, while sleeping, and to adapt ventilation to the occupancy. The idea being that even an eventual lack of ventilation for a short while (a few minutes) is not a problem.

Several nights have been recorded in more than 20 homes, with different occupancies (1 or 2 persons, adults, children) in order to know the frequency of movements and to determine the best parameters for the control of ventilation flows.

The result of these “1001 nights” are presented here and open a big field for products and systems provided that constructive discussions are open with regulation organisations to allow these modulated systems as energy efficient alternative to the “single value strategy” which is often the base of regulations.

1.2.2 EXPERIMENTS IN OFFICES

A first approach for demand control consists in ventilating only the spaces which are effectively occupied. Such a system was studied in detail by BBRI in an occupied office building and forms the subject of a previous publication [1].

A more advanced method would consist in controlling the ventilation as a function of the real indoor air quality in the spaces. This technique seems however difficult to apply in practice since the indoor air quality is determined by a large number of parameters often impossible to evaluate (odours, VOC's, ...).

However, in the event the occupants form the main pollution source in a ventilated space, it is possible implement a demand control strategy based on the occupancy. This can be done by using the carbon dioxide as a tracer of human pollution (see [3]). That technique is quite attractive since it takes into account the number of people in the space as well as the natural infiltration. However, up to the present, it is only appropriate for application in large rooms (classrooms, theatres,...) mainly for a question of cost. Another method is to evaluate the pollution source in space by counting the number of people effectively present and to ventilate accordingly. The AGITO type ventilation terminal is based on this principle. Two terminals were installed in two offices in one of BBRI building in order to evaluate in-situ their performances.

2. ESTIMATION OF ENERGY SAVINGS BY USING IR CONTROLLED VENTILATION

In order to provide a quantification of the energy savings that can be made thanks to IR controlled ventilation, some simulations in a well-insulated large office building were performed.

As debated in [2], large variations of the ventilation requirements can be found in different standards. Two extreme values were chosen in the simulation cases. Firstly, the minimum ventilation requirement for single office in the CEN prENV 1752 (class C, clean building, version 12-94) was used, that is 0.8 l/s.m². Then, a high value of 8.2 l/s.m² was taken which corresponds with the class A of the mentioned proposal for standard with an average pollution emission from the building (for more details see [2]).

An average occupancy of 50% was assumed which is in line with what was measured in the BBRI office building in previous experiments (see [1]).

The other assumptions are:

- 4 facade office building, 20 storeys
- ground dimensions: 20 x 20 m²
- storey height: 3 m
- 30% total glazed area
- Heating system: target value of 20°C, operation from 6:00 to 19:00 during the week
- Ventilation system operation from 6:00 to 19:00 during the week
- Component characteristics :
 - facade: $U = 0.4 \text{ W/m}^2\text{K}$
 - windows: $U = 1.5 \text{ W/m}^2\text{K}$, solar factor = 0.37
 - Roof: $U = 0.3 \text{ W/m}^2\text{K}$
 - Floor: $U = 0.4 \text{ W/m}^2\text{K}$
 - Internal gains : 30 W/m²
- Natural infiltration = 0.2 l/s.m²

The following table gives the simulation results for both ventilation requirements. The energy gain compared with a constant air flow strategy is given as well.

	0.8 dm ³ /s.m ²	8.2 dm ³ /s.m ²
Constant air flow rate during working hours	347000 kWh 43 kWh/m ² 100%	2472000 kWh 309 kWh/m ² 100 %
50% operation because of occupancy control	252000 kWh 32 kWh/m ² 72 %	1283000 kWh 160 kWh/m ² 52%
Gains due to presence control	95000 kWh 12 kWh/m ² 27 %	1189000 kWh 149 kWh/m ² 47%

Table 1 : Predictions of energy demand for heating and relative gains due to demand controlled ventilation in a well-insulated office building

As it can be seen, the impact on the total heating energy consumption is very important. Indeed, it is a fact that ventilation is a large energy consumer in this kind of office building. It must also be noted that the impact is much larger with the high ventilation requirement because, in this case, the transmission losses are almost negligible compared with the ventilation losses.

Important to mention is also that passing from the low ventilation requirement to the high ventilation requirement translates in an increase of a factor 7 of energy consumption.

3. DESCRIPTION OF THE VENTILATION TERMINALS

3.1 INFRA-RED DETECTION

The principle of the infra-red detection is widely used for alarm purpose. The general principle of a passive infrared detector consists in an electrical signal which varies with the intensity of the heat (infrared wave length). Any modification in the electrical signal is associated to a movement (the waves lengths range of infrared is close to the range of infrared emitted by human body).

This principle has been adapted to the OPTO and AGITO ventilation terminal. There were several constraints such as the installation which can be on the ceiling or the wall, the detection zone unit has to be as accurate as possible to avoid any dead zone. These problems

were solved by using a Fresnel lens with 31 view faces facing the room with a 110 ° angle. Moreover the passive infrared detector is a twin-head so the 31 view faces are then doubled. The accuracy of the infrared detector associated with the Fresnel lens is based upon the sum of the length of the borders between the view zone and the blind zone. Thus the number of Fresnel faces time the number of detectors gives here the total accuracy of the system.

3.2 OPTO TERMINAL

The OPTO terminals have the infrared sensor built-in the unit. The electronic card is activated by 4 batteries lasting more than 4 years. When a movement is detected, the electronic card supplies an electric current to the first micro-capsule which switches a micro-valve, blowing down a plastic bag attached to a shutter which then opens and allows the airflow passing. When no movement is detected during 15 min, the electronic card supplies a current to a second micro-capsule which switches the micro-valve, blowing up the bag and closing the shutter. This is instrumental for the batteries to last 4 years, the energy used to open and close the shutter is NOT electric but pneumatic.

3.3 AGITO TERMINAL

The AGITO uses the same infrared detector than the OPTO. The AGITO terminal has a microprocessor activated by a 12 VAC or DC hard wired. The micro-valve is replaced by a micro mixing valve. The AGITO modulate the shutter opening by mixing the pressure of the plastic bag.

One movement is counted when at least one movement has been detected over a period of 4 seconds. The microprocessor counts the number of the above kind of movements over a period of 3 minutes. It compares this number to a fixed table in order to set the AGITO opening. The AGITO opening can only move from a position to the next or the previous one. Therefore, the AGITO changes its opening only every 3 minutes. It goes never from the maximum to the minimum opening in once but step by step to the minimum. It takes 15 minutes which is equivalent to 60 m³/h run on during these 15 minutes.

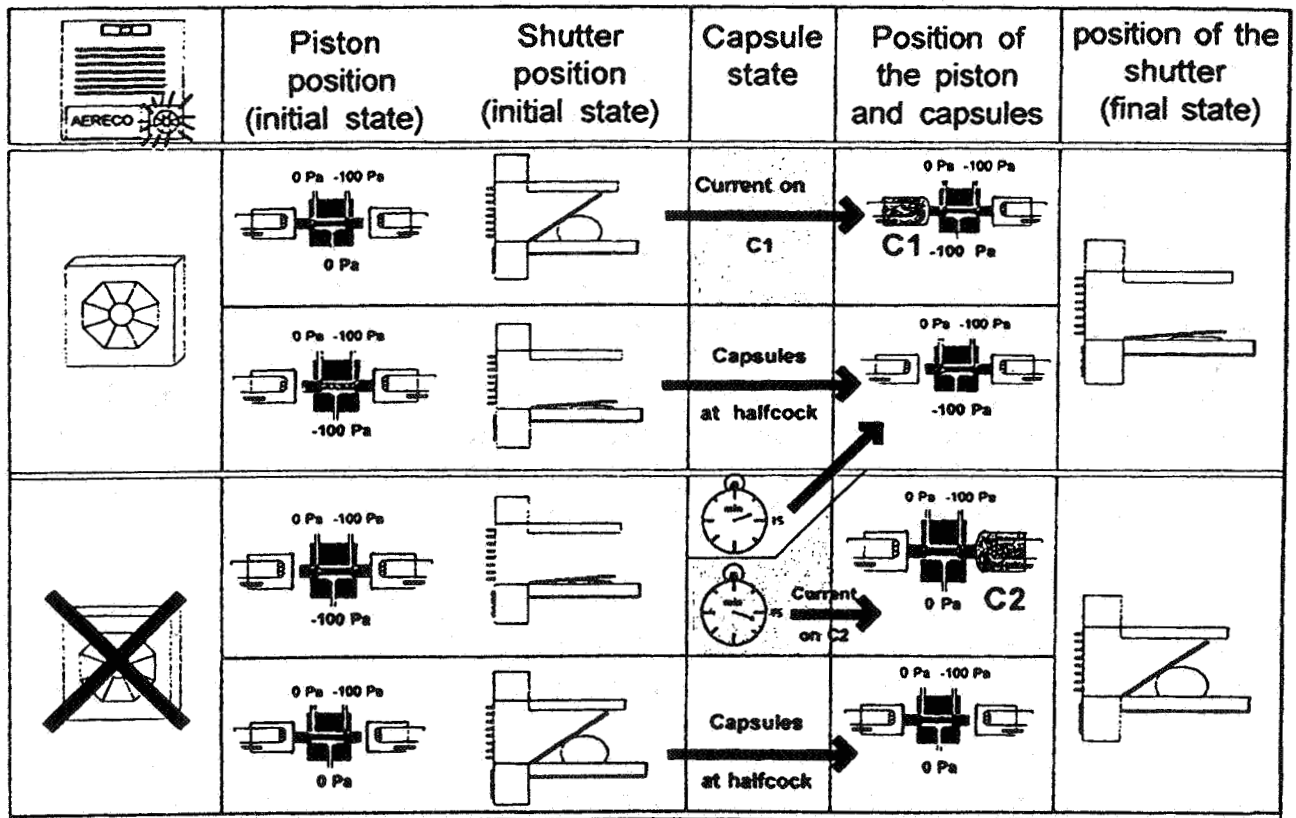
The AGITO is easily adaptable to specific requirements.

It is possible :

- to accelerate the functioning (2 minutes rather than 3 minutes)
- to delay the closing of the AGITO by having a 6 minutes sum downward and 3 minutes sum upward
- to leave a minimum opening when nobody is in the room
- to have a non-linear opening versus the number of movements
- to use a special connection to switch a separate routine in the program so over ride the AGITO for a given time for example.

Figure 1 illustrates the operation of the OPTO and AGITO ventilation terminals.

PRINCIPLE OF THE OPTO FUNCTIONNING



Février 1982

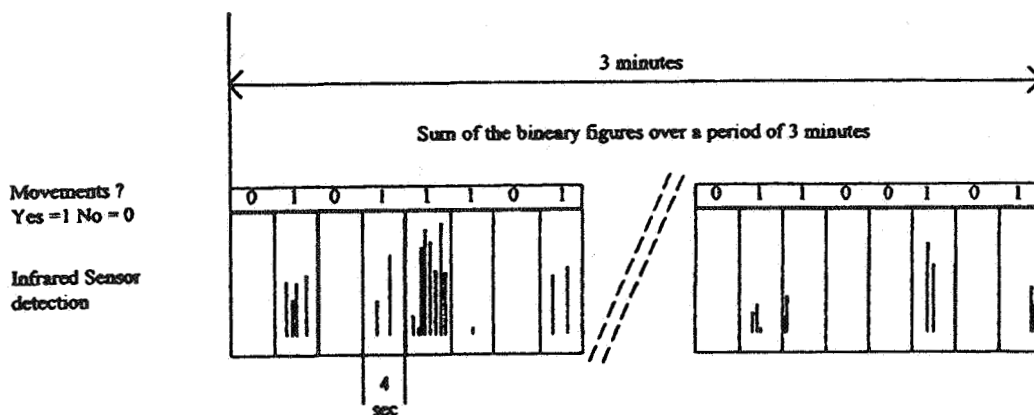


Figure 1 - Operation of the infra-red controlled ventilation terminal

4. EVALUATION OF THE AGITO VENTILATION TERMINAL IN TWO OFFICES

4.1 EXPERIMENTAL SET-UP

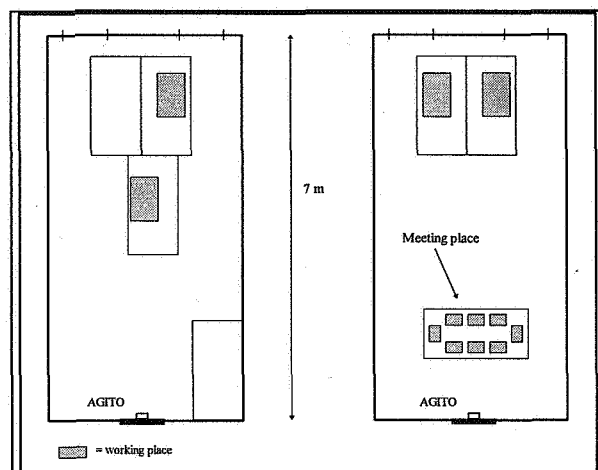


Figure 2 - Rooms with AGITO type ventilation terminals - Top views

As explained, two AGITO ventilation terminals were installed in two office rooms. The following figures show the position of the working places and the AGITO terminals. It is important to note that the distance between the AGITO terminal and the working places is quite long (excepted for the meeting place). Generally the detector is placed closer to the working place, for example in the ceiling.

The state of both AGITO ventilation terminals was measured in parallel with the real occupancy in both rooms. In addition, the carbon dioxide concentration was recorded in the office with two working places.

4.2 MEASUREMENT RESULTS

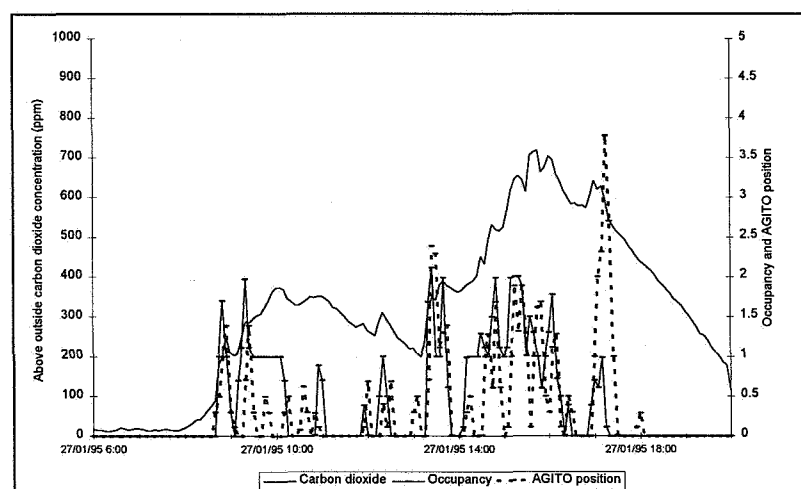
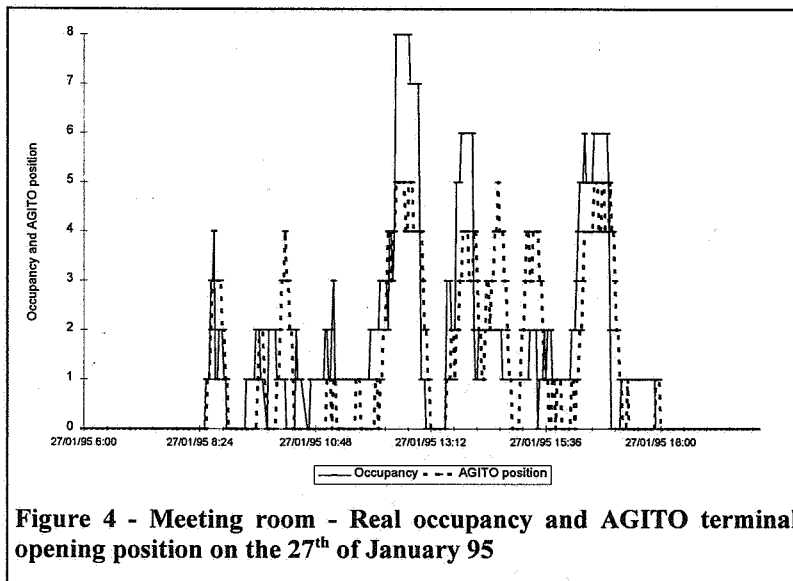


Figure 3 - Two working places room - Above outside CO₂ concentration, real occupancy and AGITO terminal opening position on the 27th of January 95

The following figures show the measured values (AGITO opening position, real occupancy as well as the CO₂ concentration in the two working place room) for one typical day chosen in the measurement period. It must be noted that the AGITO opening position corresponds with the number of “detected” people in the ventilated space, i.e. the number of people corresponding with the number of movements observed according to the conversion table

programmed in the terminal. The outside CO₂ concentration has been subtracted to the CO₂ concentration measured in the room.

It has to be added that the AGITO opening position and the real occupancy were measured every 2 minutes while the CO₂ concentration was recorded each 5 minutes. In order to perform the analysis, all the 2 minutes measurements were averaged on a 5 minutes period which explains that not entire occupancy values or opening positions are observed on the following figures.



reminded that the occupancy detection is based on the integration of movements in the room during a 3 minutes period and that, as a consequence, the occupant behaviour can strongly impact on the number of people “seen” by the terminal. Moreover, the detector is quite far from the working places (excepted the meeting place) which probably makes the detection more difficult.

The following figures give the average opening position of the ventilation terminal in function of the real occupancy. The number of recorded samples is also indicated for each point.

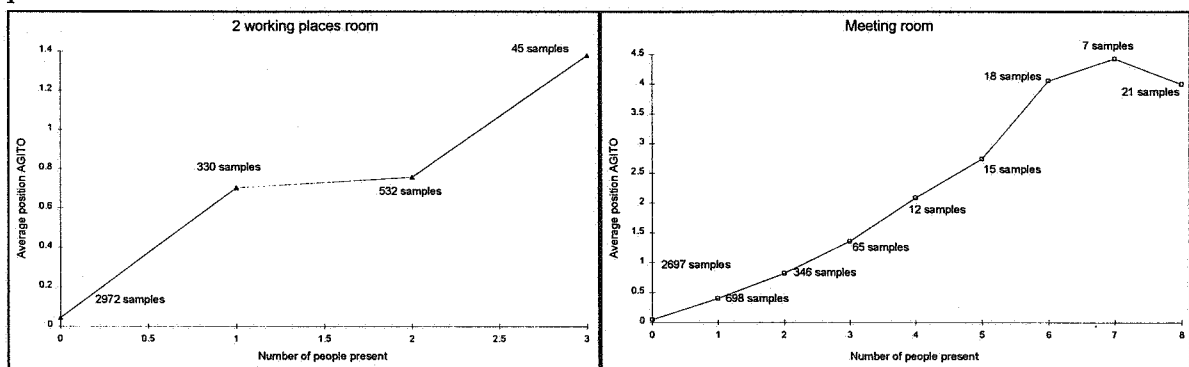


Figure 5 - Average opening level versus occupancy

It can be seen on Figure 5 that the relation between the average AGITO opening and the real occupancy is quite linear when the occupancy is low. As previously mentioned, the lowest the number of samples, the lower the representativity which, with the fact that the AGITO terminal are limited to 5 people occupancy, gives reasons why deviations from the linear relation are observed for high occupancy values.

It is interesting to observe that the relation is similar in both rooms expected for the high occupancy values in the meeting rooms and for an occupancy of 1 person in the two working place room. This last deviation is difficult to explain and could be due to a bad measurement of the real occupancy. The average behaviour is given by the following formula relating the AGITO terminal opening to the real occupancy:

Eq. 1
$$\text{Average opening position} = 0.5 \times \text{number of people} \quad [/]$$

The measurement points corresponding with real occupancy value higher than 5 and an opening position of 5 were removed to proceed to the analysis since the terminal is limited to 5 people. However, this affect not strongly the result since a very few points are concerned.

It should be said that the reaction time of the AGITO terminal does not influence this last analysis since the delay at the opening of the terminal (when more people are detected) is compensated by the delay at the closing of the terminal (when less people are detected).

It can be concluded that the number of people detected by the AGITO terminal is on average the half of the real occupancy. The reason why the occupancy is underestimated could be the long distance between the working places and the AGITO detector. It is therefore advisable in such configuration either to lower the detection level of the terminal or to place the detector closer to the working places.

4.3 CONCLUSIONS

- The AGITO terminal seems to react to the presence of people in the ventilated space and to follow more or less the occupancy variation. However, the occupancy estimated by the sensor was on average only the half of the actual occupancy. This is probably due to the long distance between the occupants positions and the infra-red sensor.
- Therefore a special attention should be paid to the sensor location when installing these terminal. In the event the sensor cannot be placed close to the occupants (in renovation for example), the conversion table of the terminal could be adapted to take account for the distance.

5. THE USE OF THE OPTO TERMINAL FOR VENTILATION OF BEDROOMS

5.1 EXPERIMENTAL SET-UP

The recording of movements during the nights in bedrooms was achieved through the same motion detectors used in the OPTO terminal. The detectors were placed in different locations in the rooms, in possible positions for a ventilation terminal, “looking” in the direction of the bed. Two detectors were used at each time, at different locations, to verify the variability of response in the same conditions of sleeping. (A previous experiment had shown the reliability and repeatability of 16 detectors in the same place, looking at the bed, but with small differences in the angles).

In order to avoid the saturation of the memory we used “reduced movements” : a “reduced movement” is a period of 4 seconds in which at least 1 movement occurs:

If 1 or more movements occur during this period, the counter is set to 1 and kept to this value. Each recorded value is the number of “reduced movements” detected in a 3 minutes period. The maximum value is thus 45 (3*15 reduced movements) (which value is very rarely attained as it implies quasi continuous detection for more than 3 minutes).

A full day gives 480 values. Eight nights (and days) were recorded at each campaign and transferred to a PC for storing and analysis. More than 1000 nights have been recorded from the beginning of the experiment.

5.2 MEASUREMENT RESULTS

5.2.1 TYPICAL DETECTION PATTERN

Following figures show typical detection pattern as recorded in two different bedrooms during the night. One can observe that a lot of movements are recorded during the night although during some periods the sensor “does not see” the sleeping people anymore.

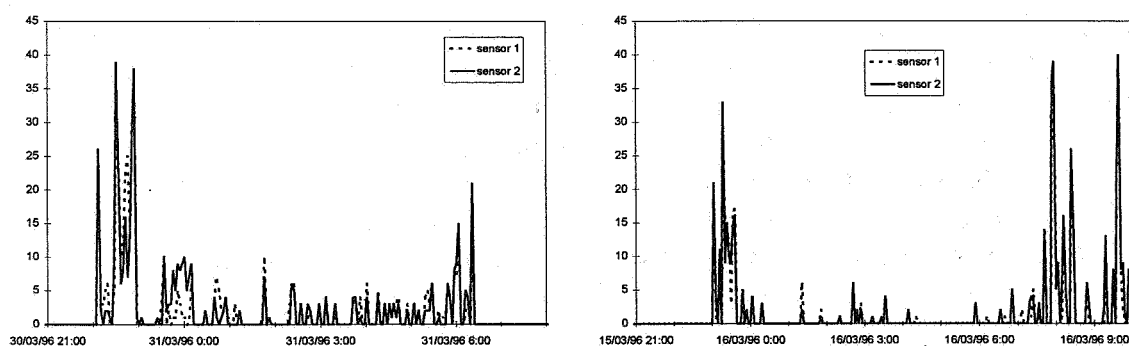


Figure 6 - Typical infra-red detection pattern in bedrooms

5.2.2 WHAT DELAY SHOULD BE TAKEN FOR SWITCHING OFF THE VENTILATION?

As illustrated by the previous figures, there are periods during which sleeping people are not making any movement that can be detected by the sensor. Therefore, the delay for switching off the ventilation after last detection must be chosen with care. Indeed, a too long delay would diminish the energy gain that can be obtained thanks to the IR control and a too short time constant would result in an unacceptable air quality. The intention is to study in detail the large amount of data recorded so as to determine on a statistical basis the optimal delay. The outcome of this study will be proposed for publication at the next AIVC conference.

6. GENERAL CONCLUSIONS

Simulation of demand controlled ventilation based on IR detection in a large well-insulated office building showed that substantial energy savings can be obtained.

The study of the AGITO terminal in two offices showed that the device is able to adapt the flow rate as a function of the real occupancy. It was highlighted that a special attention must be paid to the sensor location.

A large amount of data on the detection of occupancy in bedrooms during nights were collected. A first analysis of these measurements shows that it could be possible to control ventilation on the basis of infra-red detection. However, the time constant of the terminals must be chosen with care. A statistical analysis of the measurements will occur next year so as to determine optimal time constant.

ACKNOWLEDGEMENTS

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

The Influence of Outdoor Air Vents and Airtightness on Natural Ventilation - Calculations Based on Measurements

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Synopsis

Many modern buildings in the Nordic countries have mechanical ventilation. Passive stack ventilation is, however, an accepted ventilation system in the Nordic countries according to the current building codes. The building authorities need to be able to supply guidelines on natural ventilation systems in modern buildings, in order to fulfill the requirements on a healthy indoor climate at a reasonable energy cost. Therefore a project was initiated by the Nordic committee on building regulations. The aim of the project was to present functional and technical requirements on natural ventilation systems.

The paper discusses the influence of outdoor air vents and airtightness on passive stack ventilation for Nordic dwellings. Calculations, based on measurements, were performed for a one storey one-family house, a 1½-storey one-family house and two apartments in a three-storey building. Cumulative distributions of ventilation rates for a year were calculated, using a multi-zone air flow model, for individual rooms. It is shown that the ventilation rate will vary very much, without any interaction from the occupants or an automatic control system. A modern system for passive stack ventilation must include a system for control of the air flows in order to avoid excessive use of energy and inadequate indoor air quality.

1. INTRODUCTION

Most modern buildings in the Nordic countries, especially in Sweden, have mechanical ventilation. During the last couple of years natural ventilation is being discussed very much in the Nordic countries. Natural ventilation (passive stack ventilation) is an accepted ventilation system in buildings in the Nordic countries according to the current building codes. The building authorities need to be able to supply guidelines on requirements which should be made on a natural ventilation system in modern buildings, in order to fulfill the requirements on a healthy indoor climate to a reasonable energy cost. Therefore a project was initiated by the Nordic committee on building regulations. The main purpose was to present functional and technical requirements on natural ventilation systems (Säteri 1996), but also:

- summing up of current studies
- description of the real performance of natural ventilation systems
- control alternatives
- influence of outdoor air vents and airtightness
- design of natural ventilation systems
- requirements on outdoor air vents and passive stacks
- ventilation measurements

This paper discusses the influence of outdoor air vents and airtightness on passive stack ventilation for Nordic dwellings. Calculations, based on measurements, were performed using a multi-zone air flow model.

2. THE DWELLINGS TESTED

A one-storey one-family house with basement (total floor area 158 m²) was examined, as this type is a very common type of house with passive stack ventilation. Most of these houses were built before 1970. Many houses built later on are equipped with exhaust fan ventilation and built as a 1½-storey house without basement. A conventional 1½-storey house (floor area

119 m²) was therefore converted (in theory) from mechanical exhaust to passive stack ventilation, and examined. A conventional apartment building (floor area/apartment 56 m²), built during the fifties, was also chosen, of which all have passive stack ventilation. The passive stacks were designed to remove air from bathrooms and kitchens, to the outside. All examined dwellings have been tested i. e. pressurization, tracer gas measurements etc..

3. METHODS

A multi-zone model, MOVECOMP, was used as a tool for further evaluating the measurements (Herrlin 1987, Bring 1988). In the program the building and its ventilation system is modelled. The model consists of pressure nodes connected to each other with flow paths. The nodes are different zones and duct components, while the flow paths are different leakage paths and ducts. The air flows are calculated by seeking a flow balance in each node. Mass balance has to be achieved.

In the calculations weather data from the reference year 1971 from Stockholm was used. The entire year was reduced to 100 combinations of different outdoor temperatures and wind speeds with frequencies for every combination. Wind pressure coefficients from wind tunnel studies were used (Chien 1951, Wirén 1985).

For the calculations the dwellings were divided into different zones, where each zone is a room. The distribution of leakage paths is based on IR-scans and knowledge about the construction technique in question. In most cases the leakage has been distributed between exterior walls, ceiling and floor according to each components area. For each facade the assumption was made that 25 % of the leakage is located 0.1 m above the floor, 50 % 1.2 m above the floor (representing windows) and 25 % 2,3 m above the floor. Windows, exterior doors and doors to bathrooms were assumed to be closed. All other doors were assumed to be open. Bathroom doors have an opening for transferred air. Interior walls and intermediate floors were assumed to be airtight.

4. RESULTS

A previous comparison of measured and calculated ventilation rates showed for the examined one-family houses, that MOVECOMP can overpredict the outdoor air ventilation with 15 - 40 %, but can make a reasonable estimation of the variation in ventilation rate over time (Blomsterberg 1990). It should therefore be possible to use MOVECOMP for the parametric studies presented in the following chapters.

4.1 A one-storey house with basement, Svaneholm

The first calculation was carried out with the ventilation system as installed, which means e. g. no outdoor air vents in kitchen and bedroom. The airtightness (excl. outdoor air vents) was assumed to be 3 ach at 50 Pa. This calculation resulted in a yearly average outdoor air ventilation rate of 0.22 ach, if the pressure coefficient for the top of the passive stacks is 0.0 and 0.30 ach if the pressure coefficient is - 0.3 (see table 4.1). In the bedrooms the outdoor air ventilation rate is lower than 0.3 ach during 90 % of the year (see figure 4.1). According to the current Swedish Building Code the outdoor air ventilation should be 4 l/(s person), which corresponds to 1.1 ach for bedroom 1 and 0.8 ach for bedroom 3 i. e. the ventilation is too low.

Table 4.1 Calculated outdoor air ventilation for the Svaneholm house. The airtightness (excl. outdoor air vents) is 3 ach at 50 Pa.

Total passive stack area, cm ²	Total outdoor air vent area, cm ²	Pressure coefficient, passive stack	Yearly average $\pm$ variation, ach	Max, ach	Min, ach
700	150	0,0	0,22 $\pm$ 0,08	0,59	0,01
700	150	-0,3	0,30 $\pm$ 0,11	0,81	0,02
700	300	-0,3	0,32 $\pm$ 0,12	0,94	0,11
1050	300	-0,3	0,37 $\pm$ 0,13	0,98	0,12
1050	425	-0,3	0,42 $\pm$ 0,16	1,32	0,14
1050	740	-0,3	0,54 $\pm$ 0,22	2,05	0,17
1050	1050	-0,3	0,65 $\pm$ 0,29	2,71	0,18

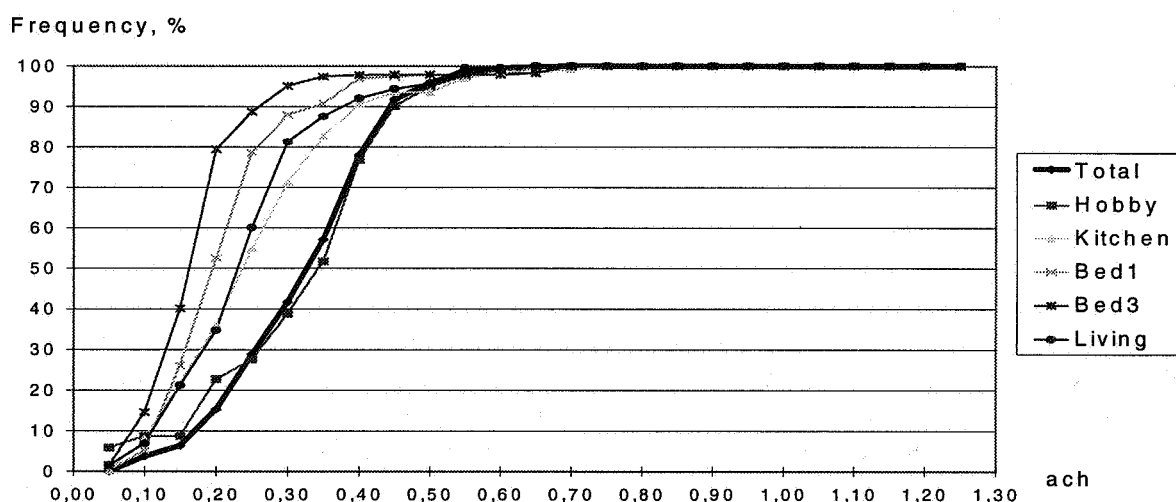


Figure 4.1 Cumulative distribution of outdoor air change rates in the Svaneholm house. The passive stack pressure coefficient is - 0.3, the total area of outdoor air vents 150 cm² and the ventilation system is according to the Swedish Building Code of 1967, but no outdoor air vents in kitchen, living-room and bedrooms.

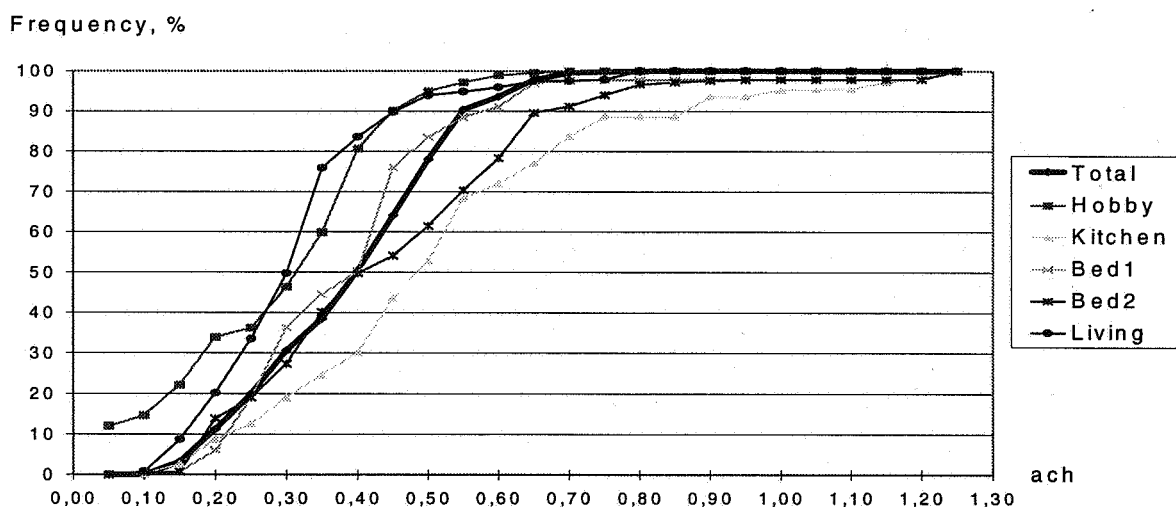


Figure 4.2 Cumulative distribution of outdoor air change rates in the Svaneholm house. The total area of outdoor air vents is 300 cm² and of passive stacks 1050 cm².

If the ventilation system was designed according to the Swedish Building Code of 1967, which means that outdoor air vents are added to kitchen and bedroom, the yearly average ventilation rate will increase from 0.30 to 0.32 ach. If the passive stacks are designed according to new recommendations (Boverkett 1994), which means that the cross section is increased from 700 cm² to 1050 cm², then the ventilation rate is increased by another 0.05 ach, to 0.37 ach. For the bedrooms this means that during 75 % of the year the ventilation is lower than 0.5 ach (see figure 4.2). Calculations show that the outdoor air vents have to have a total area of appr. 700 cm² to arrive at a yearly average total ventilation of 0.5 ach (see table 4.1 and figure 4.3). During a substantial part of the year the ventilation will then be too low and other parts of the year too high. If the outdoor air ventilation is to be higher than 0.5 ach during 75 % of the year the outdoor air vents have to have a total area of at least 1050 cm² (7 cm²/(m² floor area)) (see figure 4.4). During some parts of the year the ventilation will then be very high, up to 2.7 ach.

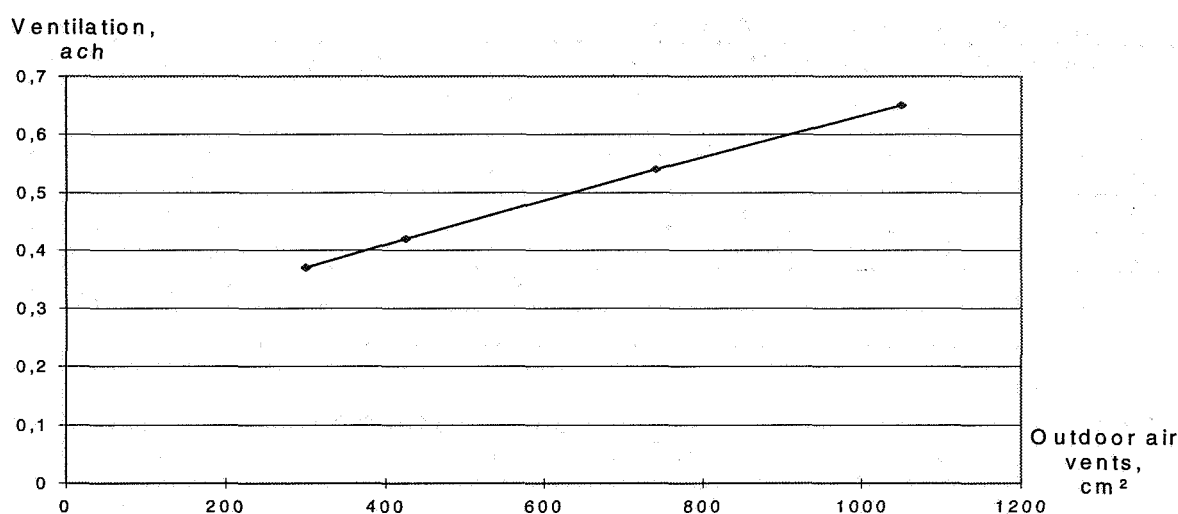


Figure 4.3 Yearly average total outdoor air ventilation as function of the total area of the outdoor air vents for the Svaneholm house with a total passive stack area of 1050 cm².

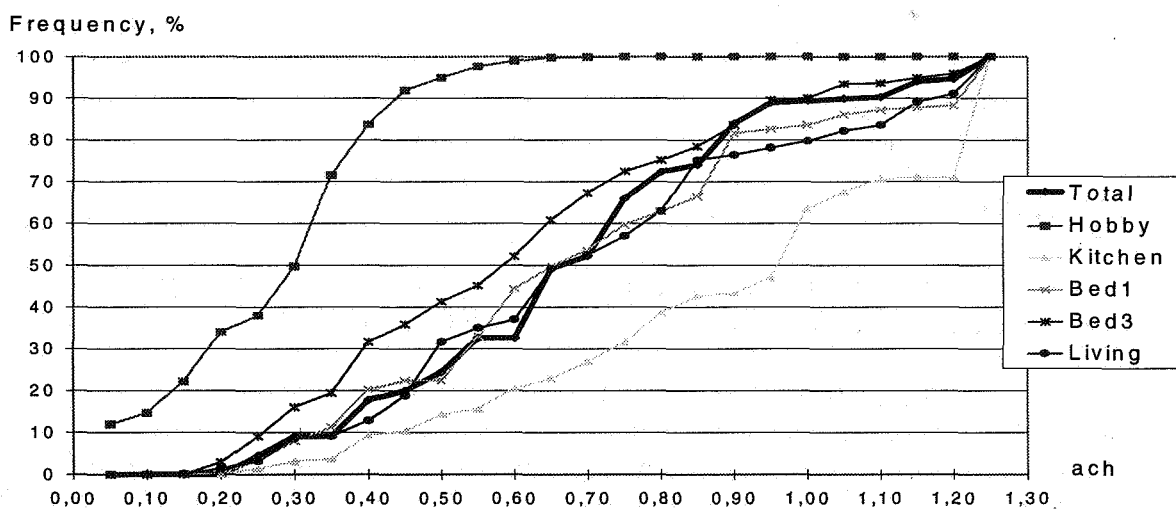


Figure 4.4 Cumulative distribution of outdoor air change rates in the Svaneholm house. The total area of outdoor air vents is 1050 cm² and of passive stacks 1050 cm².

4.2 A 1½-storey house with crawl-space, Lättbygg85.

The first calculation was carried out with a ventilation system designed according to the Swedish Building Code of 1967. The airtightness of the building envelope is assumed to be 3 ach (excl. outdoor air vents) at 50 Pa. The outdoor air vents were the same as for the original exhaust fan ventilation system. This calculation resulted in a yearly average outdoor air ventilation of 0.34 ach, if the pressure coefficient at the top of the passive stack is 0.0 and 0.41 ach if the pressure coefficient is - 0.3 (see table 4.2). The Lättbygg85 house has a higher ventilation rate than the Svaneholm house thanks to e. g. an open staircase between ground and first floor. In two of the bedrooms the ventilation rate is lower than 0.3 ach during 60 % of the year (see figure 4.5). According to the current Swedish Building Code the outdoor air ventilation rate should be 4 l/(s and person), which for the master bedroom corresponds to 1.2 ach and for one of the smaller bedrooms 1.1 ach i. e. the ventilation is too low.

Table 4.2 Calculated outdoor air ventilation for the Lättbygg85 house. The airtightness (excl. outdoor air vents) is 3 ach at 50 Pa.

Total passive stack area, cm ²	Total outdoor air vent area, cm ²	Pressure coefficient, passive stack	Yearly average $\pm$ variation, ach	Max, ach	Min, ach
700	70	0,0	0,34 $\pm$ 0,12	1,01	0,03
700	70	-0,3	0,41 $\pm$ 0,15	1,19	0,03
1050	70	-0,3	0,45 $\pm$ 0,16	1,22	0,03
1050	225	-0,3	0,63 $\pm$ 0,23	1,87	0,04
1050	610	-0,3	1,01 $\pm$ 0,37	3,20	0,07

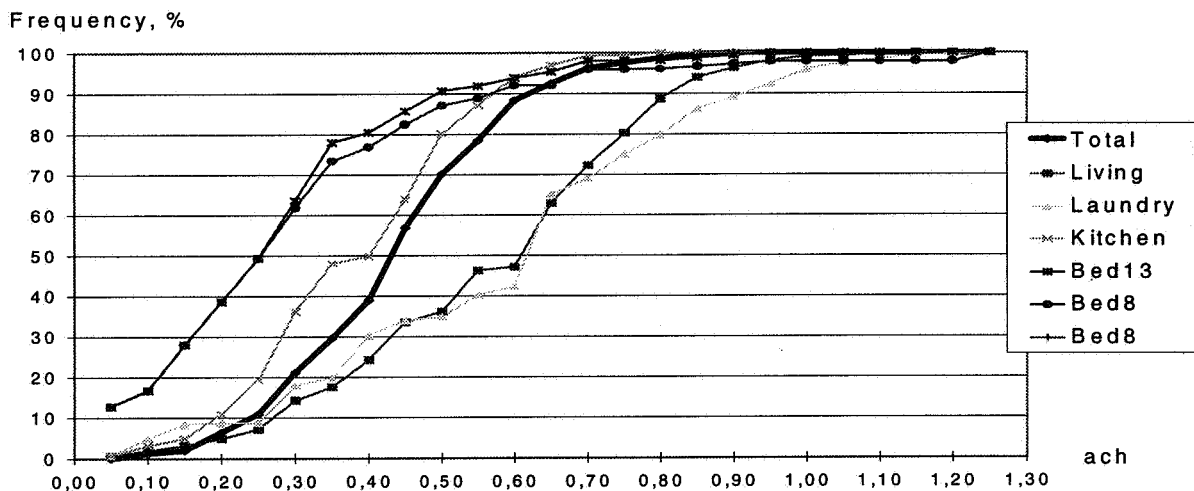


Figure 4.5 Cumulative distribution of outdoor air change rates in the Lättbygg85 house. The ventilation system is according to the Swedish Building Code of 1967.

If the passive stacks are designed according to new recommendations (Boverkett 1994), which means that the cross section is increased from 700 cm² to 1050 cm², then the ventilation rate is increased by another 0.05 ach, to 0.45 ach. For two of the bedrooms this means that during 90 % of the year the ventilation is lower than 0.5 ach (see figure 4.6). Calculations show that the outdoor air vents have to have a total area of 115 cm² to arrive at a yearly average total ventilation of 0.5 ach (see table 4.2 and figure 4.7). If the outdoor air ventilation is to be

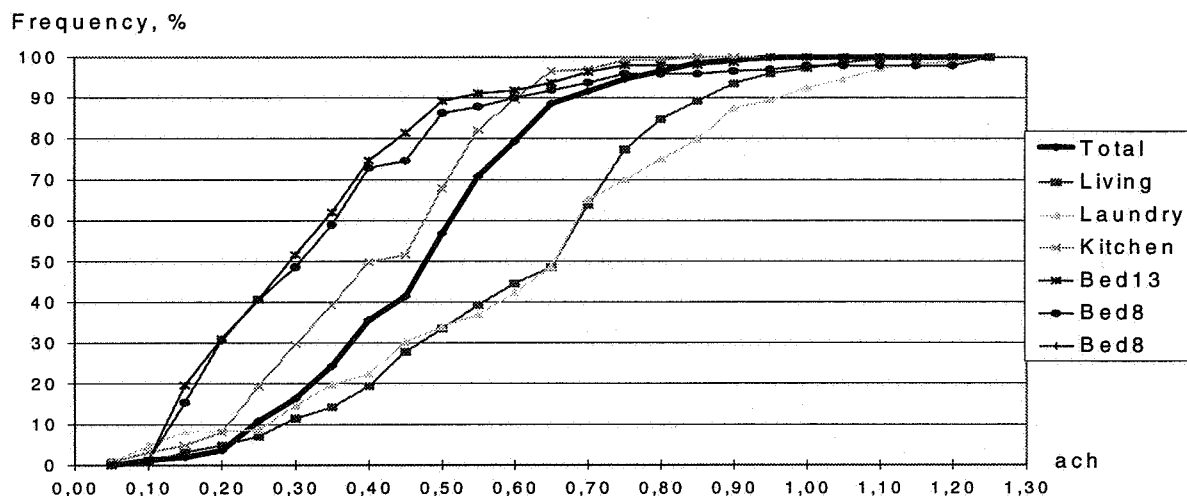


Figure 4.6 Cumulative distribution of outdoor air change rates in the Lättbygg85 house. The total area of outdoor air vents is 70 cm² and of passive stacks 1050 cm².

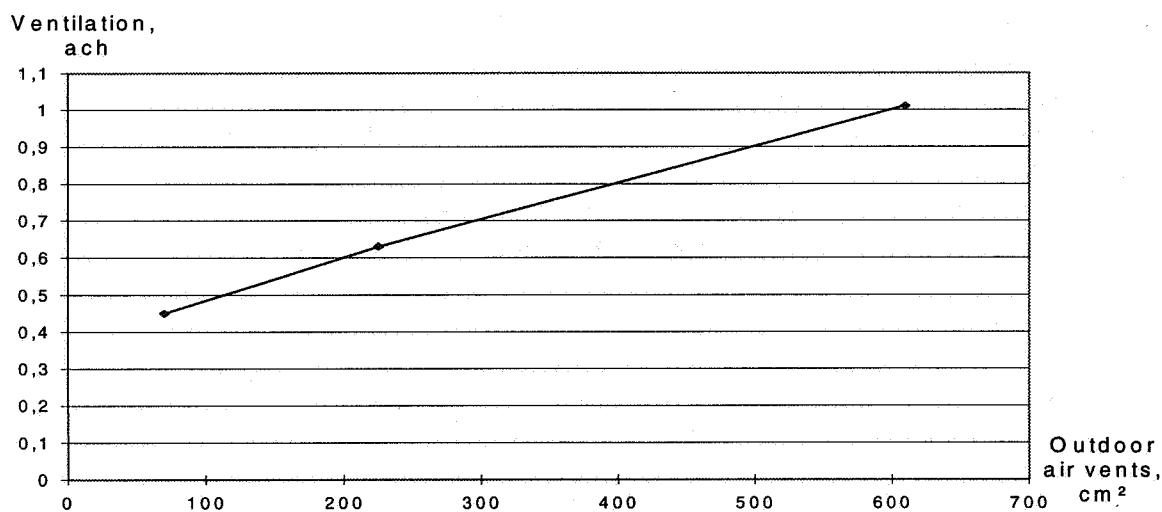


Figure 4.7 Yearly average total outdoor air ventilation as function of the total area of the outdoor air vents for the Lättbygg85 house with a total passive stack area of 1050 cm².

higher than 0.5 ach during 75 % of the year the outdoor air vents have to have a total area of at least 225 cm² (2 cm²/(m² floor area)) (see figure 4.8). During some parts of the year the ventilation will then be very high, up to 1.9 ach.

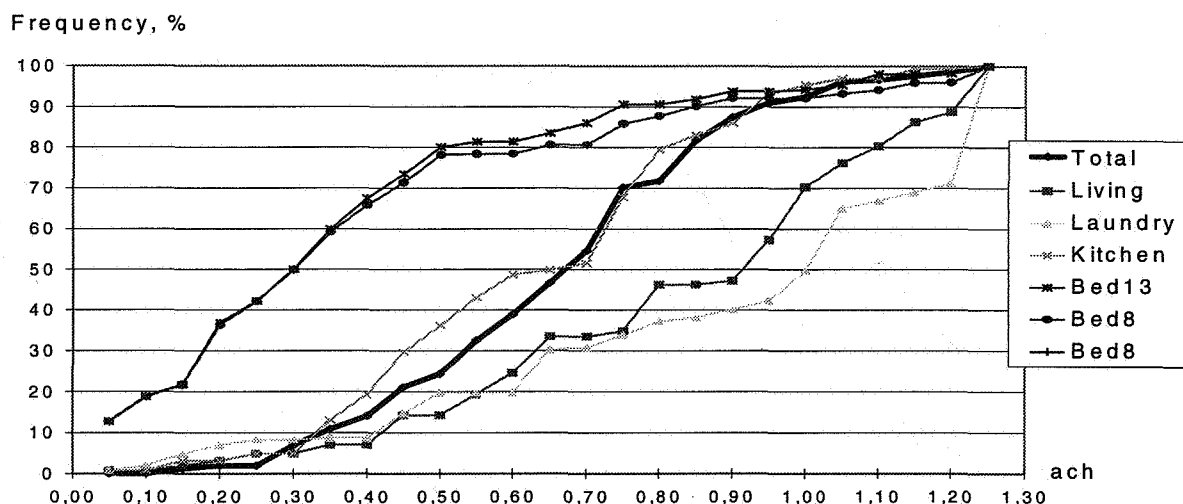


Figure 4.8 Cumulative distribution of outdoor air change rates in the Lättbygg85 house. The total area of outdoor air vents is 225 cm² and the total passive stack area 1050 cm².

4.3 An apartment building with three storeys, Sjöbo

The first calculation was carried out with the ventilation system as originally designed. The airtightness of the building envelope is assumed to be 1 ach (excl. outdoor air vents) at 50 Pa. This calculation resulted in a yearly average outdoor air ventilation of 0.48 ach for an apartment on the third floor, if the passive stack pressure coefficient is -0.3 (see table 4.3). In the bedroom the ventilation rate is lower than 0.3 ach during 50 % of the year (see figure 4.9). According to the current Swedish Building Code the outdoor air ventilation rate should be 2 x 4 l/s (and person) for the bedroom i.e. 0.8 ach. The ventilation is too low in the bedroom.

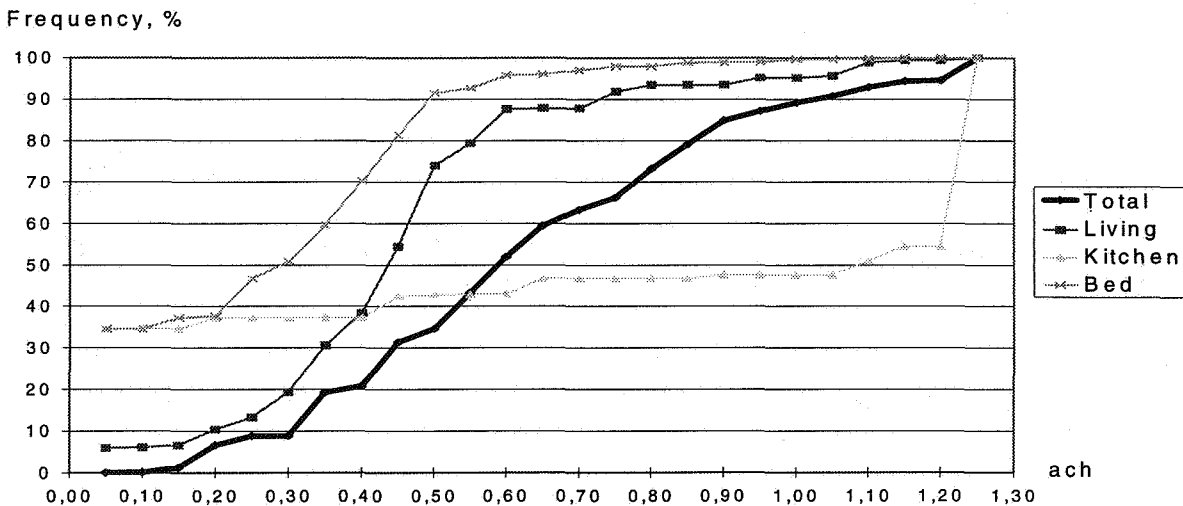


Figure 4.9 Cumulative distribution of outdoor air change rates in the apartment. The total area of outdoor air vents is 155 cm² and of passive stacks 500 cm².

If the passive stacks fullfill the new recommendations (Boverket 1994), the total cross section area is still 500 cm². The total outdoor air vent area has to be 175 cm² to arrive at a yearly average total ventilation of 0.5 ach (see table 4.3 and figure 4.10). If the outdoor air ventilation is to be higher than 0.5 ach during 75 % of the year the vents have to be at least 245 cm² (4 cm²/(m² floor area)) (see figure 4.11). At times the ventilation will be 4.3 ach.

Table 4.3 Calculated outdoor air ventilation for the apartment on third floor. The airtightness (excl. outdoor air vents) is 1 ach at 50 Pa.

Total passive stack area, cm ²	Total outdoor air vent area, cm ²	Pressure coefficient, passive stack	Yearly average $\pm$ variation, ach	Max, ach	Min, ach
500	155	-0,3	0,48 $\pm$ 0,37	3,78	0,0
500	245	-0,3	0,59 $\pm$ 0,39	4,26	0,0
500	385	-0,3	0,75 $\pm$ 0,43	5,04	0,0

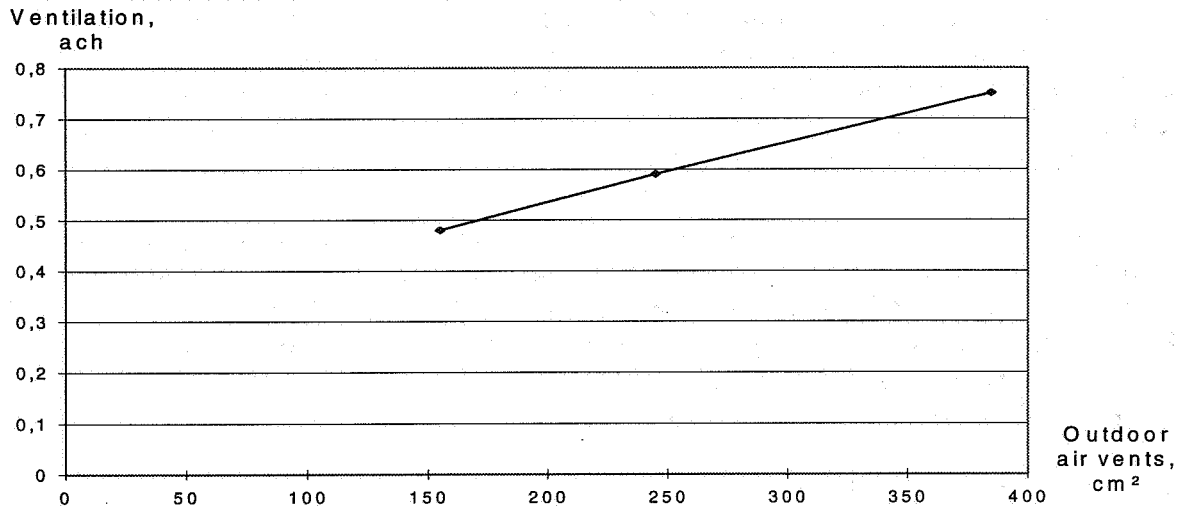


Figure 4.10 Yearly average total outdoor air ventilation as function of the total area of the outdoor air vents for the apartment on third floor with a total passive stack area of 1500 cm².

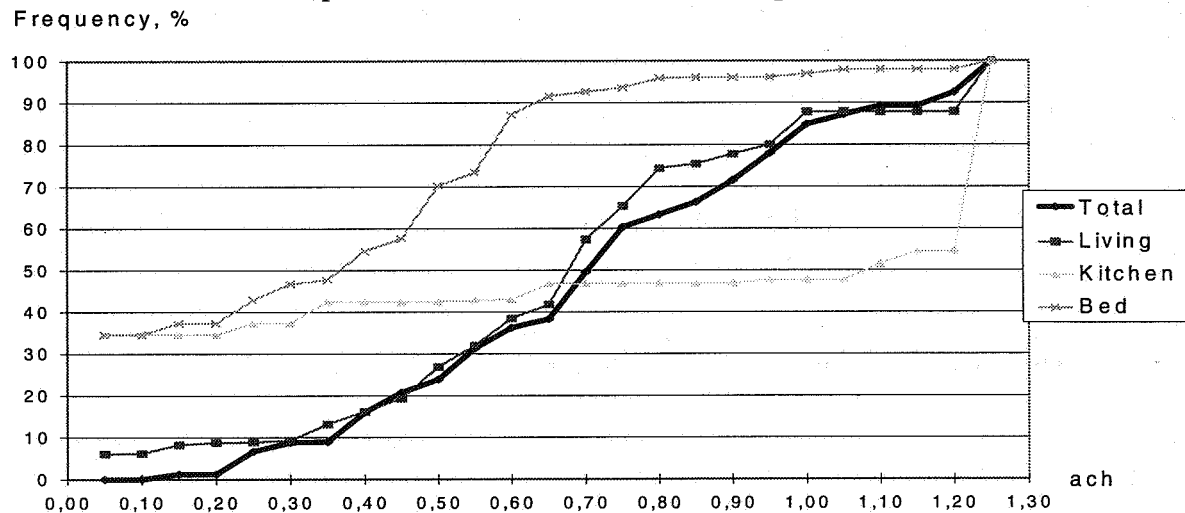


Figure 4.11 Cumulative distribution of outdoor air change rates in the apartment on third floor. The total area of outdoor air vents is 245 cm² and the total passive stack area 500 cm².

5. CONCLUSIONS

A conventional design of a passive stack ventilation system for a dwelling, which fulfills today's requirement on airtightness (3 ach for detached house and 1 ach for apartments at 50 Pa), result (according to calculations) in a dwelling, which some part of the year has too low an outdoor ventilation rate. During the winter the total outdoor air ventilation will be too

high. The variations in outdoor air ventilation rates during the year can be much larger for individual rooms than for the entire dwelling. In a 1½-storey house the rooms upstairs can during the winter have an outdoor air ventilation, which is very low.

Calculation for representative dwellings show that to arrive at a reasonable outdoor air ventilation e. g. that the total outdoor air ventilation rate for a dwelling is higher than 0.5 ach during 75 % of the year (reference year 1971 for Stockholm), a one-family house or an apartment should have outdoor air vents with an area somewhere between 2 and 7 cm²/(m² floor area). The total area of the passive stacks should be larger than the total area of the outdoor air vents. The area of the outdoor air vents depends on the type of dwelling, one-family house with one or two storeys or where the apartment is located in the building. This kind of dwelling will, however, during a major part of the winter have too high a ventilation rate i. e. some kind of control of the outdoor air vents and passive stacks is necessary. The ventilation is too low mainly during the summer. During this part of the year (25 %) the ventilation can be improved by airing and/or fan assistance.

The design of a passive stack ventilation system should be based on a dwelling, which fullfills today's requirement on airtightness and thereafter the size of the outdoor air vents and passive stacks is determined, in order to ensure that most of the ventilating air flows through the ventilation system and thereby is controllable. A modern system for passive stack ventilation must include a system for control of the air flows in order to avoid excessive use of energy and inadequate indoor air quality.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

**17th AIVC Conference, Gothenburg, Sweden,
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MAXIMUM VELOCITY OF RETURN FLOW CLOSE TO THE FLOOR IN A VENTILATED ROOM - EXPERIMENTAL AND NUMERICAL RESULTS

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Maximum velocity of return flow close to the floor in a ventilated room - experimental and numerical results

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Synopsis

The problem of sensation of draught in ventilated spaces is connected to inappropriate velocities in the occupied zone. In Scandinavia, velocities higher than 0.15 m/s are said to be an indicator of that occupants are likely to feel discomfort. Therefore knowledge of the flow field (both mean velocities and fluctuations) is necessary. Both experimental and numerical analysis of the flow field in a full scale room ventilated by a slot inlet, with two inlet Reynolds numbers 2440 and 7110, have been carried out. Results from both approaches show that the location of the maximum velocity near the floor is nearly independent of the Reynolds number. For a two-dimensional room, the maximum velocity at the floor level occurred at about 2/3 room length from the supply. The distance from the floor level is dependent on the inlet Reynolds number. The velocity profiles far away from the wall opposite to the inlet device have the same character as a wall jet profile. However, close to the corners they are transformed. The relative turbulence intensities measured in the return flow region are questionable, because of a hot wire's inability to record large fluctuations at low mean velocities. These turbulence intensities close to floor level vary from 15 to 80 % and as the authors have pointed out previously hot wires do not indicate the real value of the turbulence intensities beyond 20%. Difficulties appear in numerical predictions of return flow properties. Comparison between predicted values and experimentally obtained values show a reasonable agreement. This is promising for future CFD-predictions. However, there is a need for an appropriate measurement technique that can cope with reversing flow.

1 Introduction

The comfort parameters in a ventilated room are described in terms of appropriate temperatures, and the risk of draught. The latter is related to the velocities in the occupied zone. Existing design procedures require a velocity level below 0.15-0.20 m/s to avoid discomfort of occupants in the region of their activities. According to Fanger et al. (1988), high relative turbulence intensities will increase the sensation of draught. It is not easy to

achieve low turbulence levels and low velocity levels at the same time. Measurements are more difficult due to high turbulence intensities which are out of the range of sensitivity of frequently used hot wire anemometers, see Karimipناه & Sandberg (1994a,b). The air movements in a room result in complex processes involving dynamic, thermal, climatic, mechanical and structural parameters, see Jin & Ogilvie (1992a,b). Thus, a general conclusion it is not possible to draw. But, as mentioned before, one of the important comfort parameters to control is the velocity level and its fluctuations. Usually, the supply air terminal is close to the ceiling level where the velocity profiles are more or less like the traditional wall jet profiles. But the problem occurs when the jet impinges on the opposite wall and deflects to create the return flow above the floor. There, the low velocity levels cause difficulty in measurements.

This report is just one in a long series in this area of fluid mechanics. Hanzawa et al. (1987), Melikov et al. (1988, 1990) and Jin & Ogilvie (1992a,b) investigated air flow patterns in the occupied zone. All these efforts have contributed valuable information in this area but the problem due to its overall complexity still remains unsolved.

2 Experimental set-up

A full-scale test room with dimensions $6 \times 3 \times 3$ m ($L \times W \times H$) is used. The inlet height is 1 cm and spans the whole room width to ensure two-dimensionality. The outlet height is 4 times inlet height and the locations of inlet and outlet are shown in figure 1, see also Karimipناه & Sandberg (1994b) for more details. To measure the velocities at the floor level, 8 stations were chosen. The stations were located at $x=2.5, 3.0, 3.5, 4.0, 4.5, 5.0, 5.3$ and 5.6 [m] where x was measured from the supply inlet, see the co-ordinate system in figure 1. For each station, measurements were conducted from the floor up to $y=70$ cm using a single hot-wire. Both mean velocities and turbulence fluctuations were obtained for each point. The total measuring time for each point was 10 minutes. Two different supply velocities 3.72 m/s ($Re=2440$) and 10.82 m/s ($Re=7110$) were used to study to some extent the Reynolds number dependence.

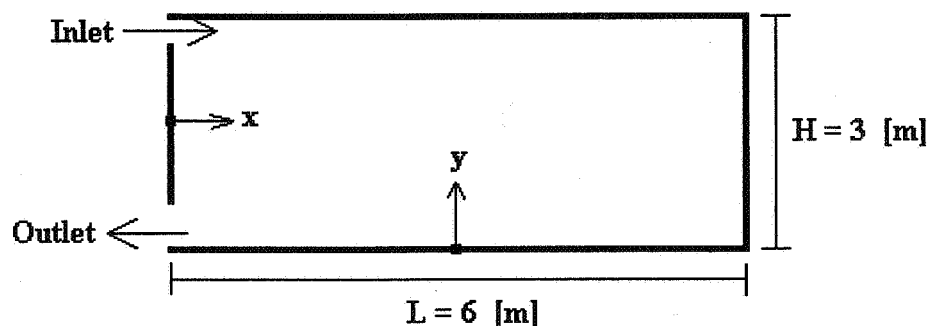


Figure 1 The co-ordinate system and definitions in measurements.

3 Results and discussion

The measured *mean floor velocity* U_r profiles compared to the predicted ones are shown in figure 2. In numerical predictions Launder's (1993) anisotropic version of $k-\epsilon$ turbulence model, see also Karimipناه & Sandberg (1996), was used. The results show a good agreement indicating the quality of both measured and predicted values. For both inlet velocities the maximum velocity at the floor level lies at $x=4.0$ m ($=0.33L$) from the wall opposite to the inlet. Jin & Ogilvie (1992a,b) also found the maximum velocity for a shorter room, $L=4.8$ m, at the location $x=0.3L$. This is in agreement with the findings of Nielsen (1989). One possible contribution to the small discrepancies between our measurements and those reported by Jin & Ogilvie may be that their outlet location lies a short distance under the supply inlet. Our profiles were almost the same as those of Jin & Ogilvie (1992a,b). Based on these results and other investigations mentioned, one may generalise the findings by concluding that the maximum velocity at the floor level of a two-dimensional ventilated room lies about $2/3 L$ from the supply. It is worth mentioning that in a three-dimensional room studied by Sandberg et al. (1991) exhibits a different behaviour. They found that the maximum velocity at the floor level occurs at $5/6 L$ from the supply device.

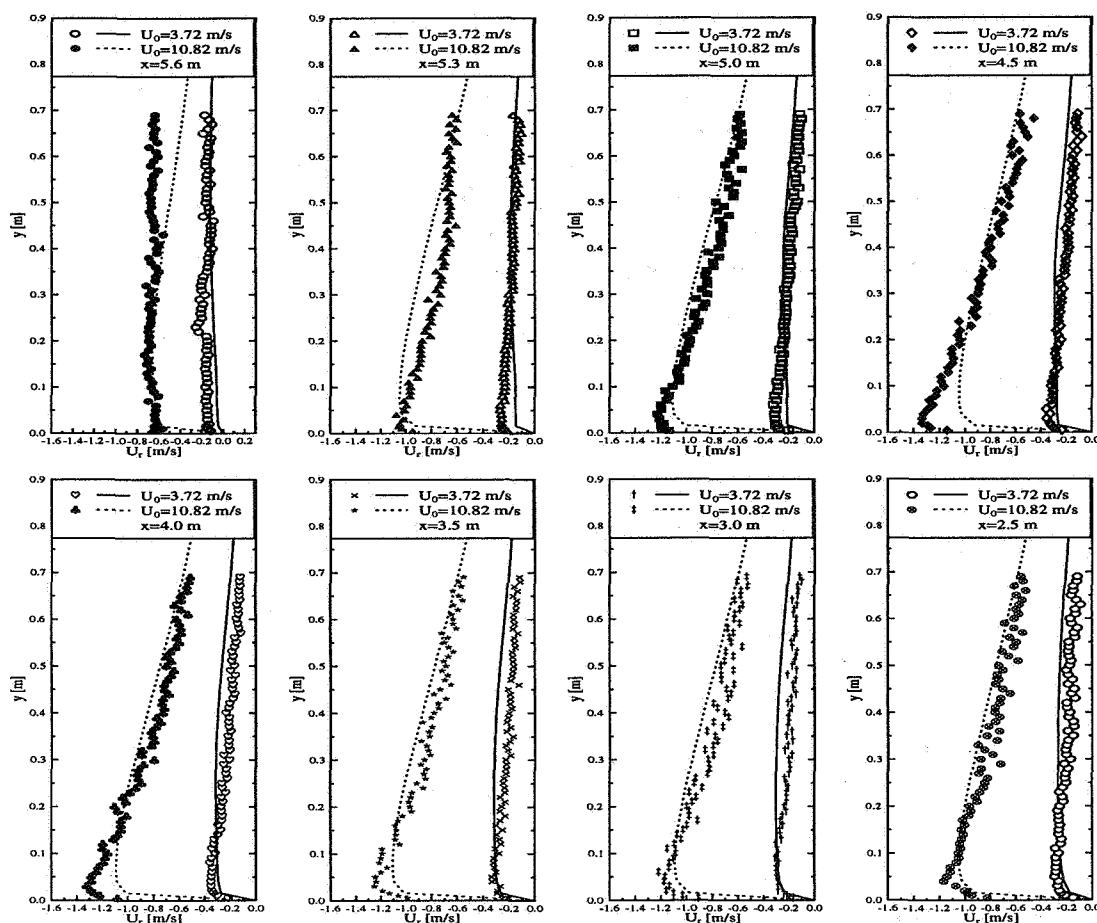


Figure 2 Measured (symbols) and predicted (lines) velocity profiles above floor level for different sections.

From figure 2 it is evident that the velocity profile close to the wall opposite to the supply ($x=5.6$ m) is unlike the wall jet profile, while the profiles at the other distances are almost similar to wall jet profiles. According to Nielsen (1989), the maximum velocity at the floor level, return flow velocity U_m , in normal cases is assumed to be the maximum velocity in the occupied zone. This maximum velocity in the occupied zone is proportional to the inlet velocity, inlet height, and the room length L . Nielsen used a relation between the maximum floor velocity and the velocity, U_L , extrapolated to the distance L from the supply for a wall jet in an "infinite" space as

$$\frac{U_m}{U_L} = K_m \quad (1)$$

where K_m was roughly estimated to be about 0.7.

In this paper we extrapolated U_L to room length based on two different situations (see table 1):

- a) A jet in an open room (only ceiling) of the same length as in Fig. 1.
- b) A jet in a closed room as in Fig. 1.

For case (a) we obtained a K_m of 0.7 and 0.92 respectively. For case (b) we obtained a K_m of 0.58 and 0.76 respectively. As Nielsen indicates this might be due to the geometry effect and based on these results also due to the Reynolds number effect. Nielsen (1989) uses a model room and here a full-scale room is used. The quotient (1) in case (b) are about 83% of that in case (a) for both Reynolds numbers. This is due to that the rate of decay of the velocity in the wall jet is less rapid in case (b), see Karimipناه (1996) and figure 3. It is worth mentioning that although the location of the maximum floor velocity shows very small Reynolds number dependence, see figure 3, the distance above the floor level is decreasing by increasing the Reynolds number. These distances were $y=3$ cm and 5 cm for $Re=7110$ and 2440 respectively. The following relation of Nielsen (1989) may be used with acceptable accuracy to calculate the maximum floor velocity

$$U_m = K_p K_m U_0 \sqrt{\frac{h}{L + x_0}} \quad [\text{m/s}] \quad (2)$$

where h is slot height, L is room length, x_0 is the jet virtual origin, U_0 is the supply inlet velocity and K_p and K_m are decay constants for ceiling level velocities and floor level velocities respectively.

To illustrate the turbulence characteristics the recorded rms values of the fluctuating velocities and the relative turbulence intensities, $I = \frac{U_r}{\sqrt{u'^2}} \times 100$, are shown in figures 4 and 5 where.

One can see from figure 4 that the values of the rms velocities are nearly constant in all locations, except for the distances $x=5.6$ m and 5.3 m. But the low mean velocities result in a high turbulence intensity of more than 70% which is outside the capability of hot wires. Jin & Ogilvie (1992a,b) argued that the maximum averaged turbulence intensity in their study was

less than 50% but they never commented on the quality of their measured mean velocities and the rms values some distances above the floor level.

With respect to the above mentioned difficulties one may accept that new measuring techniques are needed for the occupied zone in a ventilated room and until these have been found the problem of draught will be unsolved.

Table 1 Test conditions and U_L for open and closed room.

Conditions				Open room (a) Extrapolation to L		Closed room (b) Extrapolation to L	
U_0 [m/s]	Re	U_{rm} [m/s]	$\frac{U_{rm}}{U_0}$	U_L [m/s]	$\frac{U_{rm}}{U_L} = K_{rm}$	U_L [m/s]	$\frac{U_{rm}}{U_L} = K_{rm}$
10.82	7110	1.320	0.12	1.43	0.92	1.73	0.76
3.72	2440	0.345	0.092	0.49	0.70	0.59	0.58

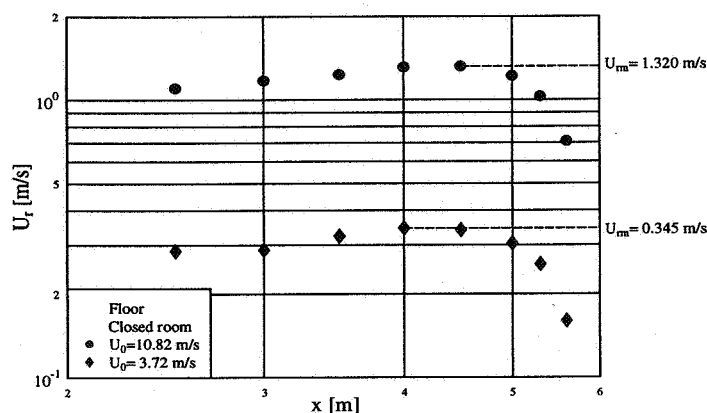
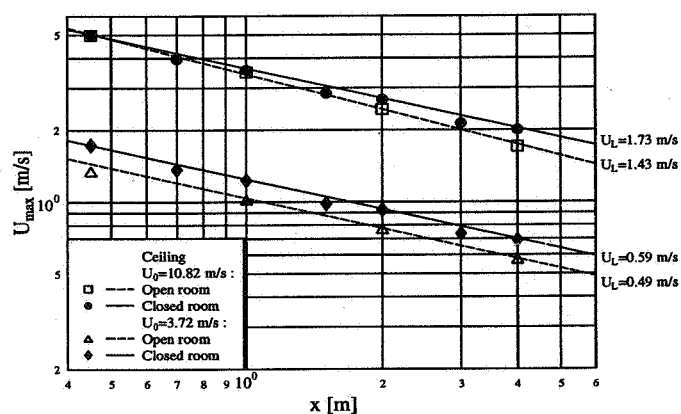


Figure 3 Maximum velocity for ceiling level (upper graph) and floor level (lower graph) respectively.

4 Concluding remarks

Regarding air movements in the occupied zone the following conclusions may be drawn:

1. According to these results and previous investigations, the location of the maximum floor velocity for a two-dimensional flow is at about $\frac{2}{3}L$ measured from the supply air terminal and is almost independent of the Reynolds number and room size.
2. The value of maximum velocities at the floor level, U_{rm} , is proportional to the velocity in the wall jet, U_L , at the end of the room. The proportionality constant is approximately 0.67 (but higher for higher inlet Reynolds numbers and lower for lower ones).
3. The effect of degree of room's confinement is shown by that the extrapolated jet velocity at the end of a closed room U_L is reduced by 83% compared to a wall jet in an open room.
4. Although the location of the maximum floor velocity shows some Reynolds number dependence, the distance above the floor level is decreasing with increasing Reynolds number.
5. By using a hot wire anemometer, the relative turbulence intensities vary from 15% up to more than 80%. This points out the difficulty of measuring the turbulence in the occupied zone with a hot wire. Thus, there is a need for a new measuring technique capable of measuring in the occupied zone.

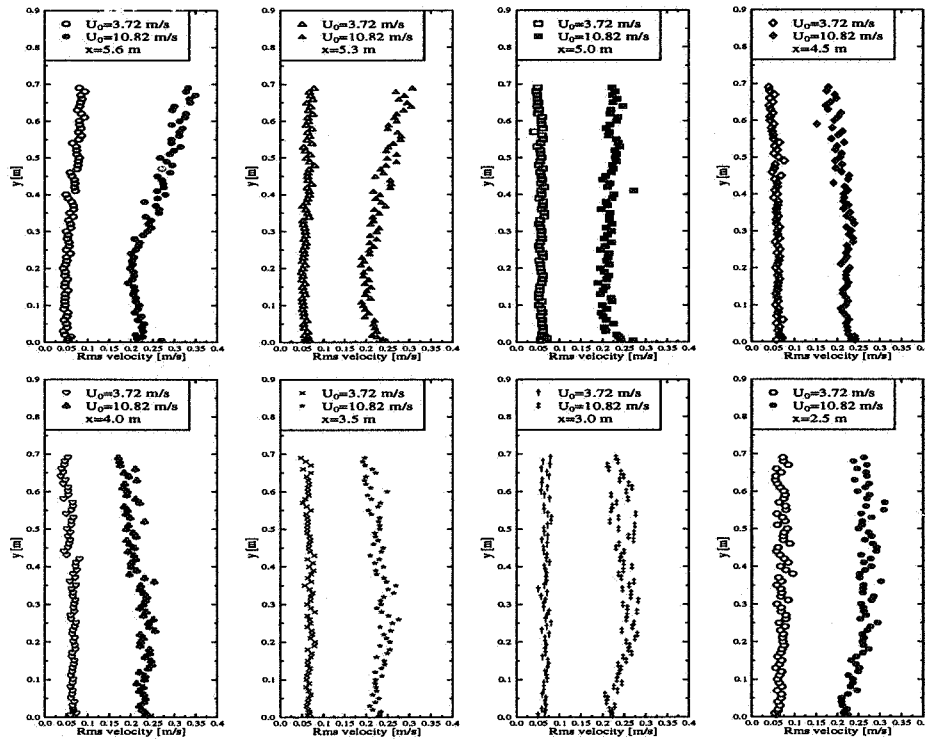


Figure 4 Measured rms velocity profiles above floor level for different sections.

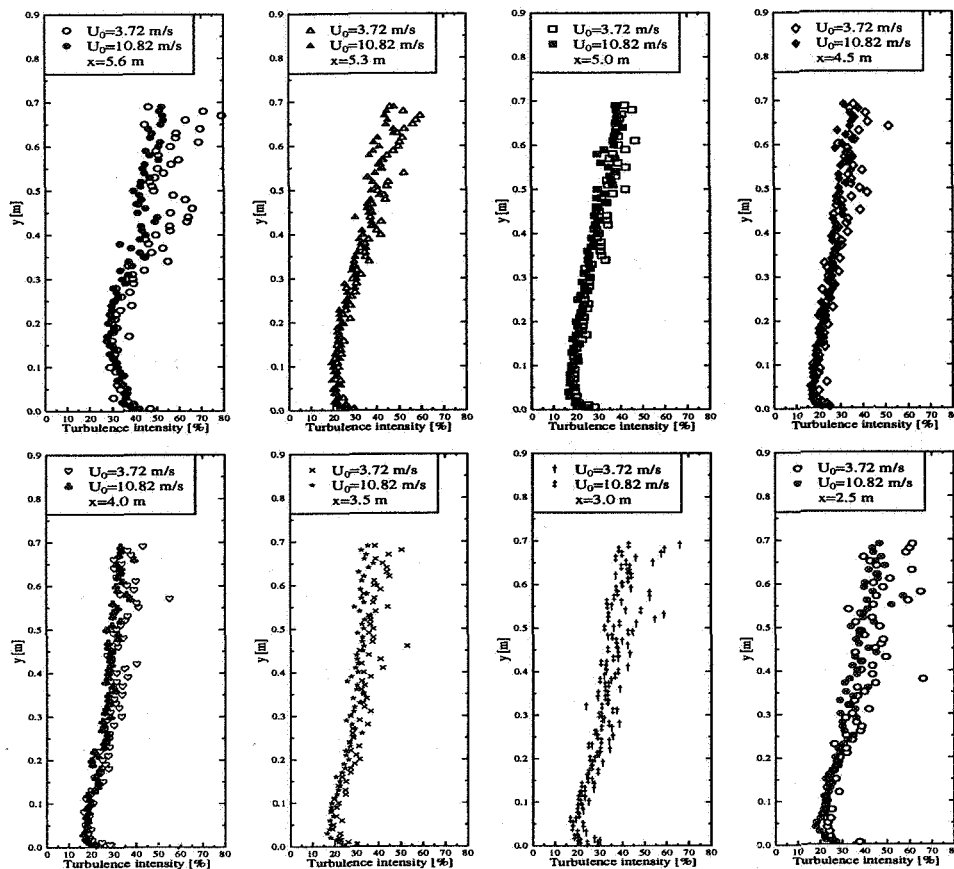


Figure 5 Measured relative turbulence intensities above floor level for different sections.

Acknowledgements

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

**17th AIVC Conference, Gotheburg, Sweden,
17-20 September ,1996**

(Title) **FULL-SCALE MEASURMENTS OF INDOOR AIR FLOW**

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SUMMARY

Full scale measurements of air flow velocities, temperature, intensity of turbulence and air exchange rate are carried out on two rooms with different type of ventilation located in the department of architecture at Chalmers University of Technology. The measurements have shown that *mixed* ventilation gives variable mean flow velocities with a high risk of draught as compared to the room provided with displacement ventilation. Air exchange rate for the room with *displacement* ventilation is obtained from tracer gas monitor by employing *decay* and *constant emission* methods. The measurements give similar results for both the methods.

1. INTRODUCTION

For probabilistic modelling of air quality in a room, access to full-scale data is a pre-requisite condition for describing the distribution of air flow velocities, intensity of turbulence, draught and temperature. With this in mind, the work described in this paper deals with full-scale measurements for two different types of ventilation systems. The first test object is provided with *mixed* ventilation and the second test room consists of *displacement* ventilation. Full-scale measurements of air flow velocities, intensity of turbulence, draught and temperature are carried out at 24 locations and five different heights. Measurement of air exchange rate is only carried out for test *room B*. The note presents measurements at two levels only, namely, 0.6 m and 1.4m height. Details of measurements at other levels are given in reference 1.

2. INSTRUMENTATION

Air flow velocities and temperatures are measured with constant temperature instrument developed by *SWEMA*. The instrument is designed for measurement of low velocities with high sensitivity and is based on the principle of measurement of convective flow caused by temperature difference. A total number of nine *SWEMA 300* anemometers are acquired and out of which eight are omni-directional and one is uni-directional.

The air exchange rate is measured with multigas monitor instrument developed by *Brüel & Kjaer (B&K)*. The instrument consists of a multipoint doser and a sampler which is operated from a desk top computer. The measuring system can be operated either by recording the decay of tracer gas over a length of period or as a continuous measuring device by providing extra supply of the gas to keep a constant level. Although the system is slow in time response, yet it is one of the most reliable measuring system available at the present time.

3. DESCRIPTION OF TEST ROOMS

Full-scale measurements of air flow velocities and temperature have been carried out for two rooms with identical construction but with different ventilating systems. The first object *room A* is ventilated with old type system of *mixed* ventilation, while the second object *room B* is recently fitted with ventilation based on the principle of *displaced* flow. Figure 1 shows the two rooms with locations of air inlets and outlets together with their dimensions and volume. It should be observed that the construction properties of the two rooms are identical and any differences in the air flow are due to functioning of the ventilating systems.

Location of the measurement points is shown in figure 2.

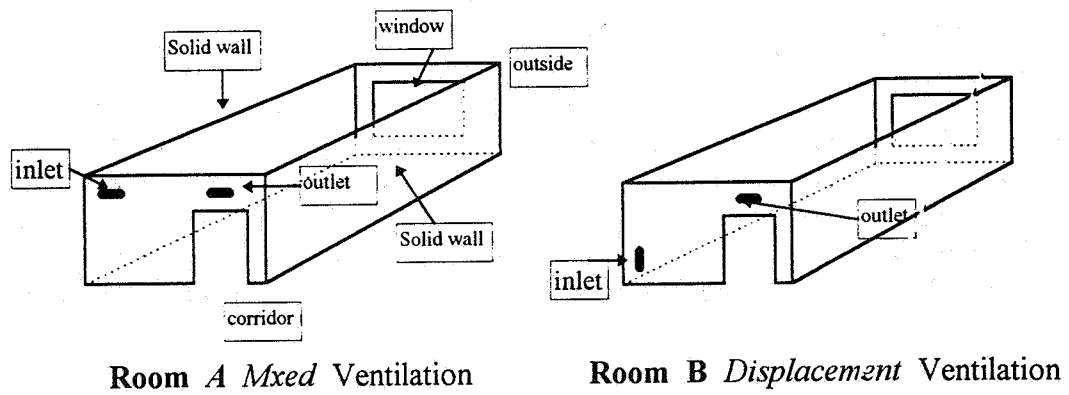


Figure 1 Test rooms with *Displacement* and *Mixed* Ventilation Systems

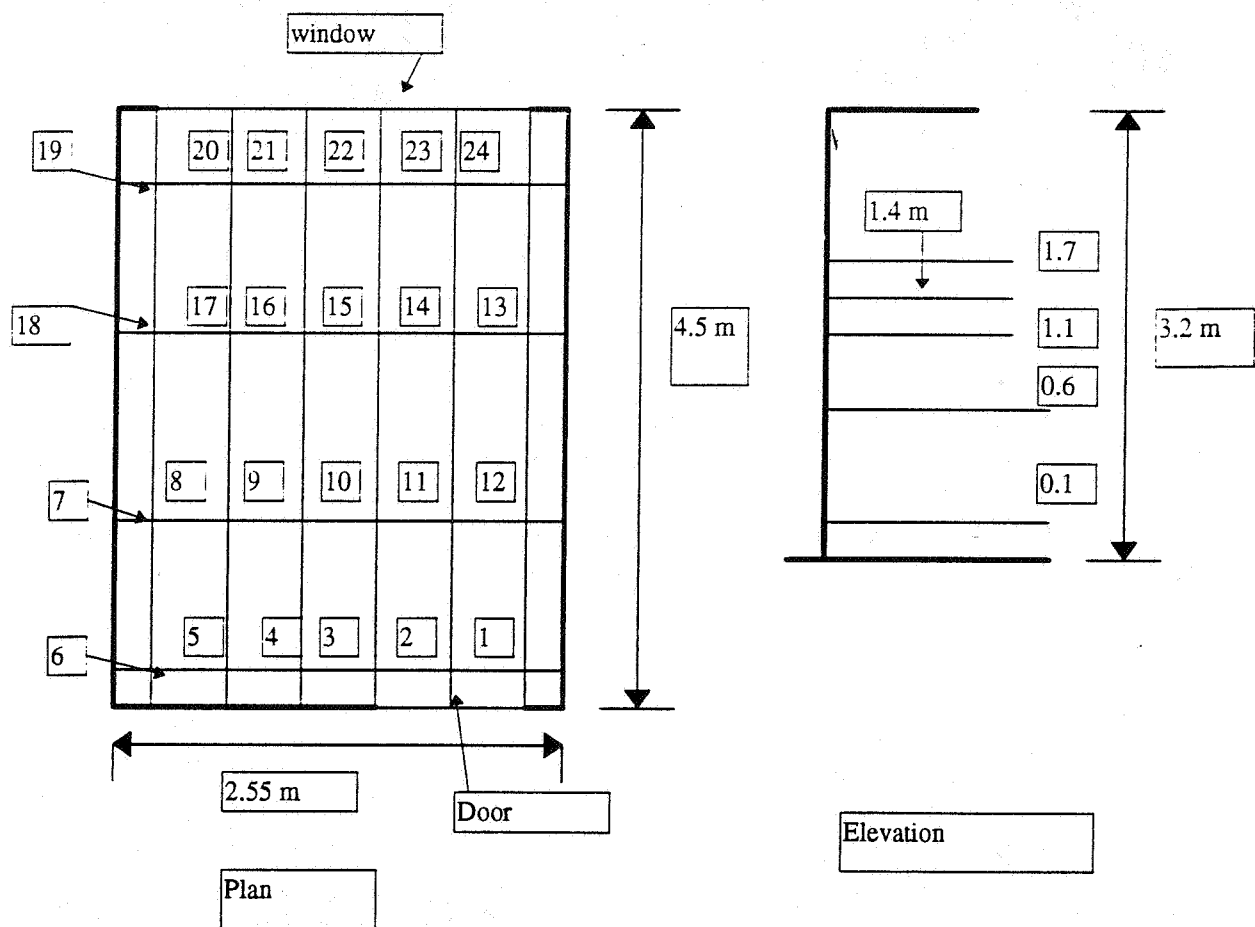


Figure 2 Location of measurement points at different levels

4. MEASUREMENTS OF AIR FLOW VELOCITIES, INTENSITY OF TURBULENCE, AND DRAUGHT

4.1 Distribution of Air Flow

Ventilation in test *room A* is of mixed type with the inlet and outlet located near the ceiling as shown in figure. A three-dimensional view of the flow for levels 0.4m and 1.6 m is shown in figure 3. The plot shows that the air flow velocities are generally higher near the window which is further away from the ventilation inlet and low near the entrance. The measurements have shown that the mean air flow varies considerably from location to location and lies between 5 and 15 cm/s. The range for minimum and maximum velocities was from 0 to 30 cm/s for all levels.

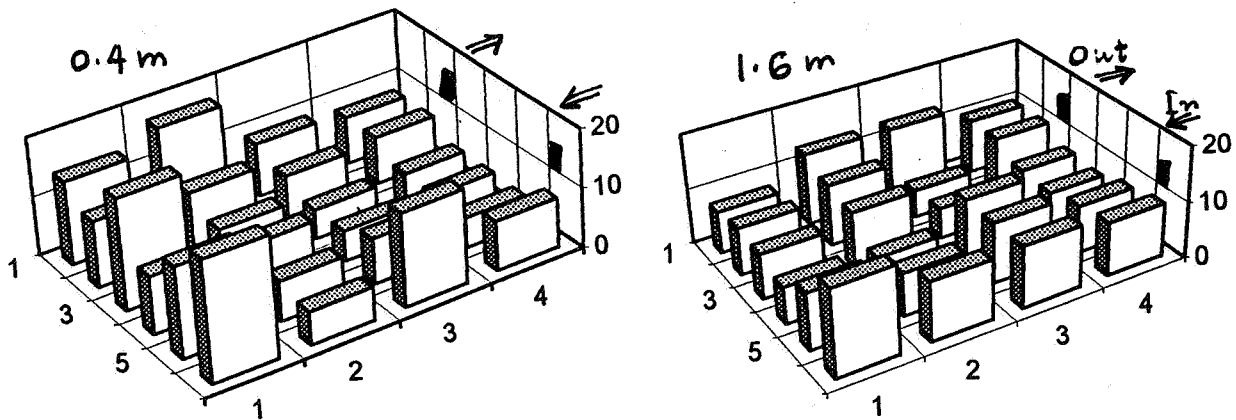


Figure 3: Three-dimensional View of Flow for *Mixed Ventilation (Room A)*

Ventilation in *room B* is of *displacement* type with inlet located near the floor and the outlet at the top corner at a distance of 10 cm from the ceiling. The details are shown in figure 2. In both test rooms, all the measurement points have been selected with due consideration given to their similarity in position.

Three-dimensional plots of the flow velocities are shown in figure 4. A general impression from these figures is that the flow is not varying as opposed to the case for *room A*.

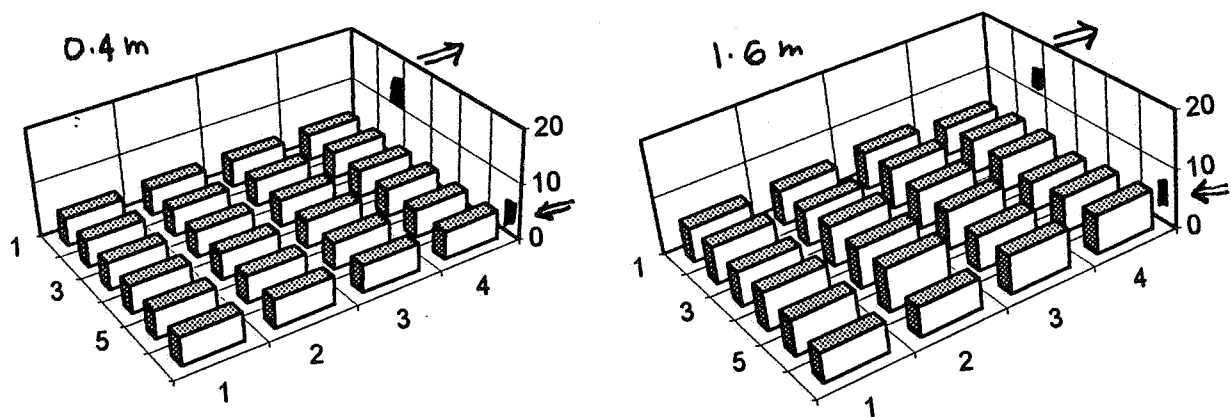


Figure 4: Three -dimensional View of Flow for *Displacement* ventilation (*Room B*)

4.2 Intensity of Turbulence

The air flow in room fluctuates randomly and is generally expressed as a ratio of standard deviation σ and mean flow velocity v . Larger the intensity of turbulence, greater is risk of draught. For the case of measurements of intensity of turbulence for *mixed* ventilation systems, I_V is approximately 20% as shown in figure 5. This is valid for all levels except at certain locations where it can be about 40%.

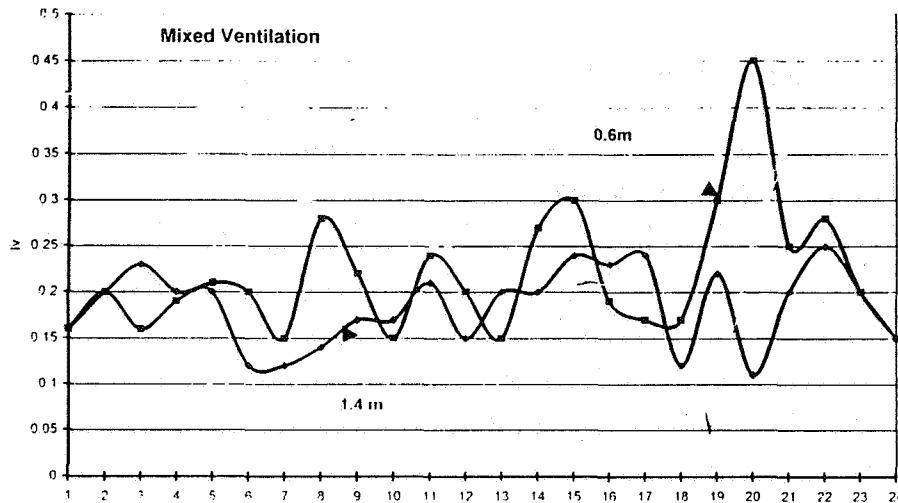


Figure 5: Intensity of Turbulence for *Mixed Ventilation (Room A)*

For the case of *displacement* ventilation, the intensity of turbulence I_V varies considerably from point to point in the room as shown in figure 6. It lies between 20–40% depending on the height of the measuring point.

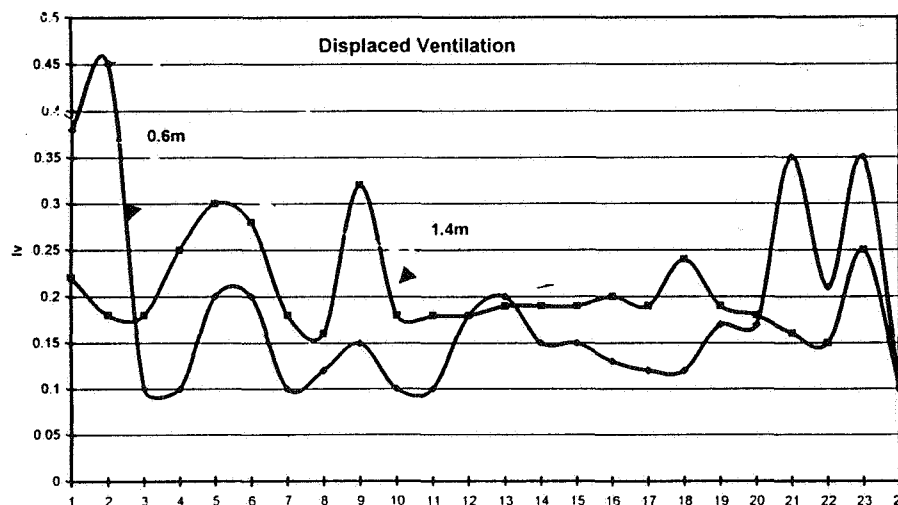


Figure 6: Intensity of Turbulence for *Displacement Ventilation (Room B)*

4.4 Risk of Draught

Sensation of draught is dependent not only on the mean flow velocity and temperature but also on the intensity of turbulence. Reference 2 gives an expression for calculating the percentage of dissatisfied inhabitants which can be written as:

$$PD = \{34 - t_a\} \{0.01 v - 0.05\}^{0.623} \{0.00369 \sigma + 3.143\}$$

where $v \geq 5$ cm/s

t_a - temperature inside the room in °C

v - mean flow velocity cm/s

σ - standard deviation of velocity

It can be noted that the risk of draught PD increases with increase in the mean flow velocity and intensity of turbulence. Figure 7 shows the risk of draught as a percentage of dis-satisfied people for all the measured points at different levels. It is approximately 8% and in some cases it approaches to 15% in *mixed* ventilation

For the case of *displacement* ventilation, the intensity of turbulence I_v varies considerably from point to point in the room as shown in figure 5. It lies between 20–40% depending on the height of the measuring point. However, these variations have insignificant effect on the risk of draught as the mean flow velocity is rather low (approximately 5 cm/s).

It should be mentioned that the sensation of draught is subjective and varies from person to person and also the above formula is approximativ

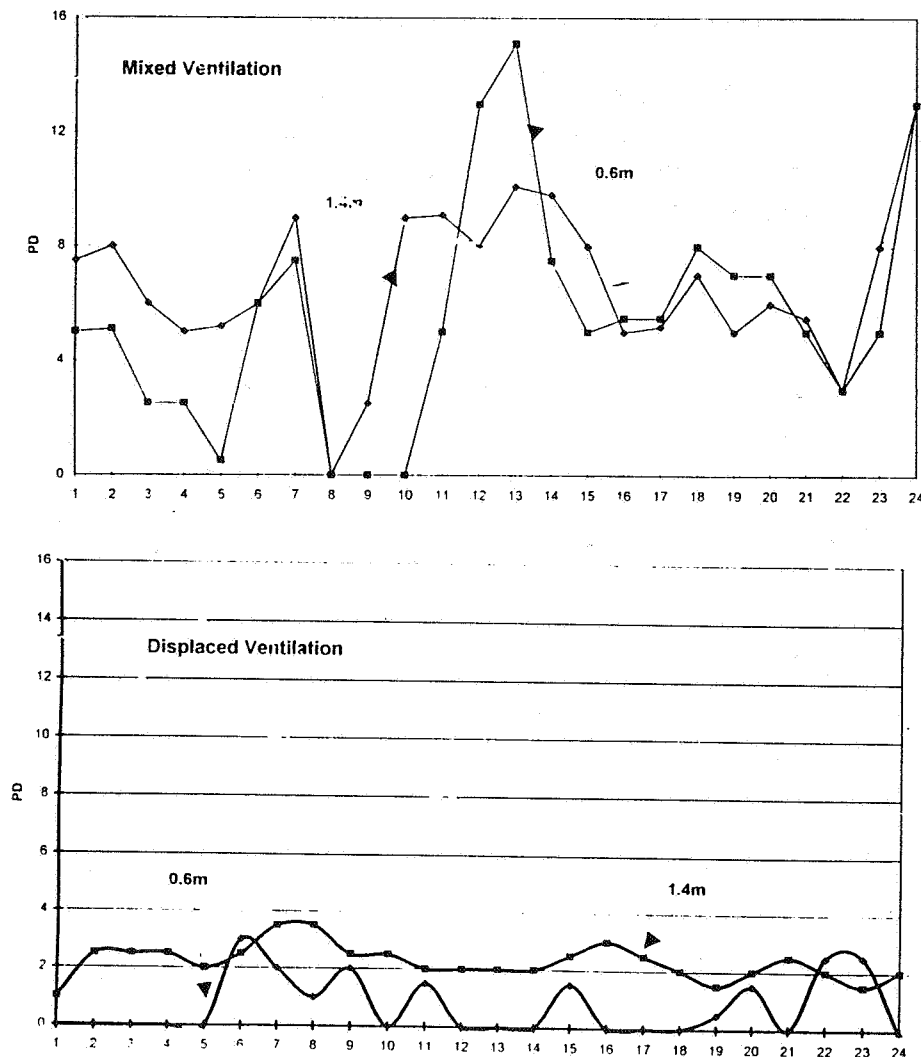


Figure 7: Draught in Room A and Room B

5. MEASUREMENT OF AIR EXCHANGE RATE FOR ROOM "B"

Air exchange rate for test room B is measured at six locations and four different levels as shown in figure 8. Locations 1 and 2 are situated near the inlet and outlet respectively. The other measuring points are related to the flow in the middle and near the window

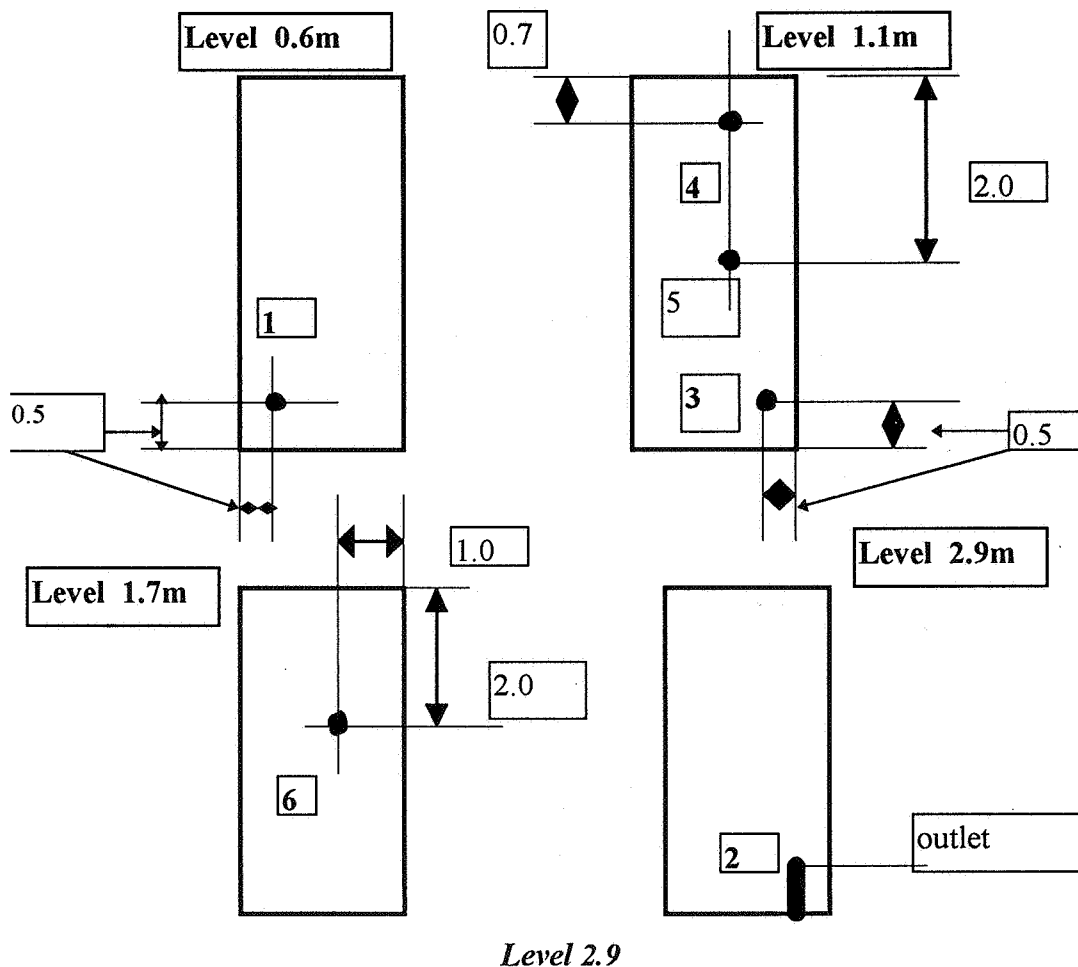


Figure 8: Location of Measuring Points

Two different methods are employed for studying the air exchange rate in the room. The first method is based on the principle of decay of the concentration of the injected gas. Figure 9 shows the curves for the decay of the tracer gas for various locations.

An average value of 1.8 change/hour is obtained for all the points. Minimum value needed to fulfil the hygienic criteria given in the Swedish Standard amounts to 0.7 change/hour based on the volume of the room. It implies that the criteria for hygienic conditions is satisfactorily fulfilled by the displacement ventilation.

The second method for measuring the air exchange and air flow rate is to use constant emission method. In this method, the tracer gas is emitted at a constant rate for the duration of the measurement period. Figure 10 shows the results of measurements for all locations and it can be noted that all points have the same amount of concentration of gas after an initial period of one hour. Values of air exchange rate are automatically calculated by the *Brüel & Kjaer* (B&K) measuring system and amounts to 1.9 change/hour.

Age-of-air and air-exchange efficiency are obtained from measurements with the decay and the constant emission method. Table 1 presents the results based on the calculations performed by (B&K) computer programme on the measured data. It can be noted that both methods give

similar results for air exchange efficiency, local and room average age-of-air. However, constant emission method shows larger variation in air exchange efficiency between measurement points 1 and 4 as compared to decay method. The measurements show that the efficiency of the ventilation is over 50%.

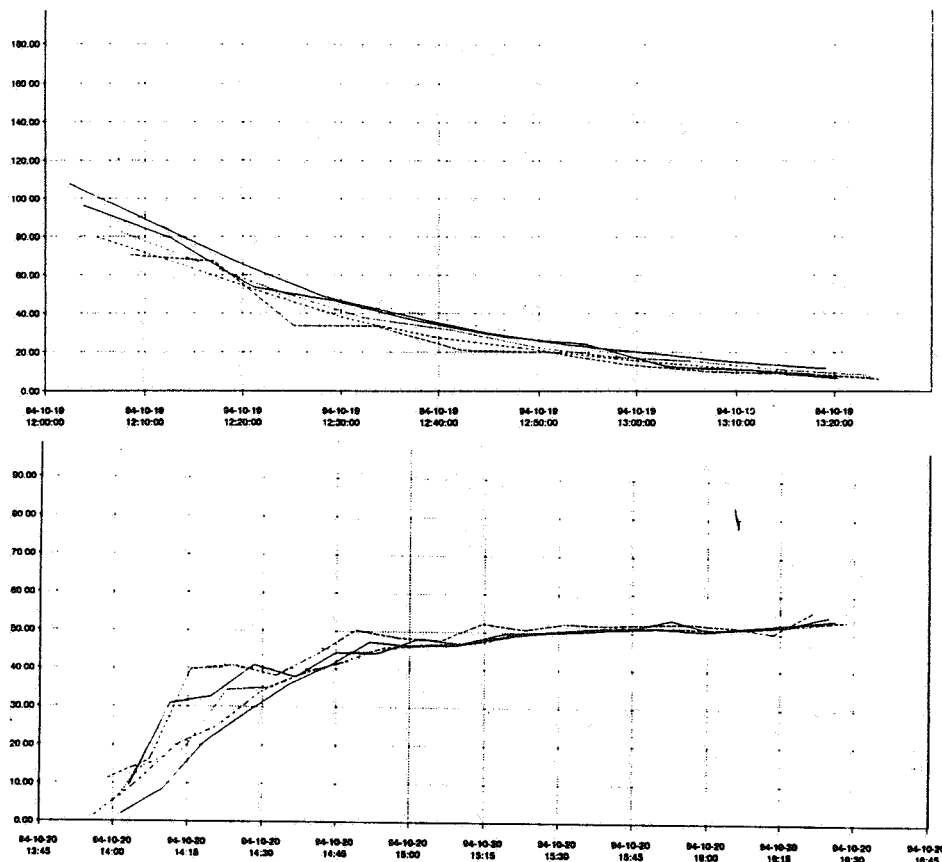


Figure 10: ACH based on *Decay* and *Constant* tracer gas methods

LOCATION	HEIGHT ABOVE THE FLOOR (m)	AIR EXCHANGE EFFICIENCY %		LOCAL AGE OF AIR (MINUTES)		ROOM AVERAGE AGE OF AIR (MINUTES)	
		Decay	Constant Emission	Decay	Constant Emission	Decay	Constant Emission
1	0.6	63	68	42.8	38.8	33.9	28.2
2	2.9	60	64	42.0	46.6	35.1	36.3
3	1.1	60	68	43.0	41.7	35.7	32.0
4	1.1	58	52	38.7	32.4	33.0	31.0
5	1.1	60	58	40.5	32.4	33.7	27.5
6	1.7	57	67	43.3	42.8	38.0	32.2

Table 1: Air Exchange Efficiency at different Locations

6. DISCUSSION OF RESULTS

Air flow velocity, intensity of turbulence and risk of draught are measured in two different rooms, identical in construction but with different ventilation systems.

The mean air flow is approximately 5 cm/s for the *displacement* ventilation and the air flow is evenly distributed over all levels and the risk of draught is minimal due to low flow velocities. In the case of *mixed* ventilation, the flow velocities over the whole room behave in an erratic manner. The flow velocities vary considerably from point to point and are much larger near the window and in the direction of the outlet. The risk of draught for the user is much higher for the mixed ventilation system as compared to displacement ventilation.

A comparison of results for two test rooms in figure 11 shows that the *displacement* ventilation provides air flow with low velocities and subsequently low risk of draught.

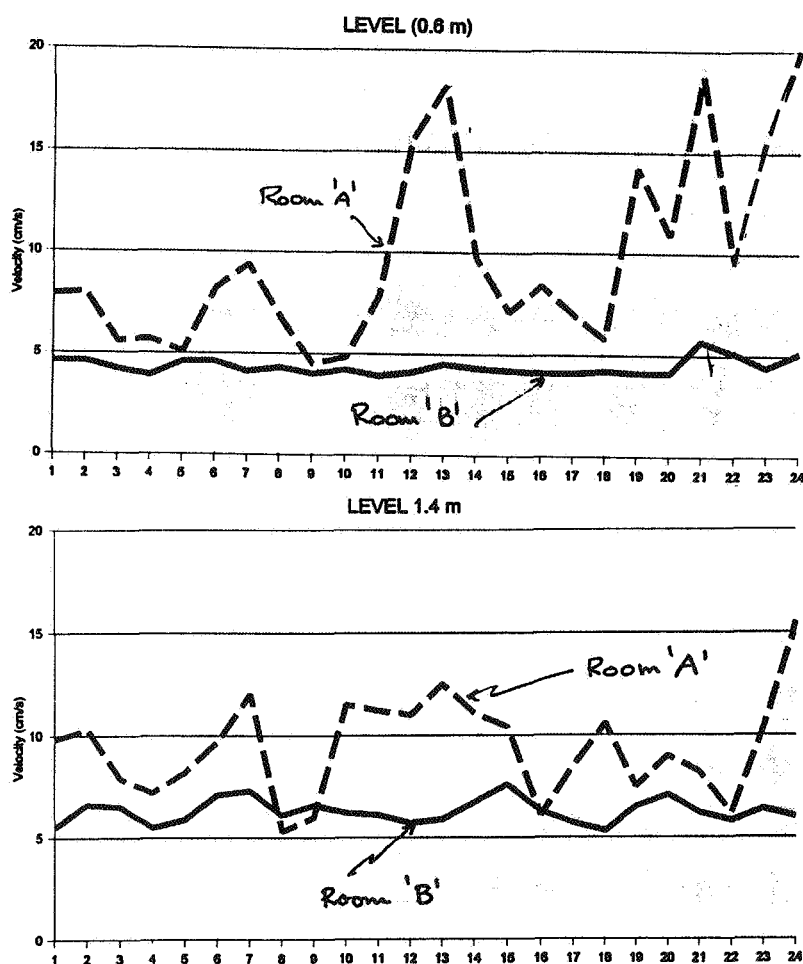


Figure 11: Comparison of Air Flow Velocities in *Room A* and *Room B*

Measurements of air exchange rate are carried out with the help of tracer gas monitor manufactured by *Brüel & Kjaer (B&K)*. Two different methods, namely, decay and constant emission are used in measuring the air exchange rate. It is found that the two techniques give similar results. Age of air and ventilation efficiency are also obtained from the measurements.

7. ACKNOWLEDGEMENT

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

APPLICATION OF A PASSIVE TRACER GAS TECHNIQUE IN NATURALLY AND MECHANICALLY VENTILATED SCHOOL BUILDINGS.

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Application of a passive tracer gas technique in naturally and mechanically ventilated school buildings.

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SYNOPSIS

The homogeneous emission passive tracer gas technique is described. This technique relies on an even distribution of constant tracer gas emission rate within the object to be measured, so that the emission rate per volume unit is constant. The local steady state concentration of the tracer gas is directly proportional to the local mean age of air and the emission rate per volume unit.

The technique was applied in 10 school buildings, of which 4 were newly constructed naturally ventilated buildings, 2 were old naturally ventilated buildings, 2 were old mechanically ventilated buildings and 2 were newly constructed supply and exhaust ventilated buildings.

The result shows that the ventilation rates in naturally ventilated schools are relatively low compared to required standards. It demonstrates the importance of not only rely on the naturally induced ventilation rate, but also to use other means for improving the air quality during or between lessons, like window opening.

1. BACKGROUND

There is much concern about the air quality in school buildings in Sweden. One reason for this is the rapidly increase in the number of school children suffering from allergy and other hypersensitivity. Another reason is that the air quality often is quoted unsatisfactory by pupils and teachers. Many school buildings are therefore now retrofitted with new, more efficient ventilation systems. Another trend is to try "natural" ventilation principles even in newly constructed school buildings.

The most popular technique to estimate the air quality in schools, is to perform measurement of the carbon dioxide concentration during lessons. At present, regulations in Sweden, state that the carbon dioxide concentration in rooms, where the major part of contaminants emanates from people, must not exceed 1000 ppm on a regular basis. In a room, which is only ventilated with outside air, this means that the flow rate of supplied outside air must be 7-8 litres per second and person.

However, it is difficult to draw conclusions about the ventilation rate and the distribution of ventilation air in a school building only from measurement of carbon dioxide concentration. There are a number of reasons for that:

- Carbon dioxide concentration tend to be very fluctuating in time
- Concentration varies in space - measurement position is critical
- Steady state concentration is often not attained during a lesson
- Emission rate per person is uncertain and varying
- Window and door opening influences the dilution of carbon dioxide
- Equipment cost prevents simultaneous measurement in many rooms

In order to evaluate the ventilation rate and air distribution in school buildings as offered by the ventilation system and the building itself, measurements were carried out with a passive tracer gas technique during an extended time. Thus the result is not necessarily typically for periods of lessons, when the ventilation rate can be appreciably enhanced due to window opening and other circumstances.

2. THE LOCAL MEAN AGE OF AIR

A convenient choice of ventilation parameter to describe the air distribution patterns is to use the concept of "local mean age of air". Until recently it has been difficult to map the distribution of local mean ages of air in large enclosures and other large premises. The "homogeneous emission technique" is a relatively new tracer gas technique which allows a detailed mapping of local mean ages of air also in large premises.

The mean age of air has a close connection to "classical" ventilation quantities such as *Specific Ventilation Flow Rate*, often known as "*Air Change Rate*" (ACH). The mean age of air leaving a building ($\bar{\tau}_n$) is equal to the inverted value of the *Specific Ventilation Flow Rate*.

3. MEASUREMENT TECHNIQUES

3.1 Conventional Techniques

The local mean age of air can only be measured using tracer gas. In the *decay* technique, all air in the ventilated system is initially be marked to the same concentration ($C(0)$) with tracer gas. By following the dilution process $C(\tau)/C(0)$ as a function of time and integrating this function from $\tau=0$ to $\tau=\infty$ the average age of air at the measurement point is obtained.

There are several difficulties in the practical application of the decay technique in large buildings and buildings with many rooms. The most obvious one is that it is very difficult to achieve the necessary uniform initial concentration of tracer gas. Other difficulties are connected to sampling and analysing air from many positions in a large building

There is another important technique called the *step-up* technique, which however only can be used in mechanically ventilated buildings with a central air supply duct. In this case the tracer gas is injected at a constant rate in the supply duct and the local build-up curves of concentration are recorded. The technique is equivalent to the reverse of the decay technique and a necessary condition is that the final concentration is equal everywhere in the system. Any diluting air entering from e.g. infiltration will contribute to errors.

3.2 Homogeneous Emission Technique

Determination of the local mean age of air using the *homogeneous emission technique* relies on the fact that the local steady state concentration of a contaminant which is homogeneously emitted in a space is directly proportional to the local mean age of air. Homogeneous emission means that the source is continuously distributed in the space with equal emission rate per volume unit everywhere. This relationship has been shown by Sandberg (1984), and is also mentioned in the introduction to a NORDTEST (1988) method to measure the local mean age of air, but has not been utilised for development of any measurement technique until recently (Stymne *et al.* 1992, Stymne & Boman 1994, Walker *et al.* 1994, Stymne 1995).

A standardised method for measurement of local mean ages of air in large buildings and buildings with many rooms using the homogeneous emission technique has recently been proposed to NORDTEST (Stymne 1995).

3.2.1 Description of the homogeneous emission technique.

The basic principle of the homogeneous emission tracer gas technique is to inject tracer gas at a constant rate in all parts of a zone divided building, so that the tracer injection rate in each zone is proportional to the zone volumes. The local steady state concentration (C_p) will be equal to the tracer injection rate per volume (S/V) multiplied with the local mean age of air (τ_p). Therefore the local mean age of air can be obtained from a measured local tracer concentration.

$$\tau_p = \frac{C_p}{S/V} \quad (1)$$

The proposed NORDTEST-method (Stymne 1995) to measure local mean ages of air involves a technique to distribute inexpensive so called *passive tracer gas sources* in the building to be investigated, in order to approximate a homogeneous emission. In multi-room systems a natural choice of zones, is to consider a room of normal size as one zone, while larger rooms must be further sub-divided into several zones. In each zone a tracer gas source is positioned. In practice it is in most cases not possible to make all zones equally large due to varying room sizes. Therefore, the development of adjustable passive tracer gas sources was an important task of the technique. In the present investigation adjustable passive tracer gas sources of capillary type emitting $1.2 \cdot 10^{-5}$ g/h of a perfluorocarbon compound (PFT) were used.

The measurement of local concentrations can be performed either using pumped sampling or "passive" sampling. Both rely on adsorption of tracer gas on a charcoal sampling tube, for later analysis with gas chromatography at a specialised laboratory. In the present work passive sampling with diffusion tubes was performed.

The air sampling is performed only in those spaces, where it is interesting to know the ventilation, while tracer gas emission must be made in the whole ventilated system.

Validation of the homogeneous emission technique using adjustable passive tracer gas sources has been performed in a single family house (Stymne and Boman 1994).

One main advantage with the homogeneous emission technique, using passive tracer gas emission is that it can easily be applied also in large buildings at a moderate expense, where other techniques are inconvenient and expensive. The technique has recently been applied in 4 relatively large objects of 1200 m³ -13500 m³ (Stymne and Boman 1996).

4. DESCRIPTION OF THE SCHOOL BUILDINGS

10 different school buildings as indicated in the TABLE 1 below were investigated. The buildings are marked with Nn1-Nn4 for newly constructed naturally ventilated buildings, No1-No2 for older naturally ventilated schools, Mo1-Mo2 for older mechanically exhaust ventilated buildings and Mn1-Mn2 for newly constructed exhaust and supply ventilated buildings.

Most buildings are equipped with mechanical extract from the rest rooms.

TABLE 1. Some characteristics of the investigated school buildings.

Notation	year of construction /year of retrofit	ventilation system	possibility of forced ventilation *	total volume m ³	number of class/ group rooms	number of floor levels
Nn1	1993	naturally	m/c fan	850	4	1
Nn2	1992	naturally	m/c window	1300	6	1
Nn3	1993	naturally	a/c duct	2100	9	1
Nn4	1994	naturally	m/c window	2300	11	1
No1	~1910/1993	naturally	a/c fan	5000	18	3
No2	1925/1995	naturally	a/c fan	6000	32	4
Mo1	1932/1993	exhaust		1600	8	2
Mo2	1940/1993	exhaust		1700	19	3
Mn1	1995	exh./suppl.		350	6	1
Mn2	1995	exh./suppl.		1300	8	1

*m/c = manually control, a/c=automatic control (temp. or CO₂). Mechanically ventilated systems are run at a 24-hour basis during the experiment.

5. DESIGN OF THE INVESTIGATION

In all school buildings the whole building space was investigated. In each building a "reference" volume corresponding to the smallest room of importance was decided on. Closets and other small spaces was not treated as separate zones but was normally included in the volume of a larger adjacent zone. The reference volumes varied between 15 m³ and 120 m³ in the different buildings. The number of reference volumes of each room decided the number and adjustment of tracer gas sources. Two reference volumes corresponds to one un-adjusted source. The number of tracer gas sources in a room varied from one to six. The total number of tracer gas sources in a building varied from 14 in the smallest school (334 m³) to 94 in the largest school (6000 m³). The homogeneous emission rate varied from 0.1µg/h,m³ to 0.7 µg/h,m³ in the different buildings.

The sources were normally mounted 2 cm from the walls at a height of 1.7 m above the floor. If there were several tracer gas sources (e. g. in large class rooms) they were evenly distributed in the room space.

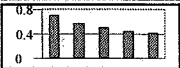
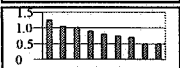
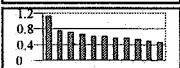
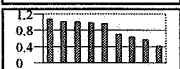
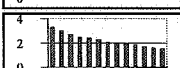
The average room temperature was recorded in all class rooms and in other spaces of importance. Passive samplers were distributed in the important occupational zones. Usually, two or more samplers were positioned in each class room - one always at a central position and sometimes at the teachers desk, at a pupils desk or close to an exhaust air device. The sampling tubes were continuously open during the measurement period, day and night and weekends.

In the naturally ventilated school buildings any possibilities to force ventilation during school hours (e. g. through window airing) could be used without restrictions during the measurement period. In the mechanically ventilated buildings, however, where the ventilation system normally are operated at a reduced level during "off-school" hours, the ventilation was continuously running (on a 24 hours basis) at day-time mode during the whole measurement period.

6. RESULTS

A summary of the results of measurements of the ventilation flow rates and the air distribution is given in TABLE 2.

TABLE 2. Results from the measurement

Notation	measurement time [h]	total ventilation rate [m ³ /h]	total volume m ³	specific flow rate [ACH]	distribution of ACH
Nn1	672	428	850	0.50	
Nn2	1040	592	1300	0.47	
Nn3	575	1379	2100	0.66	
Nn4	670	1348	2300	0.58	
No1	761	3630	5000	0.73	
No2	190	1769	6000	0.29	
Mo1	1368	2025	1600	1.25	
Mo2	1367	3654	1700	2.13	
Mn1	334	938	350	2.75	
Mn2	360	4885	1300	3.86	

The primary result of the measurement yields the local mean age of air in the different locations. However, in TABLE 2 we have chosen to present the results as the inverted values of

the local mean ages of air - here called "ACH", though the ACH-concept (Air Change Rate per Hour) is not as well defined for a local space as the mean age of air. Strictly, the inverse of the local mean age of air is equal to ACH, only in the exhaust air.

The column showing the specific flow rate refers values computed from the inverse of a volume-weighted average of local mean ages over the whole building space, while the column "distribution of ACH" only displays values obtained from a selection of class rooms and other important study-rooms.

7. DISCUSSION

7.1 Ventilation rate

The result shows that the ventilation rates in naturally ventilated schools are relatively low compared to required standards. It demonstrates the importance of not only to rely on the naturally induced ventilation rate, but also to use other means for improving the air quality during or between lessons, like window opening.

7.2 Air distribution

The result also shows that the measured air distribution is uneven in most of the schools. This result stresses the importance on not only to measure the total ventilation rate or an average of the air change rate in a building, but also to measure local ventilation parameters, like the local mean age of air.

The technique which is applied in this work - the homogeneous emission technique, has the advantage over most other constant emission techniques, in that it can yield a true picture of the air distribution. Using a well designed homogeneous emission technique, the differences in the steady state concentration of tracer gas at different locations are not caused by an uneven injection of tracer gas. Instead it reflects an uneven distribution of ventilation air. The reliability of the result is mainly depending on the success of achieving a homogeneous distribution of tracer gas emission rate in the whole building.

By the use of adjustable tracer gas sources and rather small subdivisions of room volumes the differences in the homogeneity of tracer gas emission rate was kept rather low, seldom exceeding 10%.

7.3 Measurement uncertainty

In addition to the uncertainty of measured local mean ages of air caused by incomplete homogeneity of tracer gas emission rate there are uncertainties due to *a)* variations in the emission rate of tracer gas sources (~5%), *b)* variations in uptake rate of samplers (~5%), and *c)* analysis uncertainty (~3%). By using several sources and samplers the uncertainty of the mean value decreases. A systematic error due to the difficulties of performing exact calibrations of uptake rate, emission rate and analysis can not be excluded, but is believed to be less than 5%. In the present case, an estimate of the total uncertainty of a local value is 15%, while the uncertainty of mean values of the whole building is estimated to be less than 10%.

7.4 Remarks on the effect of measurement time

The purpose of this investigation was to measure the ventilation performance of the buildings, during an extended time, in order to cancel out short term variations of the ventilation rate due to occupant habits, like window opening and other means of affecting the ventilation. The buildings were occupied approximately 20-25% of the measurement time. The measured average ventilation rate was therefore only marginally affected due to measures taken during the periods of occupation. The local mean age of air is computed from the average concentration of tracer gas during the measurement time. The instantaneous concentration is approximately inversely proportional to the ventilation rate. If we, for example, assume a ventilation rate of Q_n during non-occupation periods (75% of the time) and Q_o during occupation periods (25% of the time), the measured average concentration will be proportional to $(0.75/Q_n + 0.25/Q_o)$. If we now assume that the ventilation rate is doubled during the occupation periods compared to the non-occupation periods, the average concentration will be proportional to $(0.75 + 0.125)/Q_n = 0.875/Q_n$. Thus, the measured mean age of air will be only 12.5% smaller than that for the non-occupied periods, even if the occupants can double the ventilation rate.

Consequently, the measured ventilation performance is rather close to that, which the building and service system can yield, and only little affected by the occupants' behaviour.

In the mechanically ventilated buildings, the ventilation is normally drastically reduced at non-occupation time. However, for the purpose of continuous measurement over an extended period, such reduction was not performed during the measurement. Thus, also in this case, the measured ventilation performance is typical for what the building and service system can offer.

It would of course be desirable to be able to measure the ventilation for occupation and non-occupation periods separately. This would require samplers which can be opened and closed, thus making intermittent sampling, only during occupation or non-occupation periods. A development of devices which can perform such intermittent sampling automatically is under way.

8. CONCLUSIONS

The homogeneous emission passive tracer gas technique was easy to apply also in large school buildings. Despite the fact that the measurement time was nearly two months in a couple of cases, no destruction of the equipment was noticed and only a few samplers disappeared. An important reason is that the field equipment is simple, inexpensive and very discrete.

A detailed picture of the air distribution patterns could be obtained, showing that in most schools, the ventilation differs widely between different class rooms. This fact illustrates the importance of using a measurement technique, which like the homogeneous emission technique yields a reliable measurement of air distribution.

All measured naturally ventilated schools have a low average ventilation rate, comparable to that required in dwellings. It seems necessary to use various means of improving ventilation

during occupation time in naturally ventilated school buildings in order to approach a reasonable air quality.

It should be observed that the ventilation rate with occupants present, may be quite different from the average values reported here. However, extended window airing during lessons is not very probably, due to the cold climate during the measurement periods.

9. ACKNOWLEDGEMENT

Most of the school buildings were investigated on commission from customers. Thus, Stockholm Konsult and Falkenberg Kommun is gratefully acknowledged for kindly permitting publication of part of the measurement results. For co-operation and field work the following persons are gratefully acknowledged: Marie Hult, Lars Fyrhake, Cecilia Sundberg, Birgitta Nordqvist and school staff. Anita Eliasson is heartily thanked for performing the laboratory work and Jörgen Sundberg for valuable measurement assistance.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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Developing Tracer Gas Technique to Determine Inter-zonal Air Exchange Rate

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Abstract

The use of local exhaust is considered to be the most effective way to control pollutant dispersion from intense sources, such as in kitchens, in toilets, as well as in copy machine rooms. The optimum air exhaust rate required to prevent pollutants from escaping into the major occupant areas very much depends on the natural air exchange rate(AER) between the hooded room and the major room space. This paper presents a mathematical model and a test procedure of using tracer gas technique to quantify the AER. In this study, the mathematical model analyzes the balance of a tracer gas in two adjacent zones, and establishes two correlations between the dimensionless increase/decay rates of the gas concentrations and the AER. The mathematical model reveals two alternative ways to determine the AER: a). Tracer gas concentrations in both rooms involved be monitored; b).Tracer gas concentration in one of the two spaces be monitored. The latter method was validated in a test for AER quantification between ceiling void and room in a test chamber. Finally the proper use of the two methods is discussed, and it is suggested that the techniques developed in this study can be used for estimating minimum exhaust rate requirement for ventilation system renovation in existing buildings.

List of Symbols

AER: Air exchange rate between two adjacent spaces defined as $3600 \times v/V_c$ (1/hr)
 $C_{R,0}$: initial tracer gas concentration in the room space(mg/m^3)
 $C_{C,0}$: initial tracer gas concentration in the hooded space(mg/m^3)
 C_R : tracer gas concentration in the room space(mg/m^3)
 C_C : tracer gas concentration in the hooded space(mg/m^3)
 C_b : equilibrium concentration in the whole space(mg/m^3)
 V_R : volume of the room space(m^3)
 V_C : volume of the hooded space(m^3)
 v : air exchange volume between two adjacent spaces(m^3/s)
 t : time
 t_0 : initial time
 τ : time

1. Introduction

It is well established that the most effective way to control indoor air pollutants is to directly extract the pollutants at the source location before they are dispersed into the occupancy zones[1, 2]. Range hoods are widely used for kitchen ventilation. Window or wall mounted low-pressure fans are used in toilet/bath rooms. Local exhausts have traditionally been widely used to control pollutants in industrial places, such as welding fumes. In modern commercial buildings, there is an increasing use of various electronic/electrical equipment. Some equipment such as printers and copy machines are known emission sources of some volatile organic compounds[3, 4]. Quite often in Hong Kong, copy machines are located in separate rooms in some open plan offices and libraries. Also, locating electrical appliance in a separate space will reduce noises in the occupied zones. However, there is a question of how to determine the optimum flow rate required for the local exhaust. Ideally, the exhaust should produce such a low pressure in the separate space so that no pollutants will escape into the major spaces concerned. In industrial or laboratory applications, the exhaust air flow rate is determined in such a way so that a minimum facial velocity at the openings of the controlled space are produced to prevent toxic gases from entering the working space. However, the required flow rate determined in such a way may be too high to be cost-effective for commercial and residential applications when there are large openings such as doors in between.

The pollutant may transport through the large openings by diffusion, convection, and pressure differences. In practice, wind pressure distribution around buildings should firstly be considered, and the suspected pollutant sources should be located in low pressure zones. Ultimately, the air-tightness of the building is important in alleviating the wind pressure influences. If this is out of control, the effect of mechanical exhaust will be minimized. Suppose that the wind pressure influence is negligible, it is the natural convection of the hot plumes produced by various indoor heat sources and their interactions with the mechanical air distribution system that determine the distribution of the pollutants. The mechanical ventilation is considered as the control measure to overcome the natural convection processes. Therefore, knowing the exchange due to the convection alone will be useful for the proper design of the local exhaust in this case. The research described in this paper is on the determination of the air exchange between two adjacent zones in buildings when the mechanical ventilation effects are not included. Then, how to determine the local exhaust rate based on this AER will be discussed in the result analysis section.

2. Methods.

The inter-zonal air exchange through a large opening caused by natural convection may be illustrated as in Figure 1. The plume produced by a heat source such as a copy machine will cause the entrainment of the surrounding air. In the confined space, there will be recirculation at the higher level of the plume. This plume in combination with temperature differences existing between the two zones may create air flows through the opening, with air leaving and entering the high temperature space through the upper area and lower area respectively. If the velocity profile through the openings can be determined, the AER can be calculated. However, as demonstrated in many studies, measurements of the detailed flow

velocities are extremely difficult and subject to large errors[5]. Therefore, the method proposed in this study involves the use of tracer gas techniques, which have traditionally been used for determination of air-infiltration rate through building envelopes. The proposed method here is the development of the technique for the determination of interzonal air-exchange rates.

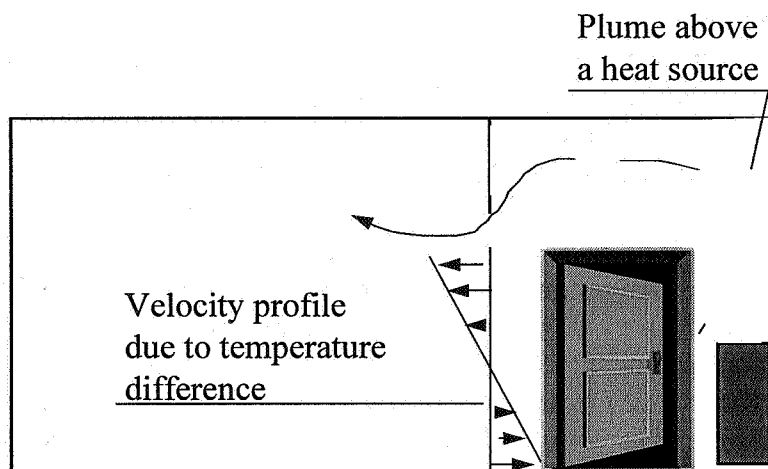


Figure 1. Possible air exchanges between two adjacent zones driven by natural convection

The tracer gas technique involves releasing a gas into the building spaces concerned and monitoring the change of the gas concentration. If there is no adsorption/absorption for the gas, the gas concentration change is caused by air-change from outside or adjacent spaces. Based upon mass balance for the tracer gas, this air change rate can be calculated from the monitored concentration change. In the case of air-exchange between two adjacent zones, different formulation from that for infiltration studies will be required to calculate the air exchange rate. The essential difference is that, unlike the air-infiltration cases, as the concentration decays in one space, there will be an increase of concentration in the adjacent space. The detailed model which addresses this characteristics is described in the Appendix of this paper. The model provides two alternative methods to calculate the AER according the two dimensionless concentration decays monitored. Consequently, two monitoring methods are proposed, with one requiring monitoring the tracer gas-concentration in one space only, and the other requiring monitoring the tracer gas concentrations in both spaces. Either of the methods will enable us to calculate AER using Equation (A.10). As illustrated at the end of the Appendix, only when one room is much larger than the other, can the correlation for infiltration study be used to quantify the AER between two enclosed zones.

3. Test Case and Test Results

To verify the feasibility of the formula described in the Appendix, a test is performed in a test chamber, in which cooled-ceiling panels are installed. Gaps exist between the ceiling panels, so that air exchange will occur between the room and ceiling void(Figure 2). Such a design is to maximise the cooling capacity of the ceiling panels since the upper surface of panel will also heat by convection. However, this cooling capacity will very much depends on the amount of air that penetrate through the ceiling gaps. The cooled-ceiling panels also function as a partition

between two adjacent spaces. Therefore, the tracer gas technique described in the Appendix is valid for this case, although the practical indication of this air exchange is different from that between two adjacent zones. Therefore, test results performed in this test chamber will be presented here to demonstrate the proper use of the technique.

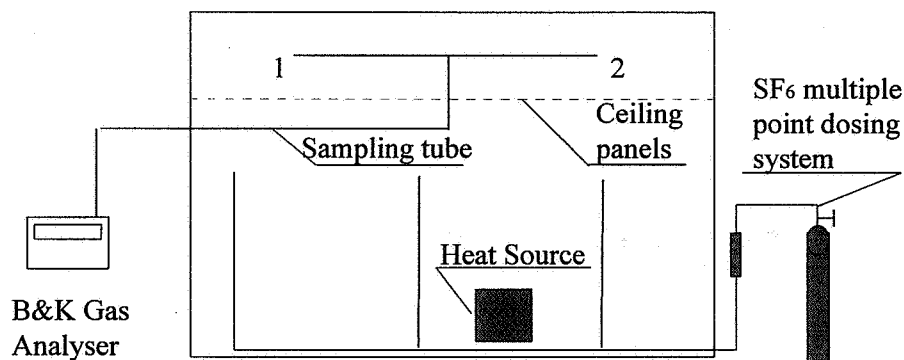


Figure 2. Test of the tracer gas technique in a room with ceiling-panels

In the test, the tracer gas used is sulphur hex-fluoride(SF_6), which is recommended based on its density, toxicity, reactivity, and easiness of being detectable. To realise a good initial well-mixed condition of the tracer gas in the room below, two approaches that enhance the mixing process are combined. Ten dosing points are located uniformly in the room to charge a sufficient amount of tracer gas into the room in a matter of several seconds. Simultaneously, an air fan located in the room was switched on for a few seconds to intensify the mixing. However, the fan will have interference on the natural convection. Therefore, it is essential that the fan, on one hand, is powerful enough to make the air fully mixed in a matter of several seconds while, and on the other hand, does not disturb or cause much air exchange through the ceiling gaps within this mixing process. Different fan mixing speeds were tested. It was found, by monitoring the concentrations at different locations in the room, that low fan speed worked very well, probably because the tracer gas is already purged into the room at diversified positions. Since the air enters the cavity from the room through the gaps scattered all over the ceiling, the average concentration might be close to the nominal instantaneous 'fully mixed concentration', even if the concentration in the room is not 100% uniform. Therefore, very short time fan mixing at low speed was used at the beginning of the measurements.

The monitored SF_6 concentration of the mixed samples from four locations in the cavity is shown in Figure 3. Due to the preliminary test, some background concentration exists in the room: the initial concentration $C_{C,0}$ in the cavity is about 17 mg/m^3 . Since the tracer gas is dosed into the room space, the concentration in the cavity is increasing. It takes about 10 minutes to approach the equilibrium state. Observing from the curve, the equilibrium concentration is about 225 mg/m^3 .

Some measurement errors are taken into consideration in the later data processing. In the later stage of the concentration increasing, the measured C_c will be very close to the final equilibrium concentration C_b . Consequently, any error in the measured value of C_c at this stage will create

large errors in the calculation of the logarithmic value of the dimensionless concentration $\ln(C_b - C_{C,0})/(C_b - C_C)$. Therefore, in the plotting of logarithm of the dimensionless concentration against time, only the measurements in the early stages- the first seven minutes - are used for regression analysis. The plotting shows a good straight line(Figure 4). The slope of the line k is $1/120 \text{ s}^{-1}$. Then according to Equation (A.10), the air exchange rate v/V_C will be $k/(1+V_C/V_R) = 25 \text{ l/hour}$. This amount is considerably high, and its effect certainly should not be overlooked.

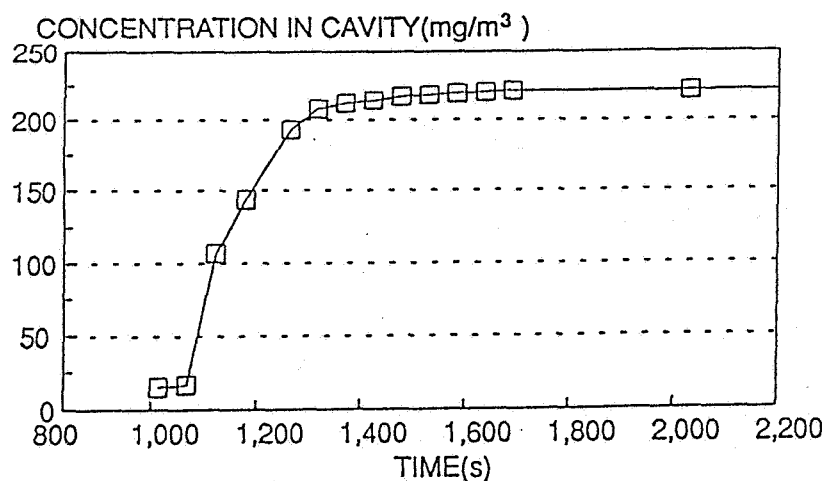


Figure 3. Monitored concentration increase of SF6 in the cavity

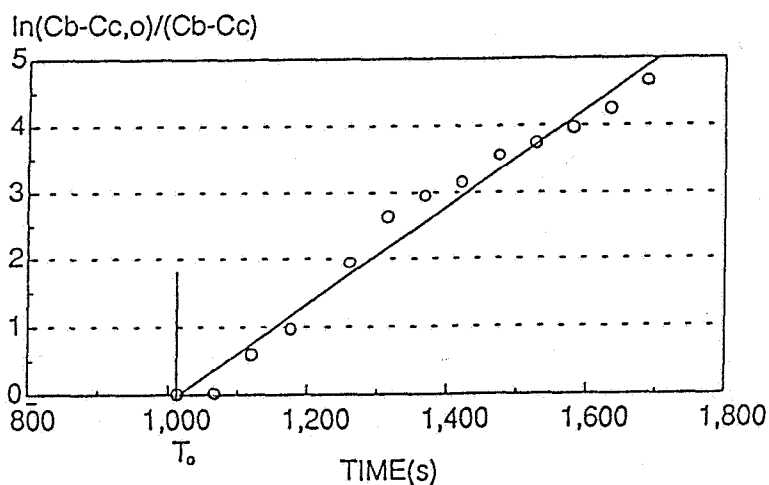


Figure 4. Logarithmic plotting of the measured dimensionless concentration in the cavity

4. Conclusions and Remarks:

1. Tracer gas test procedures have been proposed to determine the air-exchange rate between two adjacent rooms. The method can be used to determine the air-exchange rate caused by buoyancy-driven flows.
2. Attention should be paid to air-flow control to the spaces tested. Basically, all external air-change routes should be cut-off while the test is being conducted.
3. Two alternative test procedures are proposed, with the single monitor method tested in an enclosed room. The single monitor method only requires one tracer gas detector to fulfill the measurement. As demonstrated in the test, when the v/V_C is around 25 1/hour, the equilibrium concentration required can be easily approximated from the first ten-minutes data. Therefore, the value of v/V_R should be estimated beforehand. If this value is too low, the second method described in the Appendix should be used since the time required to achieve equilibrium might be too long.
4. In using the methods, only the data obtained in the initial increase or decay stage should be used for calculating the slope k to avoid numerical errors that will be introduced in the dimensionless concentration calculation.
5. While the test method does provide an air exchange rate, the remaining question is how to design the exhaust air rate according to this air exchange rate. This exchange should be considered as the minimum exhaust air rate required. Below this rate, there are always pollutants escaping into the major room. However, providing a flow rate above this value does not guarantee that no pollutant escape will occur. This is because air exchange may still occur across the a large opening such as a door due to the buoyancy effect. Therefore, the relative location of the pollutant source and exhaust grill to the opening should be carefully considered. Actually, this is the other complicated aspect of the same issue that deserves special investigations. Bearing this in mind, the technique will be useful in estimating the minimum exhaust rate required when ventilation system upgrading is done for existing buildings.

APPENDIX

The idea of using tracer gas technique to measure air change rate is not new. However, for the measurements of air exchange through two adjacent room spaces, some specific mathematical formulation and test procedures will be required. The present contents are the derivation of these formulas.

Consider a room with two air spaces - the cavity above the ceiling and the room below the ceiling, or consider two adjacent rooms in a building, the living room and the kitchen. In this context, the two spaces will be called room spaces and hooded spaces respectively. Assume that natural convection will cause convective air exchanges between the two spaces. To investigate this buoyancy effect, no external air should be supplied. Imagine that a certain amount of tracer gas is purged uniformly into one of the two room spaces and quickly mixed in the room in a very short time interval(Figure A.1 a). Then we have an initial concentration of $C_{R,0}$ in the room and a $C_{C,0}$ in the cavity. After the tracer gas is switched off, the concentration in the room will decay while the concentration in the cavity will increase, as indicated in Figure A.1 c, or

vice versa. Ultimately, the concentration will reach the equilibrium state, with a uniform concentration C_b through out the room(Figure A.1 b). This decay or increase process can be described by the following mathematical method. Consider the cavity and the room as two well-mixed chambers. Then we have balance equation of the tracer gas in the room volume:

$$v (C_R - C_C) = - V_R \frac{dC_R}{d\tau} \quad (A.1)$$

where, v is the air exchange volume between the two spaces, V_R is the volume of the room space. C_R and C_C are the tracer gas concentrations in the room space and the hooded space respectively. Similarly we have the balance equation of the tracer gas in the cavity space:

$$v (C_R - C_C) = V_C \frac{dC_C}{d\tau} \quad (A.2)$$

where, V_C is the volume of the hooded space.

We also have the mass conservation of the tracer gas in the whole space:

$$V_C C_C + V_R C_R = (V_C + V_R) C_b \quad (A.3)$$

where, C_b is the final equilibrium concentration in the whole space.

From Equation (A.3), we have

$$C_R = (1 + \frac{V_C}{V_R}) C_b - \frac{V_C}{V_R} C_C \quad (A.4)$$

Substitute C_R in Eq.(A.2) with Eq.(A.4), we get

$$v (1 + \frac{V_C}{V_R}) (C_b - C_C) = V_C \frac{dC_C}{d\tau} \quad (A.5)$$

Let

$$C_{b-C} = C_b - C_C, \quad (A.6)$$

Since C_b is a constant, Equation (A.5) can be changed into

$$- \frac{dC_{b-C}}{C_{b-C}} = \frac{v}{V_C} (1 + \frac{V_C}{V_R}) d\tau \quad (A.7)$$

Integration of Equation (A.7) yields

$$\ln \frac{C_{b-C,0}}{C_{b-C}} = \frac{v}{V_C} (1 + \frac{V_C}{V_R}) (t - t_0) \quad (A.8)$$

where,

$$C_{b-c,0} = C_b - C_c(t_0) \quad (A.9)$$

That is to say, the logarithm of the dimensionless concentration is directly proportional to the time. Therefore, if the concentration increase in the cavity is measured, and results are plotted according to Equation(A.8), then the air exchange volume ν is related to the slope k of the curve(Figure A.1 d) according to the equation

$$\nu = k \frac{V_c}{1 + V_c / V_R} \quad (A.10)$$

It should be noted that the concentration in the room space, C_R , does not appear in Equation (A.6), (A.8) and (A.9). This means that the concentration in the room is actually not required in monitoring the concentration increase/decay process. What is required is the equilibrium concentration C_b , which is theoretically the ultimate concentration after a infinitely long period. In practice, this value should be reasonably approximated by observing the concentration increase/decay curve of C_R after a reasonable period.

An alternative set of formulations can be derived. From Equation(A.3), we can derive such a correlation:

$$C_R - C_c = (1 + V_c/V_R) (C_b - C_c) \quad (A.11)$$

Substitute this into Equation (A.8), we have

$$\ln \frac{C_{R,0} - C_{c,0}}{C_R - C_c} = \frac{\nu}{V_c} \left(1 + \frac{V_c}{V_R}\right) (t - t_0) \quad (A.12)$$

On the left hand side of this equation, only when the concentration changes in both rooms are measured, can the dimensionless concentration be calculated. Consequently, air exchange volume ν can be calculated according to Equation (A.10). This means that at least two tracer gas monitors will be required. On the other hand, the equilibrium concentration will not be required. This method may be advantageous when the AER is small and the time required to achieve the equilibrium is long.

When $V_R \gg V_c$, and if the tracer gas is initially released only into the hooded space, we will have $C_{R,0} = C_R = 0$. Then both Equation (A.9) and Equation (A.12) will become

$$\ln \frac{C_{c,0}}{C_c} = \frac{\nu}{V_c} (t - t_0)$$

which is actually the conventional equation used in the decay method to quantify air change rate in infiltration studies.

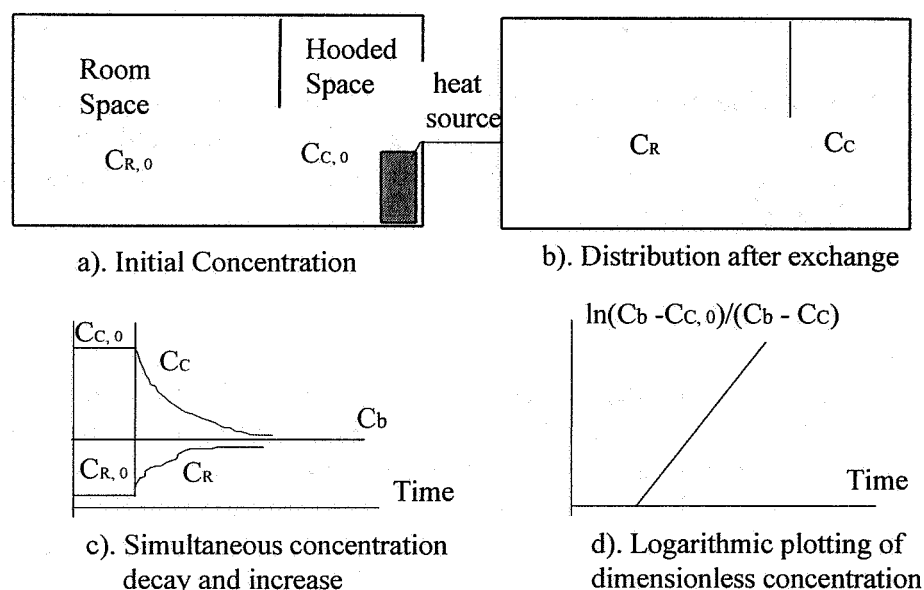


Figure A.1 Illustration of the tracer gas concentration increase and decay in two adjacent spaces

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

**17th AIVC Conference, Gothenburg, Sweden,
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Application of the tracer gas techniques for the assessment of ventilation indexes in real buildings

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Synopsis

The work presented in this paper is aimed at the definition of tracer gas experimental procedures for measuring the air change rate, the age of air and the air change efficiency in real buildings under mechanical ventilation conditions.

The measurement procedures, based on the decay method, were validated in a special experimental chamber and implemented in two rooms of a building under real operating conditions.

Measurements of volumetric flow rate through the air ducts of two buildings, performed by means of the constant emission rate method, will be shown and commented.

Results of some measurements obtained by tracer gas technique have been compared with values measured by using other traditional methods.

List of symbols

n = air change rate (h^{-1})

Q = flow rate through the enclosure (m^3/h)

V = volume of the enclosed space (m^3).

τ_n = nominal time constant (h)

$\langle \tau \rangle$ = mean age of air in the enclosed space (h)

τ_p = age of air in the generic point of enclosed space (h)

τ_e = age of air to air exhaust zone (h)

ε = air change efficiency (-)

t = elapsed time (h)

$C_{\text{exh}}(0)$ = exhaust tracer gas concentrations at times 0 (p.p.m.)

$C_{\text{exh}}(t)$ = exhaust tracer gas concentrations at times t (p.p.m.)

r = correlation coefficient of linear best fit (-)

Q_{air} = air flow rate in the duct (m^3/s);

Q_{gas} = tracer gas flow rate in the duct (m^3/s);

C_{gas} = tracer gas concentration in the duct (p.p.m.);

C_{gasbackg} = background tracer gas concentration in the duct (p.p.m.).

1. Introduction

The improvement of indoor air quality in a building can be achieved by means of well designed ventilation systems able to capture , dilute and remove the pollutants emitted in the space. The capability of the ventilation to provide an acceptable level of air quality is expressed by different indexes like ventilation rate, air change efficiency and ventilation effectiveness.

In the last years, several techniques have been developed either for the preventive analysis of ventilation systems not yet installed, or for the diagnosis of those already existing.

One of the most powerful and suitable technique for measuring in situ the performance of the ventilation systems is the tracer gas technique (TGA), firstly developed for studying the natural ventilation and now used to determine the above mentioned indexes characterising the mechanical ventilation conditions. Moreover this technique allows to measure in real buildings the flow rate through the air ducts, avoiding the traditional methods, based on pitot tubes or anemometers, quite inaccurate in case of turbulent flow due to curves, elbows and shutters.

Generally N_2O and SF_6 are widely used as tracer gas since are not contained in the outdoor air and can be measured with sufficient accuracy even in low concentrations.

There are three basic measurement methodologies according to the conditions of tracer gas emission in the monitored space: concentration decay method (step-down), constant emission rate method (stet-up) and constant concentration method.

In this work the concentration decay method was applied for measuring the ventilation rate, the age of air and the air change efficiency under mechanical ventilation conditions firstly in a special experimental chamber and then in two rooms of an existing building.

The constant emission rate method was used to determine the volumetric flow rate through the air ducts in two buildings.

2. Theory

The ventilation rate of a single enclosed space is defined by

$$n = Q / V$$

The nominal time constant of the enclosure, τ_n , is defined as the inverse of n

$$\tau_n = 1/n = V/Q$$

The index n is a flow rate indicator and does not provide any information about distribution and quality of air. The path and pattern distribution of the supply air can be studied by the concept of age of air.

The age of air is the time in which the air remain into the enclosed space: the youngest air is located near to the supply air register, while the oldest one can be located anywhere according to the air pattern distribution.

The mean age of air is the spatial mean of the local ages calculated over the whole volume of the enclosed space

$$\langle \tau \rangle = (\int_V \tau_p dV) / V$$

For whatever ventilation system the age of air near to the exhaust air grill is equal to the nominal time constant of the enclosure, τ_n

$$\tau_e = \tau_n = V / Q$$

For a mixing ventilation system the age of air is the same in all points of the enclosure and its value is equal to τ_n .

For a displacement ventilation system the age of air varies according to a linear relationship from zero, near to the supply air register, up to τ_n near to the exhaust air grill and its mean value is equal to $(\tau_n / 2)$.

In the actual ventilation systems only a fraction ε of the air flow rate supplied to a room is utilised for the change of air. This fraction is called air change efficiency and is defined by the following expression

$$\varepsilon = \tau_n / (2 \langle \tau \rangle) = \tau_e / (2 \langle \tau \rangle)$$

For a mixing ventilation system the air change efficiency is equal to 0.5 because $\tau_n = \langle \tau \rangle$. In this system only the 50% of supplied air is used for the change of air.

The air change efficiency is equal to 1 when the whole supplied air is utilised for the change of air. This condition occurs only in an ideal ventilation system called "perfect displacement ventilation system".

The air change efficiency is less than 0.5 when the supplied air comes directly on the exhaust grill, by bumping against a limited volume of old air (ventilation short-circuit).

In this last case the age of air in the exhaust grill is less than the mean age of air in the room.

3. Application of the concentration decay method

This method was applied for measuring the ventilation rate, the age of air and the air change efficiency under mechanical ventilation conditions firstly in a special experimental chamber and then in two rooms of an existing building.

3.1 Methodology

In the enclosed space the tracer gas is injected up to reach a uniform concentration over all the volume. During the injection the mixing of the tracer gas can be carried out by means of two small fans located inside the space. In order to minimise the experimental errors a concentration of about 100 ppm should be reached in the monitored space.

During the injection time, the sealing of the openings of the room should be carried out included the supply air inlet and outlet, to minimise the time for reaching a suitable concentration.

The measurement of the concentration (decay phase) starts when an uniform concentration has been obtained, the injection of the gas is stopped, the fans are switched off and the supply air registers are opened.

Before starting with the decay, it is advisable to measure the concentration at least in two

different points of the room to evaluate the mixing.

The age of air in the exhaust air grill and the mean age of air in the room can be determined in the same time by measuring the decay of concentration in a point located on the exhaust grill

$$\tau_e = (\int_0^\infty C_{\text{exh}}(t) dt) / C_{\text{exh}}(0)$$

$$\langle \tau \rangle = (\int_0^\infty t C_{\text{exh}}(t) dt) / \int_0^\infty C_{\text{exh}}(t) dt$$

The air change rate n and the air change efficiency ε can be calculated by means of τ_e and $\langle \tau \rangle$ expressed by the two previous equations.

3.2 Measurements

In a special experimental chamber designed for indoor air quality tests, air change rate and air change efficiency were measured according to the above mentioned methodology.

The chamber is mechanical ventilated by a variable velocity fan through a supply air register located in the centre of a wall, at 2.9 meters above floor level. The exhaust air grill is in the opposite wall at 0.35 meters from the floor.

During the experiment the air flow rate admitted in the chamber was measured by a Venturi tube and was equal to $66 \text{ m}^3/\text{h}$. This flow rate provides a ventilation rate of 1.37 h^{-1} .

Figure 1 shows the decay curve of N_2O concentration in the exhaust air and the very good linear correlation ($r = 0.99$) between the natural logarithm of the measured concentration and the time.

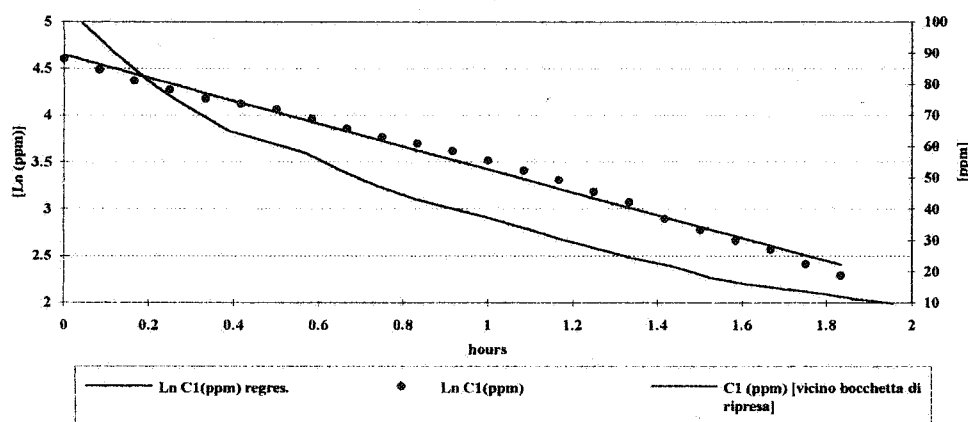


Fig. 1

Decay curve and linear regression

The analytical and numerical integration of decay curve allows the determination of τ_e , $\langle \tau \rangle$, n and ε . Table 1 reports the final results.

$\langle \tau \rangle (h)$	$\tau_e (h)$	$n (h^{-1})$	ϵ
0.65	0.84	1.19	0.64

Tab. 1
Determination of air distribution indicators

The air change rate measured by the tracer gas is about 13% lower than the value measured by means of the Venturi tube. This difference can be reduced by increasing the frequency of sampling during the decay and by stopping the analytical integration at about 40 ppm in order to avoid errors greater than 5% in the measured concentration (accuracy N_2O analyser = ± 2 ppm). Same measurements were performed also in two rooms of a building.

In both rooms there are supply air registers for the primary air and fan-coils for heating and cooling. Since in this case the exhaust air goes to the corridor through the lower slit of the door, the sampling of the tracer gas was located just in this point. During the decay only the fan-coils of room 1 were switched off.

The results of the measurements performed in the two rooms are reported in figure and table 2.

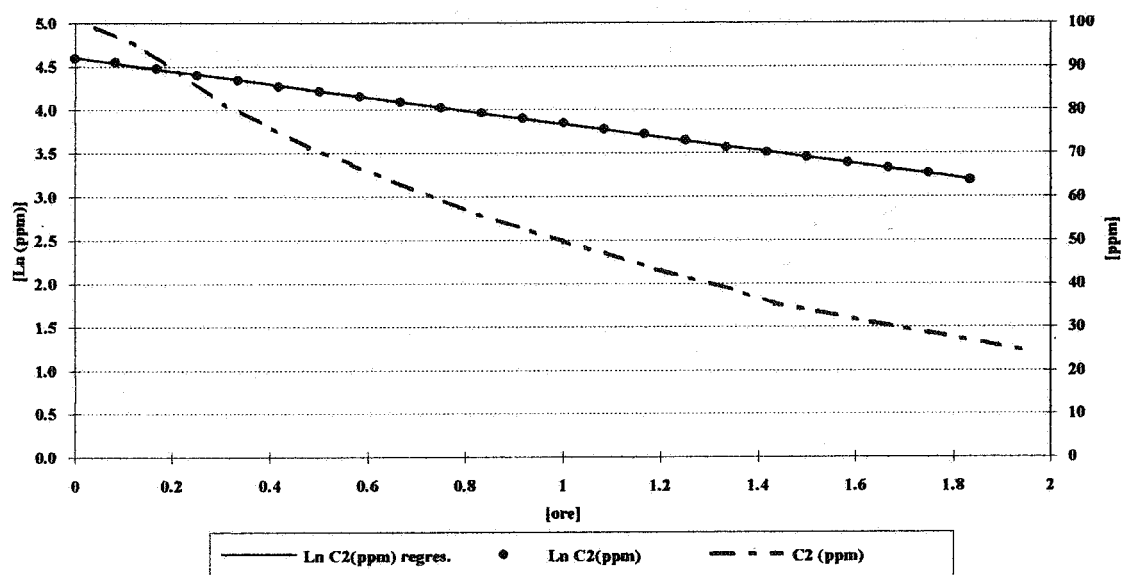


Fig2
Decay curve and linear regression for room

Room1	$\langle \tau \rangle = 0.77 \text{ h}$	$\tau_e = 1.31 \text{ h}$	$n = 0.76 \text{ h}^{-1}$	$\varepsilon = 0.85$
Room 2	$\langle \tau \rangle = 0.65 \text{ h}$	$\tau_e = 0.65 \text{ h}$	$n = 1.54 \text{ h}^{-1}$	$\varepsilon = 0.50$

Table 2
Air distribution indicators for rooms 1 and 2

In the room 1 tracer gas measurements show a low air change rate due to partial closing of the two supply air registers as confirmed by a visual inspection. In this room when fan coils are switched off the distribution of primary air is quite good as confirmed by the high value of the ε index (85 %). In these conditions the 85% of supplied flow rate really contributes to the change of air. In the room 2 the air change rate is acceptable but the primary air is perfectly mixed with the old one since the fan coils were switched on during the measurements. Tracer gas measurements show in this room a situation of perfect mixing since a value of 0.50 was measured for the ε index .

For room 2 the n index measured by tracer gas technique is consistent with the value determined by measuring the air velocity in 9 points of supply air registers and by calculating the mean velocity and the volumetric flow rate.

The difference between the two values is infact only of 3%.

4. Application of the constant emission rate method

The constant emission rate method was applied to determine the volumetric flow rate through the air ducts in two buildings.

4.1 Methodology

Tracer gas is injected with a constant flow rate in the air duct. In steady state conditions the air flow rate in the duct is given by

$$Q_{\text{air}} = Q_{\text{gas}} / (C_{\text{gas}} - C_{\text{gasbackg}})$$

In order to ensure a good mixing of tracer gas and air the distance between injection and sampling point should be 25 times the diameter of the duct in case of rectilinear way or only 10 times in case of elbows.

4.2 Measurements

In the building 1 a secondary conduit which supplies the first floor was selected for the measurements. Its cross section is $(0.46 \times 0.22) \text{ m}^2$ and inside it a steady tracer gas concentration of $48 \pm 2 \text{ ppm}$ was reached by 5 minutes from the injection starting with a constant flow rate of $1.32 \cdot 10^{-5} \text{ m}^3/\text{s}$.

In the second building the measurements were performed inside a main conduit which supplies the offices of the laboratory area. The cross section of this duct is $(0.77 \times 0.62) \text{ m}^2$ and the steady state concentration was of $38 \pm 2 \text{ ppm}$.

Table 3 shows the air flow rates determined by tracer gas technique and by velocity measurements performed on the cross section of the air ducts.

Building 1	$Q_{\text{air}} = 0.28 \text{ m}^3/\text{s}$ tracer gas	$Q_{\text{air}} = 0.25 \text{ m}^3/\text{s}$ anemometers
Building 2	$Q_{\text{air}} = 3.5 \text{ m}^3/\text{s}$ tracer gas	$Q_{\text{air}} = 3.3 \text{ m}^3/\text{s}$ anemometers

Table3
Air flow rate in the in the building air ducts

5. Conclusions

Tracer gas techniques were successfully applied both in lab and real buildings for measuring air change rate, air change efficiency and air flow rates under mechanical ventilation conditions.

Since the technique even if sufficiently accurate is quite time consuming in some situations the application of different methods is advisable.

The determination of air change rate in a room can be performed more quickly by means of balometers directly applied to supply air registers or exhaust grills.

Tracer gas technique is on the contrary absolutely necessary for measuring the age of air and the air change efficiency in enclosed space or to determine the air flow rate in ducts with curves, elbows or shutters.

The air flow rate measured by means of tracer gas in cross sections of ducts are consistent with hot wire anemometers measurements performed in the rectilinear ways of the same ducts.

Acknowledgements

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

**17th AIVC Conference, Gothenburg, Sweden,
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(Title)

**Zonal Model to Predict Air Distribution and Dynamic
Concentration of Pollutant in Ventilated Rooms**

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Synopsis

The first part of the paper will show some aspects of experimental research on air distribution in ventilated rooms. The study has been carried out to get an understanding of the air movement and the ventilation effectiveness by means of tracer gas measurements. It has been investigated the velocity and the distribution of the concentration in a two-dimensional isothermal flow issue of a linear supply opening.

The second part of the paper will describe a proposed zonal model in 9 zones. This model is able to predict the steady-state behaviour of ventilated rooms and the dynamic evolution of a possible pollutant concentration in such environment. The theoretical model is based on the flow element method according to the throw of an isothermal jet and on a typical configuration of a zonal model.

Comparisons between experimental measurements and the zonal model results show a good agreement on the range of usual air change rate in the ventilated rooms.

1. Introduction

One of the purpose of a ventilation system is to produce optimal conditions for occupants. Understanding room air distribution is essential for the design of ventilation systems and control of the room thermal comfort and air quality conditions. The air motion within a room affects the thermal stratification and the contaminants distribution and therefore establish the air quality available to the occupants. Modelling and predicting contamination transport within the rooms or buildings could supply the effectiveness of ventilation and choose the best solution in design.

In the last years strongly efforts have been concentrated on problems of flow patterns, measurements and simulations. The results of the performed work have been published by AICV and by other scientific teams too. But, the difficulty of the problems and the large range of applications, each with its particularity makes impossible for the moment a general solution to be used by the design engineer.

Therefore, this paper describes an experimental and theoretical study on isothermal ventilated room, proposing a zonal model to predict air movement and pollutant distribution.

Measurements and simulations have been carried out by the author in France, at the University of La Rochelle and the subject has been included in the objective of National Action of Research " Qualite de l'air dans les batiments ", beginning in 1994 by CNRS within the ECOTECH programme.

2. Experimental study

The aim of experimental study was to provide information and quantitative data, such as velocities and concentrations, of the air movement generated by an isothermal linear jet. These results were used to check the theoretical solution provided by the zonal model.

The experiments were make in a full-scale model. The geometry of the test room and the dimensions of the linear diffusers are such as given a steady two-dimensional flow in the main body of the room.

The geometrical dimensions are:

test room : $L \times B \times H = 3,50 \times 1,00 \times 2,50\text{m}$

linear diffuser : $B \times h = 1,00 \times 0,005\text{m}$

The supply opening was placed at 0,50m from ceiling and the dimensions of the test room were choose to enable the development of a free jet.

The primary purpos of these model experiments was to obtain knowledge of the air distribution in the room for varied flow rate and velocity in the supply diffuser.

Three types of measurements were performed:

- a) air distribution in the room;
- b) supply velocity and velocity on the axis of the jet;
- c) temperatures on the walls of the test room.

Flow rate was established in the range of usual building ventilation: $n = 1 \dots 10 \text{ Vol/h}$, that means: $u_0 = 0,57 \dots 5,4\text{m/s}$, velocity in the supply opening.

a) Air distribution was measured by means of tracer gas technique. It was employed a single tracer gas technique and the constant injection method. The traser gas was SF₆. Since, the development of the air movement are bidimensional and symmetrical, the points of prelevation tracer gas were placed in an axle plan like in figure 1. The values of concentration were recorded during all the time of experiments. The graphs of concentration in SF₆, is show in figure 2 and 3, for two flow rate: $n = 1 \text{ Vol/h}$ and $n = 10 \text{ Vol/h}$ ($u_0 = 0,57\text{m/s}$ and $u_0 = 5,4\text{m/s}$).

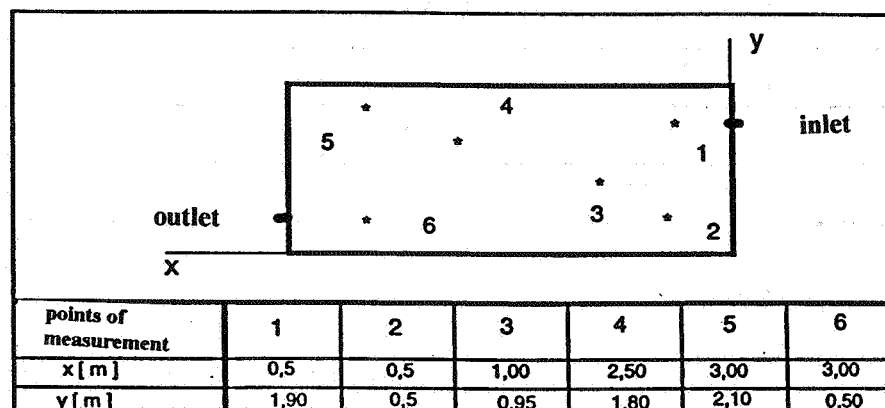


Figure 1. Points of prelevation traser gas

b) The supply velocity is established by means of a micromanometer tube Pitot. Test measurements show that the linear opening gives an uniform distribution of air velocity on its surface. In order to pick up the information on jet development, the velocity was measured in the axis of the jet, in point 1.

Theoretical, velocity in the axis of a plane jet is calculated with the following equation:

$$u^*_{ax} = \frac{u_{ax}}{u_0} = x^{*- \frac{1}{2}} \left(\frac{2k'}{\pi} \right)^{\frac{1}{4}} \quad (1)$$

where:

$$x^* = \frac{x}{h}$$

$$k' = \frac{k}{\left(1 + \frac{690}{Re}\right)^2} \quad \text{for } Re < 10^5$$

and k is a linear diffuser coefficient.

Checking the calculated velocity and measured velocity in the point 1, we have the table 1.

Table 1. Calculated and measured velocity in the point 1, $x = 0,50m$

u_0 (m/s)	Re	k'	u_{calc} (m/s)	$u_{m\acute{e}s}$ (m/s)
0,57	190	2,33	0,062	0,06
0,85	283	4,24	0,108	0,10
2,04	680	12,32	0,340	0,24
2,58	860	15,39	0,454	0,36
3,10	1033	17,98	0,568	0,40
3,91	1303	21,38	0,748	0,27
5,40	1800	26,13	1,086	0,31

The results show the followings :

1. - for $u_0 = 0,57m/s$ to $u_0 = 1,0m/s$ (low flow rate), it's a good agreement between calculation and measurement at a distance $x = 0,50m$ from the supply opening;

2. - for $u_0 > 1,0m/s$, occurs a deflection of the jet due to the Coandă effect. The air jet sticks, after some distance, to the ceiling and after another distance it detaches creating a quite

rapid rotating movement below it. This is one of the principal reason of air distribution in the room. More information and details on experimental study were given in [4], [5].

c) Surface temperatures are measured by 0,2mm chrom-aluminium thermocouples and the temperatures were recorded. The difference between the six surface temperatures of the test room was lower that 1 ° C. This boundary decay of isothermal state did not alter the general stream line pattern within the room. In the region of developed turbulence the inertial forces are dominated against the thermal forces and molecular diffusion.

3. Numerical prediction of the air movement

The prediction method presented in this paper is based on general main assumptions of zonal model:

- the ability to predict the main driving flow (boundary layer, jet or thermal plume);
- experimental and empirical knowledge to describe particular processes.

The performed work and results of zonal models have been presented within the frame of IEA Annex 20 and in the scientific communications [2], [3].

Our zonal model in nine controller zones, presented in this paper, is able to predict the air diffusion and pollutant distribution in a room ventilated by an isothermal linear jet. In order to compare the predicted and measured results, we have kept the same boundary conditions like in the experimental study: flow rate, geometry of the room and diffuser. We considered that the bottom zone of the room is covered by the isothermal jet which is the main driving flow. All other zones were assumed like " current zones " as shown in figure 2.

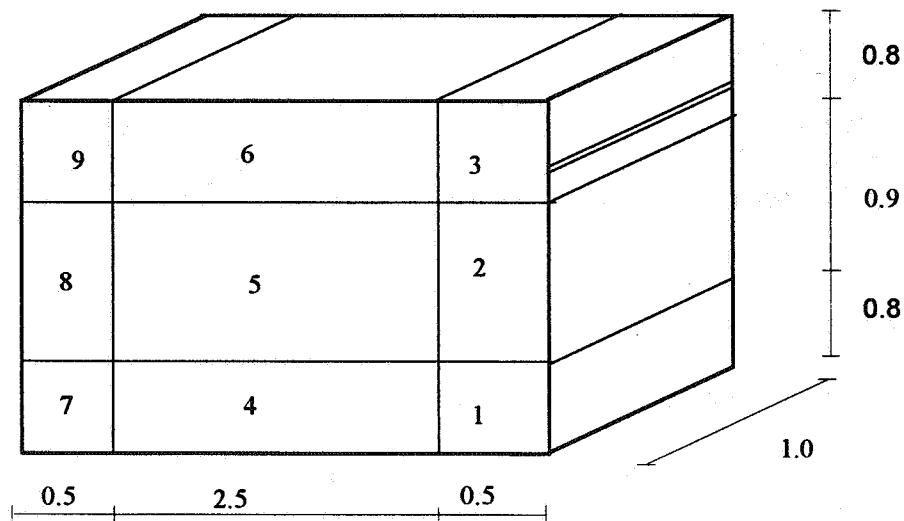


Figure 2. Theoretical room division

3.1 Characteristic equations

Knowing the air jet behaviour and considering the theoretical borders been permeable, we have proposed the air movement sketch as shown in figure 3.

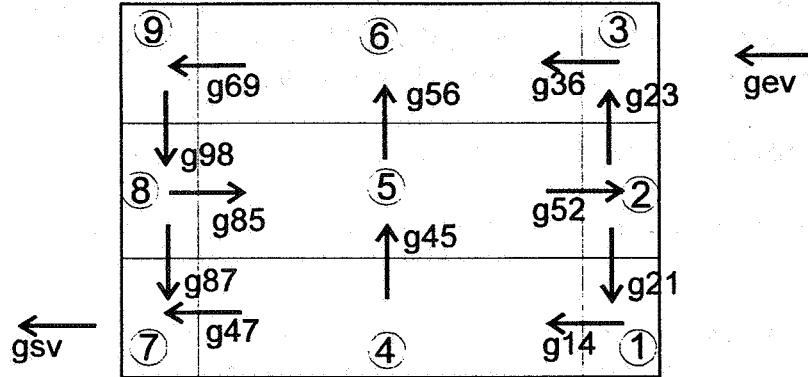


Figure 3. Sketch of air movement

a) Steady-state equations

The inside air volume is divided into nine isothermal zones coupled with mass conductance. For each zone the mass balance is written:

$$\sum_j g_{ij} + g_{Pi} = \sum_j g_{ji} + g_{Si} \quad (2)$$

To complete the system of equations other four equations must be given or identified. Jet flow equations supply mass flow through 3 - 6 and 6 - 9 borders.. We assume that the mass flow variation in the zones 2 and 7 is proportional with an entrained coefficient K_j :

$$g_{ij} = \frac{A_{ij}}{\sum A_{ji}} g_s = K_i g_s \quad (3)$$

and A_{ij} is the surface of the borders. So, we have the mass flow g_{47} and g_{52} .

Solutions of equations (3) are the mass flows through theoretical borders.

The corresponding velocities are calculated with the equation of continuity:

$$g_{ij} = \rho_i u_{ij} A_{ij} \quad (4)$$

b) Unsteady-state equations

The air movement in the room is permanently for each supply flow rate, but the evolution of a pollutant concentrations is unsteady.

For an air volume crossed by mass flow g_{ij} , which drive a pollutant concentration C_j , the local variation of concentration C_i can be calculated :

$$\rho_i V_i \frac{dC_i}{dt} = \sum g_{ij} C_j \quad (5)$$

This differential equation written for a control volume can be replaced by a difference equation:

$$\frac{dC_i}{dt} = \frac{C_i^{t_n} - C_i^{t_{n-1}}}{\Delta t} \quad (6)$$

The moment value of concentration is considered uniformly within a volume. This concentration is calculated function of the neighbourhoods concentrations and the previous local concentration :

$$C_i^{t_n} = C_i^{t_{n-1}} + \frac{\sum g_{ij} C_j^{t_n}}{\rho_i V_i} \Delta t \quad (7)$$

The nine differential equations are replaced by nine difference equations generating a matricial system :

$$[A]x[X] = [B] \quad (8)$$

This system was solved by means of Gauss method.

For solve the steady-state and unsteady equations, it was need to create a numerical programme named by author, CONNY. Figures 4 and 5 show the values of velocities through borders for $n = 1 \text{ Vol/h}$ and $n = 10 \text{ Vol/h}$, respectively, compared with a $k-\epsilon$ simulation. Figure 6 presents the concentration profile obtained by numerical simulation using CONNY programme and by measurements recorder.

Following remarks are presented:

- using a $k-\epsilon$ numerical simulation for the same boundary conditions we have obtain isovelocity curves. The distribution and the values of this curves enable us to declare available the sketch of mass flow distribution;
- comparing the measured and calculated concentration graphs we observe the general evolution on the same values and the same maximal value attained. For the smaller flow rate, $n < 4 \text{ Vol/h}$, the numerical profile has a deviation roughly 10% compared with the measurement graph. For $n > 4 \text{ Vol/h}$, the two profiles are following the same values.

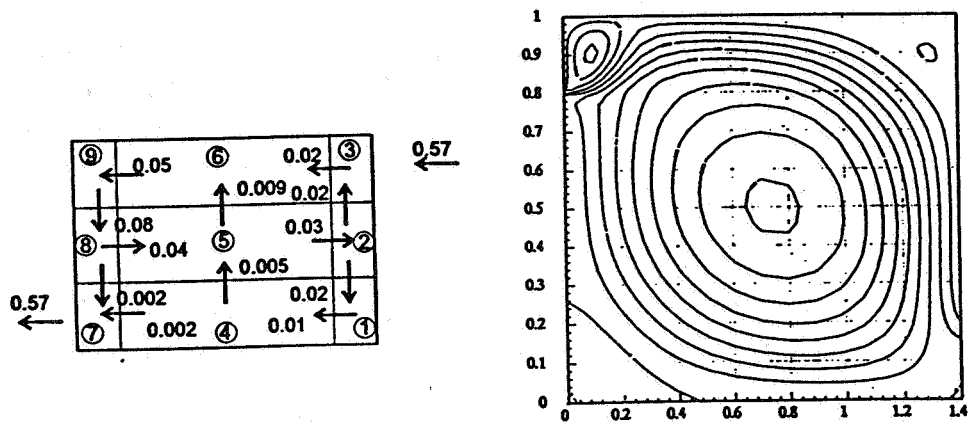
4. Conclusion

For the usual ventilated rate in civil buildings, $n = 1..10 \text{ Vol/h}$, the nine zonal model described in this paper proves its ability to predict air distribution (mass flow and velocities) and concentrations in a room. Taking into account the ahead observations, this zonal model can be a tool for the design engineer in HVAC. But, its application is limited to the rooms ventilated by an isothermal plane jet. We intend to develop this zonal model, changing the type of jet and his thermal state. Also, we study the possibility to apply this zonal model to predict the behaviour of a heated and ventilated rooms.

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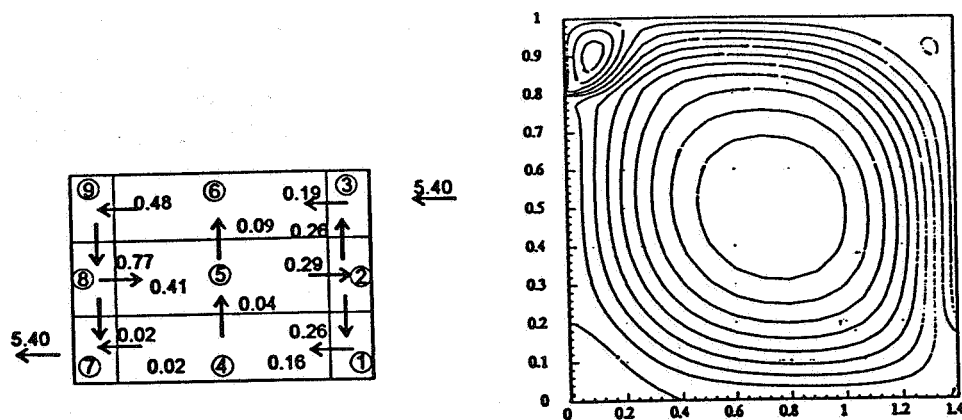
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International Energy Agency



a) velocities predicted by zonal model

b) isovelocities, k-ε simulation

Figure 4. Air movement in room, $n = 1 \text{ Vol/h}$



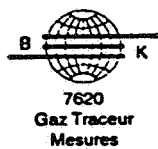
a) velocities predicted by zonal model

b) isovelocities, k-ε simulation

Figure 5. Air movement in room, $n = 10 \text{ Vol/h}$

gev= 1.288333E-02

Measured concentrations



REGLAGES:

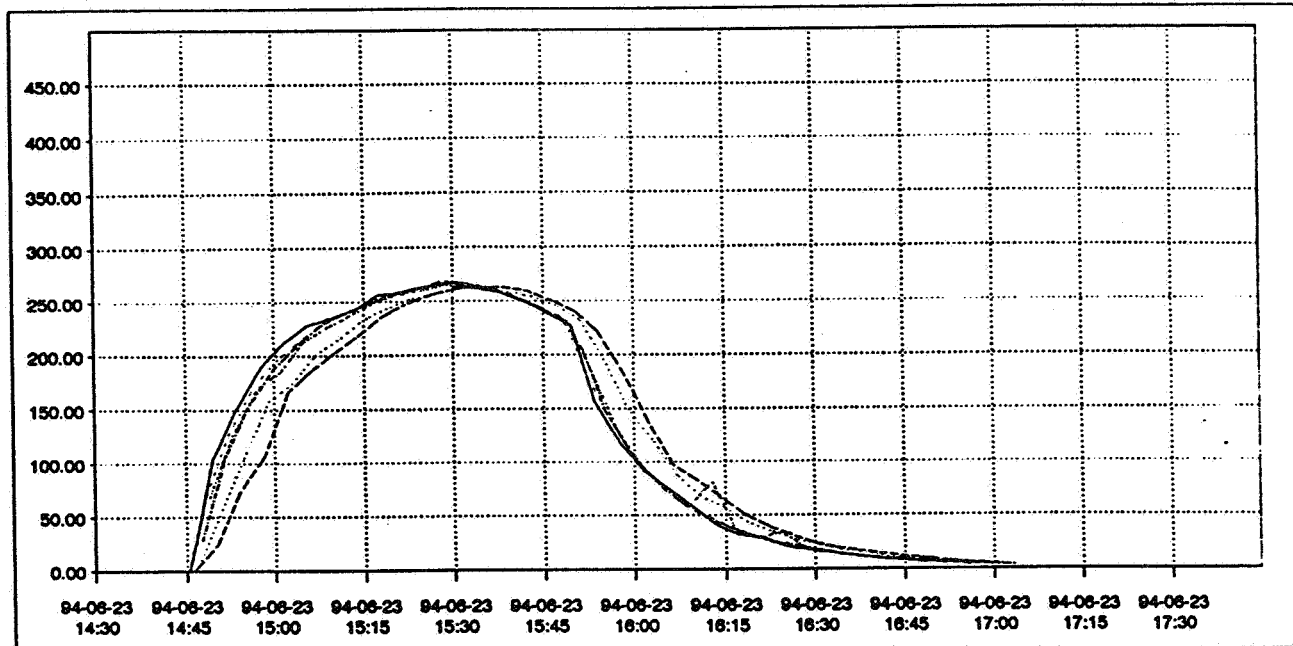
Interval. Echant.: Cont.
Interv. Temps mesure: 238 s
Temp. de normalis.: 293.1 K
Compens. croisee: Non
Compens Vapeur d'eau: Non
Pression Air: 101.30 kPa

INFORMATION GENERALE:

Heure Debut: 1994-06-23 14:40
Heure Fin: 1994-06-23 17:30
Temps Compression: Aucun
Defaults: Oui
D. Editees: Non
Resolution: 100 %

COMMENTAIRES:

T=43.9 Volt
Vo=2.58 m/s
Q=46.44 m³/h
N=5.28 Vol/h
DONNEES: JUN23PM
PRG PERSO: Aucun



C	Position:	Mesures	Trace Ech.	Unite	Vol. Piece
1	Vole 1302	GAS UA0988	1E+00	mg/m ³	8.80 m ³
2	cellule	GAS UA0988	1E+00	mg/m ³	8.80 m ³
3	cellule	GAS UA0988	1E+00	mg/m ³	8.80 m ³
4	cellule	GAS UA0988	1E+00	mg/m ³	8.80 m ³
5	cellule	GAS UA0988	1E+00	mg/m ³	8.80 m ³
6	cellule	GAS UA0988	1E+00	mg/m ³	8.80 m ³

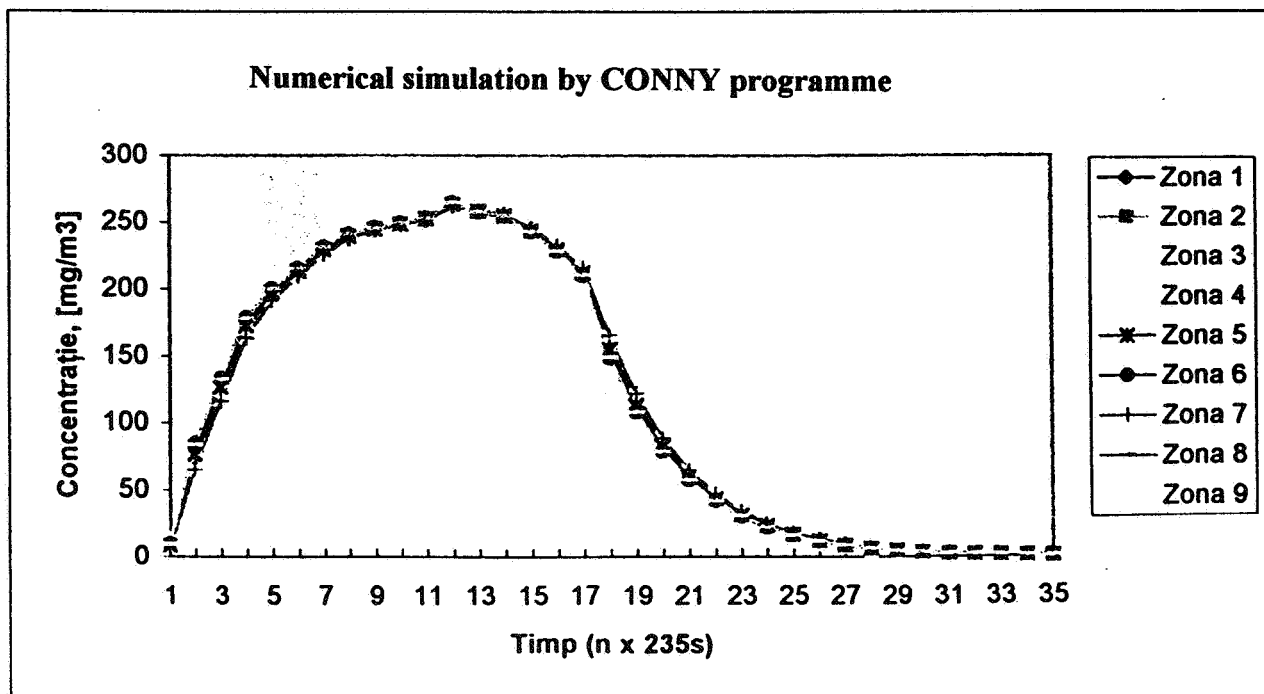


Figure 6. Local concentration measured and calculated by zonal model, n = 5 Vol/h

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

MEASUREMENT OF VENTILATION AIR FLOWS

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Synopsis

The common way to determine air infiltration, exfiltration and interzonal flows from tracer gas measurements in multizoned buildings is to rely upon the standard single or multizone model, $V \dot{c}(t) = Qc(t) + p(t)$. Here c , p are zonal tracer concentrations and injections, t is time and V , Q are the sought volumes and flows. This model may work well provided that all zones are sufficiently well mixed and all flows really are constant during the measurements. The latter can be doubtful in naturally ventilated buildings, especially as the measurements may require several hours.

This paper discusses one single question: What could be gained if the flows of the model are allowed to vary in time?, i.e. we will consider a (bilinear) model: $V \dot{c}(t) = Q(t) c(t) + p(t)$. The variation of $Q(t)$ will, however, be restricted to be piecewise linear. This extension will complicate the identification procedure, i.e. the computations, somewhat. On the other hand it does not alter the tracer gas measurements needed. The idea is rather to pick out more information from the standard tracer gas measurements.

To test the extended model, single-tracer gas measurements have been carried out in a natural ventilated row house. Some basic concepts are introduced with one and two zone models and the house is modelled with a nine zones model. As intended, the extended model matches the measurements better. However, as demonstrated in the paper, this does not guarantee that the model found actually is better.

List of Symbols

$c(t)$	tracer gas concentration vector, n by 1	q	flow
$\dot{c}(t)$	time derivatives of $c(t)$	$\hat{q}$	mean flow
n	number of zones	$Q(t)$	flow matrix, n by n
n_b	number of breakpoints -1	t	time
$p(t)$	tracer gas injection vector, n by 1	V	diagonal volume matrix, n by n

1 Introduction

This paper deals with the task how to determine multizone models of a building from tracer gas measurements. A multizone model describes the air flows within the rooms of a building and between a building and the outdoors. Once known, these models can be used e.g. to simulate the spreading of an contaminant and to calculate air quality indices as ventilation efficiency and air change efficiency.

The basic equations representing tracer gas concentrations and air flows in multizone buildings have been derived by a number of authors, for example Roulet and Vandaele (1991, Chap.3). Here follows a compact summary of the notation and concepts needed.

The standard multizone model

A multizone model consists of n zones and an outside (index o) with infinite volume and a constant background concentration, c_o . It is no restriction to set $c_o = 0$ and consider all concentrations to be above the true background concentration. The zones are indexed by i (and j), which range from 1 to n . A single zone model can be considered as a multizone model with $n = 1$.

The zones have volumes, v_i . All zones (i) may be connected to/from other zones (j), and to/from the outside (o) by one-way air flows. These flows are denoted by q_{ji} , q_{ij} , q_{oi} , q_{io} , respectively, (i.e. flow is directed to first index from second index. On the assumption of uniform air density and immediate and perfect mixing in the zones, the conservation of masses of air and tracer gas leads to the following equation (e.g. Roulet and Vandaale (1991, eq. 2.39)

$$V \dot{c}(t) = Qc(t) + p(t) \quad (1)$$

where V is a diagonal n by n volume matrix of all zone volumes v_i , Q is a n by n flow matrix where the off diagonal values are the flows q_{ij} , $i \neq j$, and the diagonal q_{ii} equal the negative values of the total inflows (= total outflows) from each zone i . Furthermore, the row sums of $-Q$ equals the inflows from the outside to each zone i , i.e. $-\sum_{j=1}^n Q_{ij} = q_{io}$, and the column sums of $-Q$ equals the outflows from each zone j to the outside, i.e. $-\sum_{i=1}^n Q_{ij} = q_{oj}$.

To have a meaningful physical interpretation there are some requirements of the V and Q matrices: all volumes v_i and flows q_{ij} , q_{io} , q_{oi} must be positive or zero. More stringent, it is natural to require all the total in/outflows $-q_{ii}$ and the volumes v_i to be positive. Otherwise, the corresponding zone(s) can be excluded from the model.

In order to identify the V and Q matrices from a tracer gas measurement a method described in Hedin (1994) is used. This is a single tracer gas method. Numerically, it is an iterative method with a derivative (or integral) approximation based on the so far found model and a constrained least square parameter estimation. This ensure the found model to be physically meaningful, according to the requirements just mentioned. The method can also handle sequential sampling in a proper way. To summarise - in the absence of measurement noise etc. - this method is able to find the V and Q matrices without error despite the approximation of $\dot{c}(t)$ and sequential sampling. $\square$

Two important assumptions above are constant flows (at least during the measurements) and perfect mixing of the zones. In many cases, especially when all internal doors are closed and a small mixing fan is used in each zone, the multizone model works fine and is able to give a good explanation of the tracer gas measurements. On the other hand, with internal doors left open or without mixing fans the assumptions of instantaneous and perfect mixing can not be fulfilled and the model fails to describe the measurements.

But even with good mixing in each zone, the model can fail. This happens if the flows are not constant, but vary in time. In a tight, mechanically ventilated building the assumption of constant flows can be good. It can also be reasonable in a building with mechanical extract ventilation, as the extract flows and the total supply flow are quite stable. Due to the internal negative pressure, the distribution of supply flows may be reasonably stable too. But, in a naturally ventilated building, both air flow rates and distribution of supply and extract flows may vary in time. This is because the air flows depend on external forces as inside-outside temperature difference and wind speed, i.e. quantities that are supposed to vary in time. Consequently, the assumption of a constant flow matrix can be very doubtful in a naturally ventilated building. This shortage is the motive of introducing a varying flow in this paper.

2 A Single Zone Test

At this point, a single ($n = 1$) zone example can be illustrative. The measurement is taken from a bedroom in a naturally ventilated row house (the house and experiment will be described in a subsequent section), see Figure 1. It is obvious from the figure, that a constant flow and volume can not describe the changing slope of this experiment.

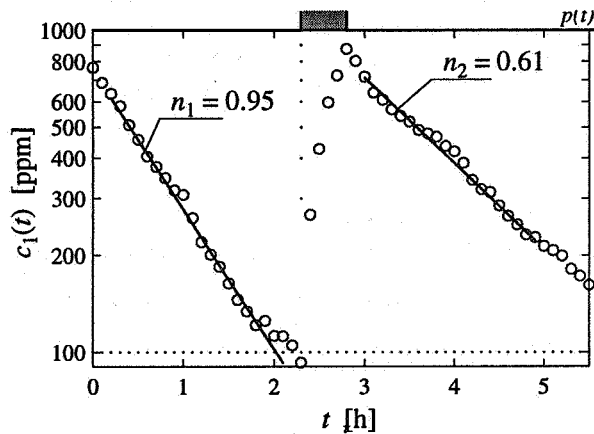


Figure 1. Measured tracer gas concentration (o) in bedroom No 1 with the window slightly opened. Tracer injection indicated on top. Two different slopes n_1 , n_2 marked with thin lines.

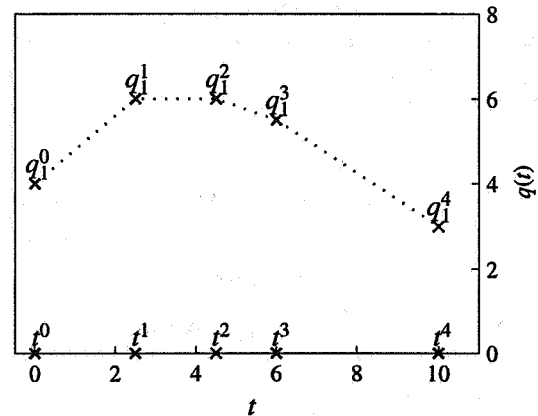


Figure 2 A piecewise varying function $q(t)$, ($n_b = 4$). (Right hand y-axis used for flows throughout this paper)

Therefore, let us introduce n_b+1 breakpoints in time: $t^0, t^1, \dots, t^{n_b}$ and then allow $q(t)$ to be piecewise linear between these points, see Figure 2. Here $n_b = 0$ means the common case with $q(t)$ constant. Now, if $n_b > 0$ the model does not longer consist of a single v_1 and q_1 but of v_1 and $q_1^0, q_1^1, \dots, q_1^{n_b}$. The identification procedure will be somewhat more complicated, but hopefully this model¹ can describe the measurements better.

Figure 3 shows model and measurements for $n_b = 0, 1, 2, 4$ and the more extreme $n_b = 32$. In this test it seems profitable to allow q_1 to vary in time. It is reasonable to believe that the new models describes the true variation of $q(t)$ better than the constant ($n_b = 0$) model does.

Up to a certain limit n_b can be increased and better and better models will be, see Table 1. But at the same time the identified $q(t)$ will be more and more prone to errors, as fewer and likely less informative data are used for each identified value. Moreover, occasional odd measurement values (due to random measurement errors or incomplete mixing) will have a larger influence. Of course, there is also an upper limit on n_b that depends on the number of data. This is due to the fact that to solve an equation there are always needed as many equations as unknowns. Unfortunately, the mean value of $q(t)$ (denoted $\hat{q}$) is not supposed to converge to a true mean value when n_b increases.

3 The bilinear model

Before continuing to discuss the used multizone model we will return to the basic mass balance equations with a non-constant flow matrix $Q(t)$:

$$V \dot{c}(t) = Q(t)c(t) + p(t) \quad (2)$$

While the ordinary mass balance (1) is a linear equation, this equation is only linear as long as $Q(t)$ is constant or as long as $c(t)$ is constant, otherwise it is non-linear. This type of equations are called bilinear, Mohler(1973).

¹ A remark is what it would be more general to also allow a non-constant volume $v(t)$. This can indeed be done, but since there is no appropriate physical interpretation of a non-constant volume, this is avoided.

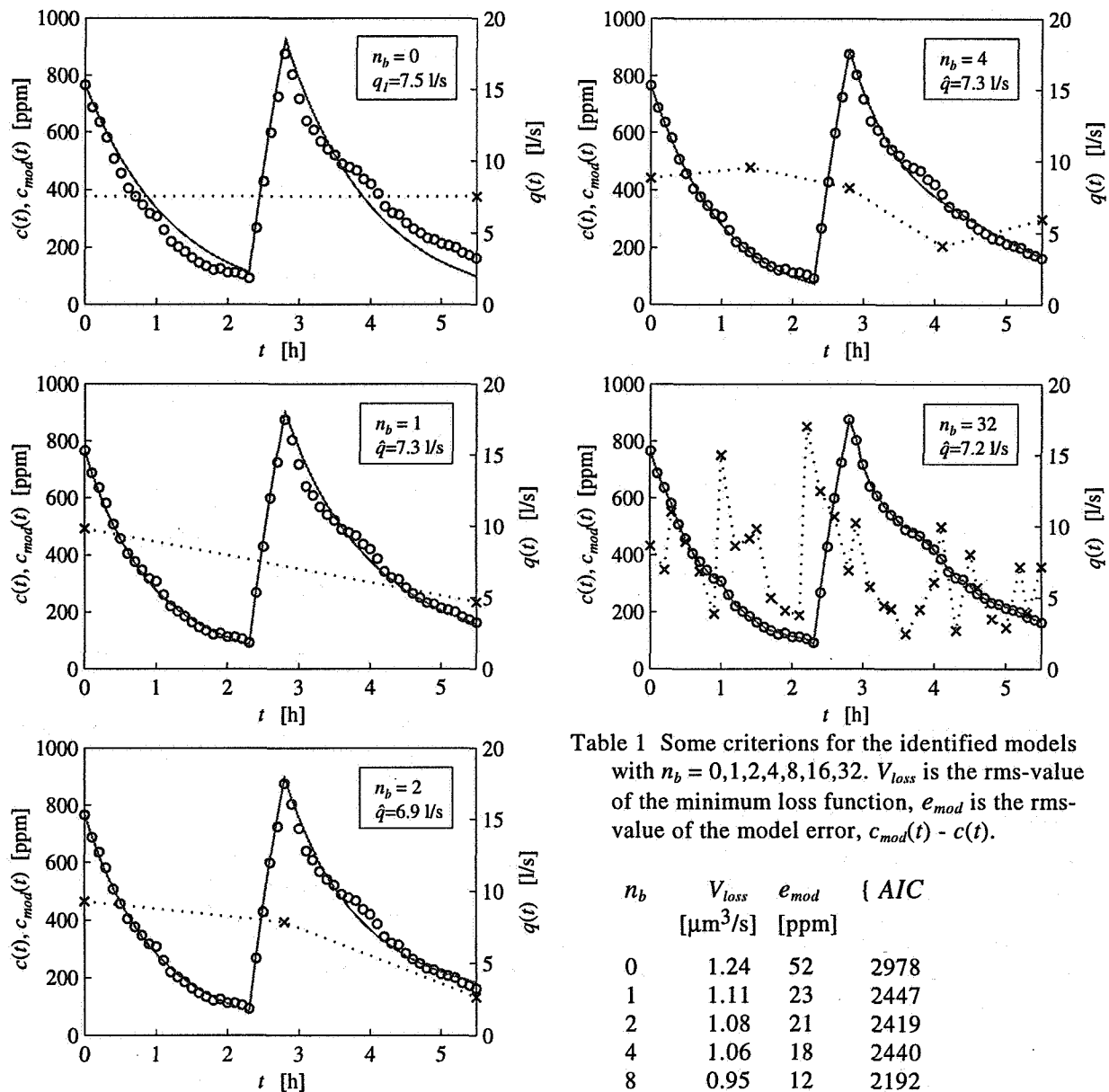


Table 1 Some criterions for the identified models with $n_b = 0, 1, 2, 4, 8, 16, 32$. V_{loss} is the rms-value of the minimum loss function, e_{mod} is the rms-value of the model error, $c_{mod}(t) - c(t)$.

n_b	V_{loss} [$\mu\text{m}^3/\text{s}$]	e_{mod} [ppm]	{ AIC
0	1.24	52	2978
1	1.11	23	2447
2	1.08	21	2419
4	1.06	18	2440
8	0.95	12	2192
16	0.83	9.5	2053
32	0.54	4.9	1418 }

Figure 3. Measurements (o), model (—) and identified flow $q(t)$ (---) in the right hand scale for $n_b = 0, 1, 2, 4, 32$.

One common method to determine a general function $Q(t)$ goes through the method of Volterra series. However, this solutions includes a lot of mathematics and seems to be too difficult, at least at this first try. A more convenient method to introduce a non-constant $Q(t)$ may be to restrict oneself to a piecewise linear variation of $Q(t)$, as was exemplified in Figure 2. Of cause this is a restriction compared with a general function. Yet, if many breakpoints are used, it is possible to mimic rather complicated behaviour.

One important point is that the driving forces (due to wind and temperature differences) can also be described by a piecewise linear function. Another important characteristic of the piecewise linear choice is that if e.g. both Q^k and Q^l comply with the linear constraints imposed on the solution, then will also the linear interpolation $\theta Q^k + (1-\theta)Q^l$, $0 \leq \theta \leq 1$ comply with these constraints.

A first order example

To get a feeling for what a variable $q(t)$ means we shall look on a simulation of a single zone system:

$$v_1 \dot{c}(t) = -q_1(t)c(t) + p(t) \quad (3)$$

where

$$v_1 = 1 \text{ and } q_1 = \begin{cases} 1+0.2t & (\text{flow increasing in time } 0 \leq t \leq 5) \\ 1 & (\text{constant flow}) \\ 1-0.2t & (\text{flow decreasing in time } 0 \leq t \leq 5) \end{cases} \quad (4)$$

Figure 4a shows a step up and Figure 4b shows a step down with these three flows. It is obvious from the figures that a varying flow can introduce a great deal of flexibility in modelling. Thus it should be possible to model the measurements more accurately. □

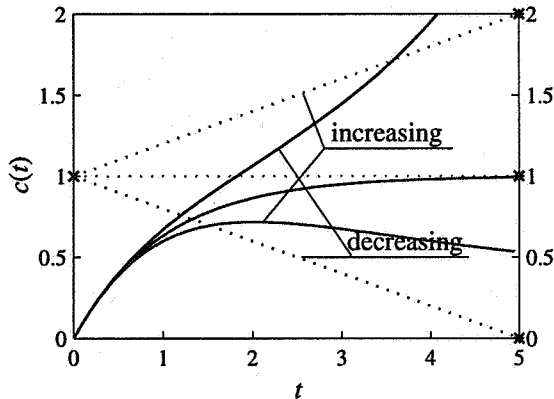


Figure 4a. Simulated tracer gas concentration during a step up (i.e. $p(t) = 1$, $c(0) = 0$).

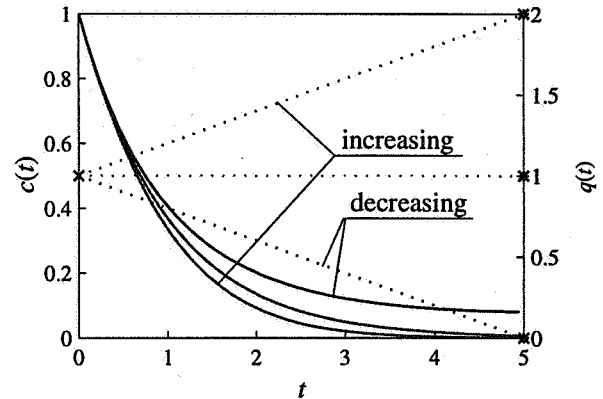


Figure 4b. Simulated tracer gas concentration during a step down (i.e. $p(t) = 0$, $c(0) = 1$).

Another first order example (poor mixing)

However, it is important to notice that this flexibility could also be a serious disadvantage. The following example is designed to emphasise that fact. Suppose that a room has a poor mixing and behaves like a two-zone space, see Figure 5, with the mixing flow q_{mix} as low as 1. Anyway it is modelled by a single zone model and $q(t)$ is (erroneously, or at least unnecessarily) allowed to be non-constant. The following results come out for $n_b = 0, 1, 2, 4$, see Table 2. (Notice that if the correct model structure is used (i.e. $n = 2$) then there will be no error at all.)

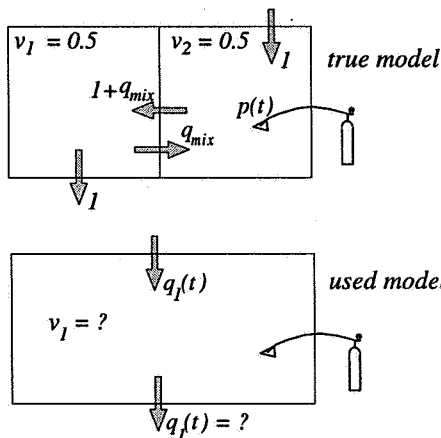


Figure 5 (to the left)

True and used model for a simulated case.

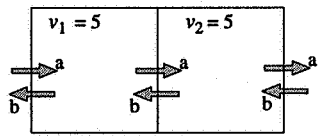
Table 2 Identified results from simulated model then an erroneous model structure is used.

n_b	V_{loss}	v_1	q_1^0	q_1^1	q_1^2	q_1^3	q_1^4	$\hat{q}$
0	62.0	10.0	1.04					1.04
1	56.5	10.9	0.92	1.07				1.00
2	52.0	11.0	0.92	0.97	2.40			1.32
4	45.6	12.2	0.60	1.02	0.93	1.97	0.03	1.06

As seen, even if $n_b > 0$ gives a more accurate model (smaller V_{loss}), the identified volume and flows are inferior to the $n_b = 0$ solution (which in fact is quite good, despite the bad mixing). As poor mixing is common in tracer gas measurements, this result should discourage the use of varying flow, unless this really is needed. □

A second order example (turning wind - varying flow)

Next, we show a simple example then it indeed is necessary to use a varying flow. Consider the flow system in Figure 6. Notice that the flow turns around from going to the left at $t = 0$, to going to the right at $t = 1$. Thus a constant model can only represent the mean flows. The results then $n_b = 0$ are shown in Table 3. Even if the model (not shown) does not match the simulation too bad, the identified mean flows are 29 - 92 % in error. On the other hand, the piecewise model with $n_b = 1$ will identify the flows and volumes without any error, as it is designed to do. $\square$



$$\begin{aligned} a &= 1 - t \\ b &= t \\ (0 \leq t \leq 1) \end{aligned}$$

Table 3. Identified results then $n_b = 0$.

$$V_{id} = \begin{pmatrix} 5.3 & 0 \\ 0 & 4.2 \end{pmatrix}$$

$$V = \begin{pmatrix} 5 & 0 \\ 0 & 5 \end{pmatrix}, \quad Q = \begin{pmatrix} -(a+b) & b \\ a & -(a+b) \end{pmatrix} = \begin{pmatrix} -1 & t \\ 1-t & -1 \end{pmatrix} \quad Q_{id}^0 = \begin{pmatrix} -1.37 & 0.79 \\ 0.96 & -1.29 \end{pmatrix} \text{ compare with } \hat{Q} = \begin{pmatrix} -1 & 0.5 \\ 0.5 & -1 \end{pmatrix}$$

Figure 6. Simulated two zone model.

4 The Studied Row House

The studied building is a two-storied row house with basement, see the plan in Figure 7. The ground floor has living-room, kitchen, laundry and hall, upstairs there are three bedrooms, bathroom and hall, and in the basement there are living-room, two storerooms and a hall.

Heating is provided by a radiator system for waterborne heat. The house is naturally ventilated with inlet vents in the storerooms and laundry and outlet vents in the laundry, toilet, upstairs hall and bathroom. The bedrooms have no vents and the air change of the bedrooms are known to be poor. Therefore the windows are always slightly opened during the night in the bedrooms and the basement living-room, which is also used as a bedroom.

The three floors are connected by stairs. In this way the three halls and stair-ways compose a large open volume. Such a space is known to be difficult to describe. It does not behave as a single zone since the mixing is too poor. A three zone description may be possible, but there are not three distinct zones, and further, the interzonal flows may be large and hence the concentration will be almost equal in the three zones. As a consequence, the measurements will not be informative enough to give good parameter estimations. The three zone description have been tried in this paper.

5 Tracer Gas Measurements and a Nine Zone Model

Single-tracer gas measurements were carried out under two conditions: with closed or slightly opened windows in bedrooms and the basement living-room. Internal doors were kept closed and small mixing fans (10 W) were used in each zone. The house was fully furnished, but nobody was at home during the measurement.

The tracer gas equipment used can only handle up to nine zones. Due to this limitation, the house was divided into the following nine zones: (on upper floor) three bedrooms, bathroom and hall (on ground floor) living-room, kitchen, and hall, (in basement) hall only. The following specific assumptions (i)-(iv) are believed to be true and are necessary to make the nine zone model valid:

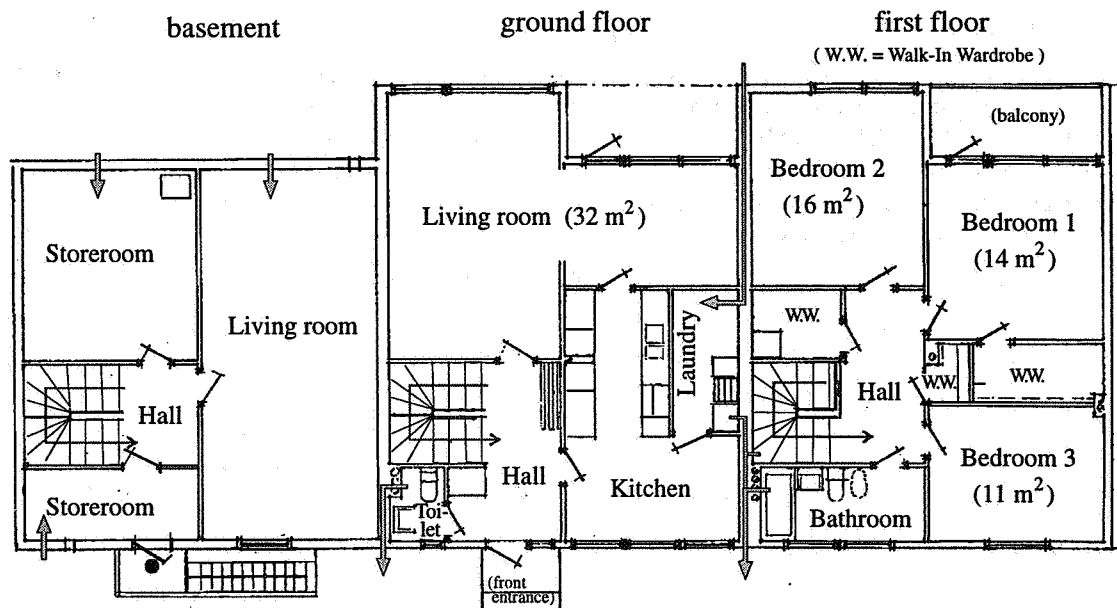


Figure 7 Plan of the row house. Location of vents indicated by grey arrows.

(i) The air flows between the basement hall and the rest of the basement are supposed to be directed (almost) only into the hall. To check this point, smoke was released near the doorways before the measurements. This means that only the total supply air rate into the basement hall (but not the distribution of supply air from the different rooms in the basement) can be determined. (ii) The laundry next to the kitchen, are supposed only to receive air flow from the kitchen and it was excluded from the model. Thus, the exfiltration of the kitchen will include the air flow from the kitchen to the laundry. (iii) Similar conditions applies to the toilet next to the ground floor hall, i.e. flow directed from hall to toilet only. (iv) The walk-in wardrobes on the upper floor have no vents and the doors were closed. They were supposed to have an air exchange with the nearby rooms, small enough to be neglected by this model. □

There is only space for some few results from the identification with the nine zone model in this paper. In Figure 8 we show the tracer gas measurements and an simulation with the identified model. The actual values of volumes and mean flow rates are inserted into the plan in Figure 9.

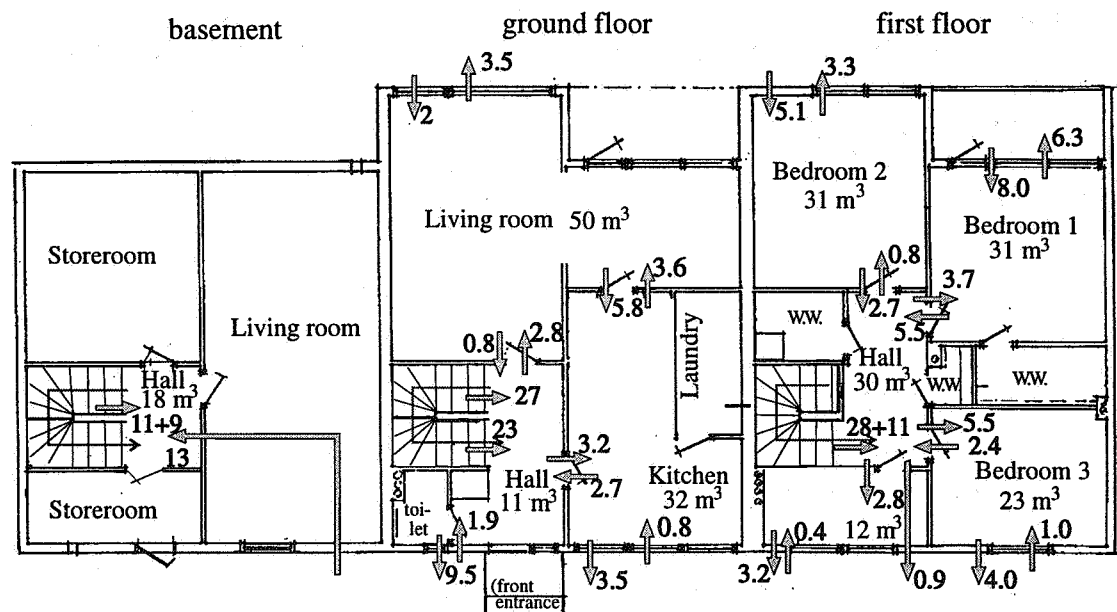


Figure 9 Identified volumes in m³ and mean air flow rates in l/s.

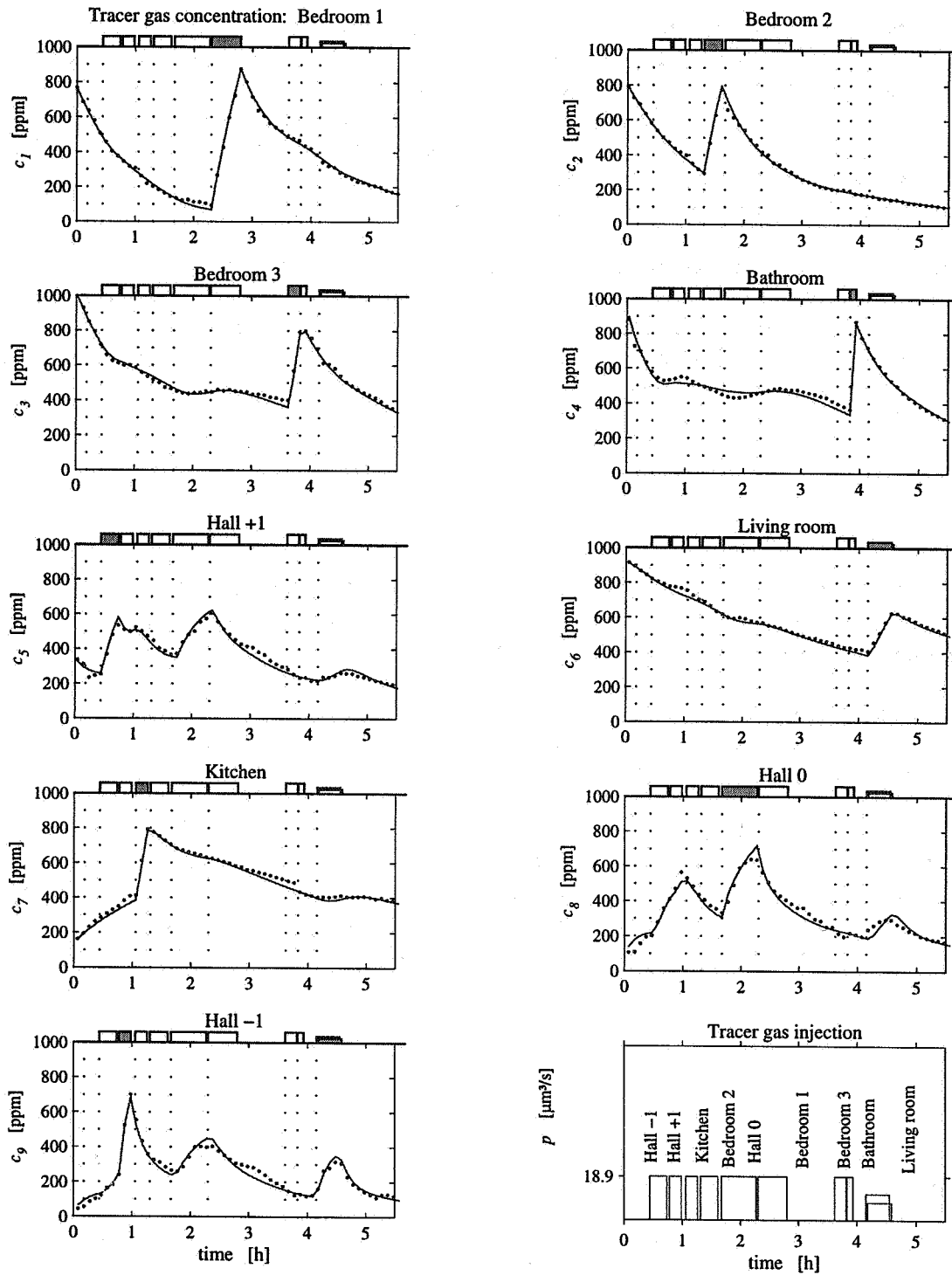


Figure 8. Measured tracer gas concentrations (o) and simulated model (—) for a nine zone model of the row house. Windows in bedrooms slightly opened. $n_b = 3$.

Another tracer gas measurement carried out some days later with different wind conditions shows about the same flow rates, but different supply paths (i.e. larger infiltration in bedroom 3 on north side and smaller in bedroom 1 and 2 on the south side of the house). A third tracer gas measurement, now with all windows closed, shows considerably smaller air change rates in the bedrooms. The air change rates in the bedrooms decrease from about fairly good $0.5 - 1 \text{ h}^{-1}$ then the windows were slightly opened to about a tenth of these values. The air change rates in the bedrooms with closed windows thus proved to be unacceptable low.

Some attempts have been done to relate the variation of identified air flow rates with measurement of outdoor temperature and wind. These attempts have so far not been successful. One problem seems to be that the models have no 'reason' to keep the variations of $q(t)$ small, even then the profit is minor. Possibly, this can be corrected, if an extra 'penalty' on the variance of $q(t)$ is included in the identification routine.

6 Summary

An extension to the traditional description of a flow system with a multizone model: $V \dot{c}(t) = Qc(t) + p(t)$, was tested. The constant flow matrix Q was exchanged by a non-constant flow matrix $Q(t)$, which may vary piecewise linear in time. The new model was applied to tracer gas measurements carried out in a natural ventilated row house.

A single zone model was used to describe one bedroom and a nine zone model to describe the main part of the house. As expected, the new model have smaller loss function and matches the measurements better than the constant model does.

It has been demonstrated by simple simulated examples that the model will use a varying $Q(t)$ not only to describe a varying air flow as intended. But it will also use it to mimic the effects of poor mixing and measurement noise seen in the tracer gas concentrations. This may introduce a false variation of air flows in the model. It was not possible to relate the identified air flow rates with climatic data of wind and outdoor temperature (measured at a site 2 km from the studied row house).

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

**17th AIVC Conference, Gothenburg, Sweden
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**Subfloor and house ventilation rates: comparing measured and
predicted values**

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SYNOPSIS

This paper reports on the use of BRE's domestic ventilation model, BREVENT, to predict subfloor and whole house ventilation rates in a BRE/DoE test house. Before the model could be used though some minor adjustments were necessary because one of its underlying assumptions was that the subfloor temperature was equal to the external temperature. Temperatures measurements over a number of months showed this assumption to be false and so an extra stack term was introduced into the model. However, the overall difference this makes is still quite small, only a few percent at most.

The predicted subfloor ventilation rate matched the calculated value well, particularly when it was stack dominated. When wind played a significant part though the level of agreement deteriorated, particularly when subfloor air bricks were located on unsheltered walls. However, both the subfloor and whole house ventilation rate of the test house appears to be heavily influenced by the stack effect because the suspended floor and ceiling are leaky in comparison to the walls. As a result, subfloor ventilation will be stack dominated about 61% of the time. To improve the prediction of wind affected subfloor ventilation better pressure coefficient data is required.

In a similar vein BREVENT can predict whole house ventilation rates best when the flow is stack dominated. Analysis of the separate stack and wind effects show that the ventilation in the test house will be stack dominated for about 86% of the time. When wind speed does influence ventilation then wind direction also has an effect: winds blowing from the East and West generally give ventilation rates 25% lower than those blowing from the North and South.

1. INTRODUCTION

BRE has conducted research on ventilation beneath suspended floors for a number of years [1,2]. Subfloor ventilation is an important subject to study because it is used as a means of controlling moisture and hazardous gases from the ground. *Hartless & White* [2] described the measurement of ventilation beneath the suspended floor of a BRE/DoE test house. The measurements showed that subfloor ventilation was very much affected by the stack effect. This is somewhat at odds to our initial thinking, i.e. wind entering an air brick in one subfloor wall and leaving through another with stack effect taking no part. However, this assumes an air-tight suspended floor; whilst this may be true of pre-cast concrete floor incorporating a gas-resistant membrane it is not the case for a suspended timber floor. In this case, large quantities of air are likely to pass through it. Indeed, *Bassett* [3], *Lilley et al* [4] and *Hedin* [5] have all shown this using pressurisation and tracer gas techniques.

In this paper, the measurements taken by *Hartless & White* are compared to predictions made using BRE's domestic ventilation model, BREVENT [6] so that the mechanisms for subfloor ventilation may be better understood. Further, whole house ventilation rate measurements for the BRE/DoE test house are compared to BREVENT predictions. Before proceeding with this it is worth discussing the subfloor temperatures recorded in one of the energy and environment test house because these data affect the way BREVENT models subfloor ventilation.

2. SUBFLOOR TEMPERATURE MEASUREMENTS

As part of their ventilation measurement programme *Hartless & White* measured the air temperature beneath the suspended timber floor of the BRE/DoE test house. These measurements were extended and since March 1995 both the air and concrete oversite temperatures have been monitored continuously. The main finding is that the subfloor temperature, T_f , varies very little throughout the day and night but it does follow the expected seasonal trend, ranging from 20°C in the Summer months to 10°C in the Winter months. In fact, it is reasonably well correlated with external temperature, T_o . Such a correlation has also been noted by *Welsh* [7] in BRE's radon test house. Further, T_f is correlated extremely well with the concrete oversite temperature. The oscillations in the concrete oversite temperature are minimal and it is approximately equal to the air temperature although it is generally slightly higher.

Subfloor and concrete oversite temperatures continue to be measured and they will be reported on in future work. The next section of this report describes the implications of these subfloor temperature measurements when modelling subfloor ventilation with BREVENT.

3. BREVENT: MODELLING SUBFLOOR VENTILATION

BREVENT has been used before to predict subfloor ventilation rates [1], but an important assumption behind the model is that T_f is assumed to equal T_o . This simplifies the ventilation calculation because stack effect terms are ignored, but as measurements have shown T_f can be significantly different from T_o (by as much as 15°C). This meant that the equations underpinning BREVENT [6] had to be altered slightly to allow for this. The main change was to incorporate a stack term (i.e. $(\rho_o - \rho_f)gh$ where ρ_o and ρ_f are external and subfloor air density respectively, g is acceleration due to gravity and h is the depth of the subfloor void) into the equations governing the pressure difference across (i) the subfloor air bricks and (ii) the suspended floor.

Despite the potentially large temperature differences mentioned above the effect of ignoring this extra stack term is still likely to be small because of the shallowness of subfloor voids in UK, typically only 20 to 30cm deep. BREVENT works on a mass balance of air flows into and out of the subfloor void and the main house, and all volume flow and ventilation rates are expressed at STP. For the purposes of this work the limit chosen for mass balance was 0.1 kg/hr, and the ventilation rates for the subfloor void and main house were expressed at the subfloor and internal air temperatures (T_f and T_i) respectively. A BREVENT comparison exercise showed that the subfloor ventilation rate was indeed affected by taking a unique T_f rather than assuming it was equal to T_o . In fact, the percentage change was directly proportional to $T_f - T_o$. For example, for the BRE/DoE test house a value of $T_f - T_o$ of 15°C gave a 4% difference in the subfloor ventilation rate. Although the effect for typical values of $T_f - T_o$ is small it was decided to continue to use the revised version of BREVENT.

4. MODELLING SUBFLOOR VENTILATION IN BRE/DoE TEST HOUSE

4.1 Input data

The dimensions of the test house (including those of the subfloor void) and the layout of the site are given elsewhere [2].

One of the key input variables for any ventilation model is the total leakage of the building envelope and its distribution around the external surfaces. A pressurisation test on the test house gave a total leakage of 17.2 ach (i.e. 3,560 m³/hr) at 50Pa ($n = 0.61$). *Hartless & White* noted that a significant leakage path in the test house was the wall/floor junction with air moving from the subfloor void into the house under the skirtings as well as into the gap behind the plasterboard lining. Since then the floor perimeter has been sealed and the envelope leakage re-measured. A value of 13.8 ach (i.e. 2,860 m³/hr) at 50Pa ($n = 0.62$) was obtained. Despite this considerable reduction in total leakage (nearly 20%) smoke tests showed that the floor was still quite leaky, particularly around the floor hatches.

This information together with data from *AIVC Technical Note 44* [8] on the size of leakage paths in housing were used to distribute the total leakage around the envelope of the test house. One of the key features of the house is the absence of windows and doors in E-W facing gable walls and so these were assumed to have minimal leakage. Table 1 shows the results of this exercise.

Surface	Leakage before floor sealing		Leakage after floor sealing	
	(m ³ /hr at 50Pa)	% of total	(m ³ /hr at 50Pa)	% of total
Floor	1600	45	900	31
Ceiling	1200	34	1200	42
Front wall (South)	330	9.2	330	12
Gable wall (East)	50	1.4	50	1.7
Back wall (North)	330	9.2	330	12
Gable wall (West)	50	1.4	50	1.7
Total	3560	100	2860	100

Table 1. Estimated distribution of leakage for test house

The percentage figures are quite interesting. They show that both before and after sealing over 70% of the total leakage is estimated to be through the floor and ceiling.

Another significant input variable for BREVENT is the pressure coefficient data. BREVENT contains sets of pressure coefficient data which have been obtained from BRE wind tunnel tests on a range of general housing models. However, none of the available sets corresponds

exactly to the environment in which the test house is situated, i.e. a single row of detached houses with partially sheltered front and back walls.

This particular problem of the wind environment around a single row of detached houses has been tackled before in a number of papers arising from ventilation work carried out on the Alberta Home Heating Research Facility (AHHRF) in Canada, e.g. *Wilson & Walker* [9] and *Walker & Wilson* [10]. Further, *Walker* [11] has reviewed pressure coefficient data for sheltered buildings as measured by a number of workers. Unfortunately, an important consideration is the need for pressure coefficients on the lower part of the wall where the subfloor air bricks are located. Generally, wall-averaged data are used so one pressure coefficient applies to the whole wall but in practice there are variations over the surface. BREVENT has data for the lower wall as derived from the original wind tunnel measurements.

Therefore, a judgement was made to use the BREVENT set of data corresponding to a 20% housing density. Housing density is defined as the area occupied by buildings divided by the area of the site. In order to simplify calculations it was assumed that the N-S and E-W directions were lines of symmetry. This meant all wind directions could be reduced to a single quadrant, and this quadrant was then divided into four sectors. As a result, only four sets of pressure coefficient data were required. Generally speaking, during the subfloor measurement programme, the prevailing wind direction was the SW.

4.2 BREVENT predictions

Following on from *Hartless & White* the results of two measurement runs are compared to BREVENT predictions. For Run 1 air bricks were open on the E-W facing walls, and for Run 2 they were open on the N-S facing walls. The results of the two runs are plotted in graphs 1a to 2b in two ways: (a) as subfloor ventilation rate against time, and (b) as predicted versus measured subfloor ventilation rate. The line shown in the (b) graph is the 1:1 line, i.e. predicted rate matching measured rate perfectly.

For Run 1, graphs 1a and 1b show good agreement between predicted and measured values. In this case BREVENT predicts the subfloor ventilation rate to within 7%. For Run 2 there are times when the agreement between measured and predicted rates is poor. Graph 2a shows that these times correspond to instances of sharp rises and falls in the measured subfloor ventilation rate. Again, as confirmed in graph 2b, the predicted rate is below the measured rate, but the general daily trend in subfloor ventilation is predicted correctly.

The main reason for the differences between predicted and measured rates can be put down to wind speed, U . As noted by *Hartless & White* the subfloor ventilation of the test house is heavily influenced by the stack effect. Wind speed only plays a part when it is reasonably high and air bricks are open on the front and back walls. For Run 1 where air bricks were open on the sheltered gable walls the agreement between measured and predicted is good because the wind cannot greatly affect subfloor ventilation. When the air bricks are open on the front and back walls the wind does have an effect when its speed is high enough. For Run 2 a dramatic deterioration in the agreement between predicted and measured rates is seen. For a further run (not shown here) where the air bricks were open on all four walls the

agreement appears to be good most of the time.

The reason for the poor agreement in some instances is that the wind is not being properly modelled since the pressure coefficient data used do not adequately represent the environment around the test house. Many of the problems with selecting pressure coefficients for this house have already been discussed, so it would appear that the only way to proceed is to build a scale model of the test houses and the environment immediately around them and obtain the required data from wind tunnel measurements.

The agreement between measured and predicted rates is good at times despite inadequate pressure coefficient data because stack effect virtually dominates subfloor ventilation in these cases. There are two reasons for this stack dominance: (i) open air bricks are located on the sheltered gable walls, and (ii) the test house is effectively acting as a chimney. As discussed previously, the gable walls are relatively air tight and so over 70% of the total envelope leakage is estimated to be through the floor and ceiling. In other words, air enters the house through the subfloor air bricks, is drawn up through the floor and exits through the ceiling. This is the dominant mechanism for air movement in the test house, and this is only affected when U is high enough to overcome it by blowing on either the front or back wall thereby causing cross ventilation. The same thing can happen if air bricks are open on the front and back walls: more air will flow across the void and not as much will flow up through the floor.

Inspecting the graphs of subfloor ventilation plotted against wind speed and subfloor ventilation plotted against temperature difference in *Hartless & White* we can estimate that subfloor ventilation is stack dominated when $T_i - T_o (= \Delta T)$ is greater than 6°C and that wind only becomes a significant influence for U greater than 3.5m/s . Analysing the weather data recorded at BRE we see that subfloor ventilation will be stack dominated 61% of the time, and the wind will only have a significant influence for 7% of the time.

5. MODELLING WHOLE HOUSE VENTILATION RATES

5.1 BREVENT predictions

Since the floor perimeter has been sealed the whole house ventilation rate has been measured continuously using SF_6 . Data is recorded at about half hour intervals. Therefore, as a further test of BREVENT, the model was used to predict ventilation rates in the test house. The input data used for these runs was as described in section 4.1 with the obvious difference of a tighter floor and a subfloor ventilation area reduced to just four open air bricks on the East facing gable wall. In the light of the experience gained from the subfloor ventilation predictions, two months in 1995 were analysed in detail: March (generally a windy month) and July (generally a calm month).

Graphs 3a and 3b show measured compared to predicted whole house ventilation rates for March 1995. It is obvious that BREVENT consistently over-predicts the ventilation rate. Graphs 4a and 4b are the equivalent graphs for July 1995 and they show a good correlation between measured and predicted values. In fact, BREVENT predicts the ventilation rate to within about 5%. The main reason for this difference is that the wind speeds during March

are consistently higher (average 3.8m/s with a maximum of nearly 20m/s) than during July (average 2.4m/s with a maximum of nearly 8m/s). As a result, the problems outlined in section 4.2 are still apparent. The floor may have been tightened considerably but it is still leaky and so stack flow continues to dominate.

5.2 Stack and wind dominated house ventilation

As described in section 4.2 it is helpful to analyse how ventilation is affected by the wind and stack effect. To this end we can use the approach of *Warren & Webb* [12]. They proposed a simple infiltration model (the forerunner of BREVENT) for houses and calculated the infiltration (ventilation) flow rate, Q_v , in terms of the total leakage of the house envelope, the height of the house envelope (H), U , ΔT and an infiltration rate function, F_v . Similarly, they also calculated two other infiltration flow rates, Q_w (for wind dominated ventilation, i.e. $\Delta T=0$) and Q_b (for stack dominated ventilation, i.e. $U=0$) and, accordingly, proposed two further infiltration rate functions: F_w and F_b . The function F_b depends on the house shape and the distribution of leakage among the walls, ceiling and floor. The function F_w in addition depends on the surface pressure coefficients. Using these relationships, *Warren & Webb* were able to separate out the wind and stack effect terms by plotting a graph with the following dimensionless co-ordinates:

$$X = \frac{Q_w}{Q_b} = \frac{F_w}{F_b} \cdot \frac{1}{(Ar)^n} \quad Y = \frac{Q_v}{Q_b} = \frac{F_v}{F_b} \cdot \frac{1}{(Ar)^n} \quad (1)$$

where Ar = Archimedes number = $\Delta T g H / T_i U^2$. Using a range of U , T_i and T_o with a constant T_i (10°C) and wind direction (North), BREVENT was used to predict whole house ventilation rates for the test house. From these predictions the average values obtained for F_b and F_w were 0.224 and 0.025 respectively. The value for F_b is comparable to the values achieved by *Warren & Webb*, but the value for F_w is much lower. It is perhaps not surprising that F_b is an order of magnitude greater than F_w since this accords with the observation that ventilation in the test house is stack dominated. With these values for F_b and F_w graph 5 was plotted using the above dimensionless co-ordinates. The graph shows a single curve bound by the two asymptotes $Y=1$ and $X=Y$ which is identical to the results achieved by *Warren & Webb*. The first of these two asymptotes is the stack dominated region, and the second is the wind dominated region.

Accordingly, *Warren & Webb* proposed the limit for stack dominated infiltration as, $Q_v/Q_b \leq 1.1$, and for wind dominated infiltration as, $Q_v/Q_w \leq 1.1$. Close inspection of the wind speeds and temperature differences at the above limits show that for the test house

- $U/\Delta T \leq 0.7$ for stack dominated infiltration, and,
- $U/\Delta T \geq 11$ for wind dominated infiltration.

Following on from this and using the wind and temperature data obtained from March to November 1995 we can see that stack dominated infiltration of the test house will occur about 86% of the time and that wind dominated infiltration will occur about 0.4% of the time.

5.3 Effect of wind direction

The effect of wind direction has not been studied in detail up to this point since stack effect has been the dominant driving force for ventilation. The main reference to directional considerations has been to the location of subfloor air bricks. Following on from section 5.2 though we can study the influence of wind direction on the ventilation rate in the test house by selecting cases where $U/\Delta T > 1.4$. This limit is somewhat arbitrary but was chosen because it was twice the limit for stack dominated infiltration thereby ensuring that wind played a significant part in driving ventilation. Reviewing the wind speed and temperature difference data for March and July 1995 shows that this condition was met for just under 6% of the time.

Having isolated cases where wind effects are significant we can investigate the effect of wind direction in particular. This can be accomplished using the method of *Wilson & Walker*. They used this approach to derive "shielding factors" for the AHHRF by normalising the measured house ventilation rates. The AHHRF and the DoE/BRE test houses are similar in that they are a row of detached houses with, in the former case, no shielding and, in the latter case, minimum shielding to the South. In other words, the shielding factor for southerly winds will be one.

This procedure was carried out on the measured data for March and July 1995 and the results are plotted in graph 6. The upper and lower dotted lines on the graph represent 95% confidence limits. The graph shows that the normalised ventilation rate has a minimum for both easterly (0.9) and westerly (0.75) winds which is to be expected considering the layout of the houses. For northerly and southerly winds the normalised ventilation rate has a maximum which is about 1.1 for both directions. (This is a consequence of similar degrees of shelter to the North and South as well as equivalent leakages for these two walls.) This means that the difference between maximum and minimum (i.e. unsheltered and sheltered) ventilation is about 25%, which is similar to the figure achieved by *Wilson & Walker* in a comparable case. The 95% confidence limits are quite narrow for winds from the South and SW simply because this is the prevailing wind direction and hence there are more cases of wind influencing ventilation. Overall, this analysis would benefit from studying data for the whole year.

ACKNOWLEDGEMENTS

Many thanks to Derek Whiteside for providing the wind and temperature data as well as the whole house ventilation data.

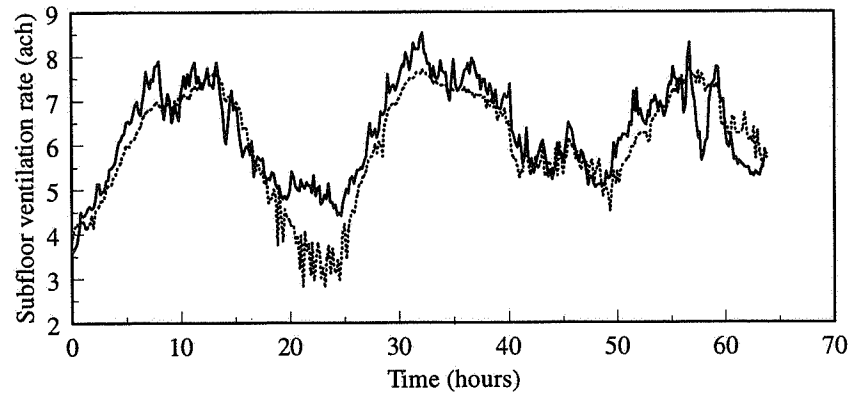
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1a. Measured and predicted subfloor ventilation rates

Run 1: 6-9th August 1993

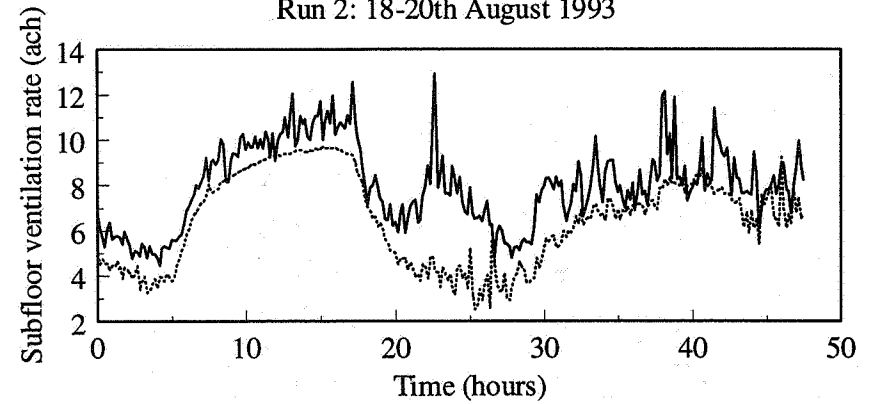


Measured Predicted

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2a. Measured and predicted subfloor ventilation rates

Run 2: 18-20th August 1993

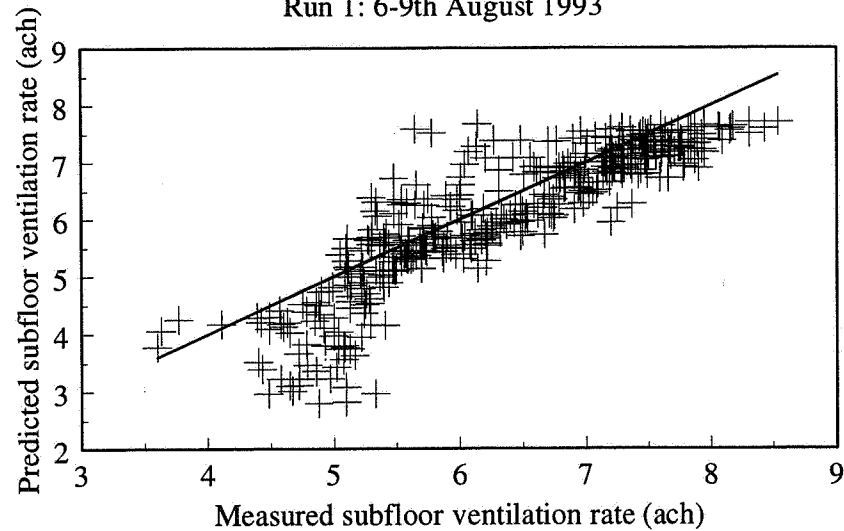


Measured Predicted

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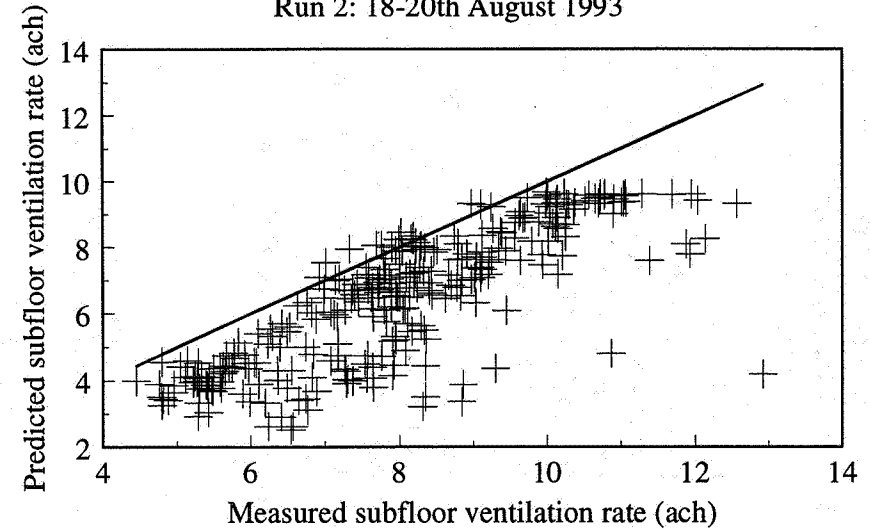
1b. Predicted versus measured subfloor ventilation rate

Run 1: 6-9th August 1993

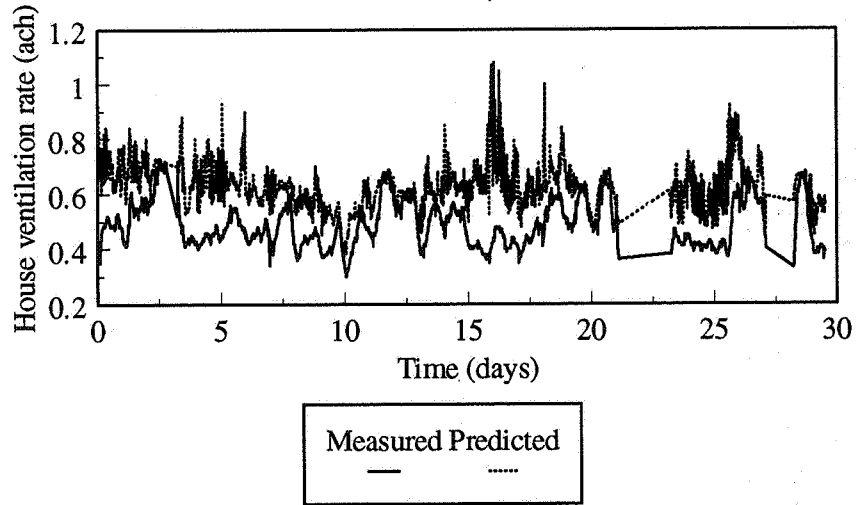


2b. Predicted versus measured subfloor ventilation rate

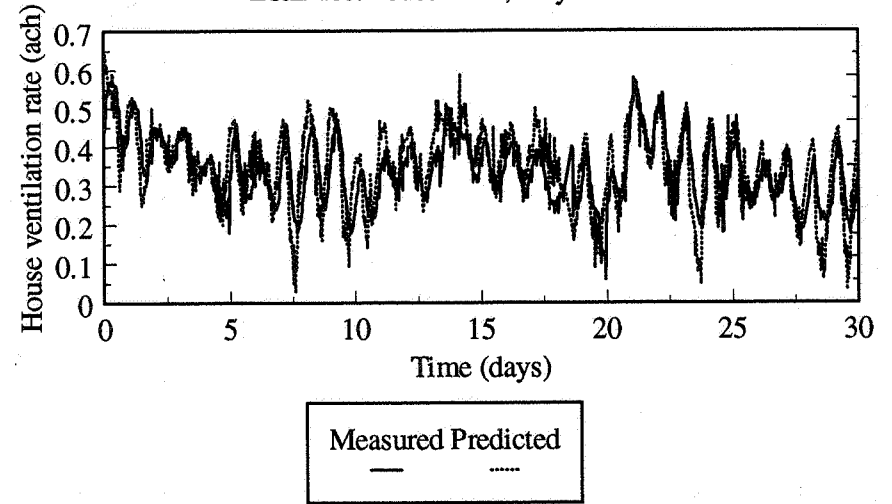
Run 2: 18-20th August 1993



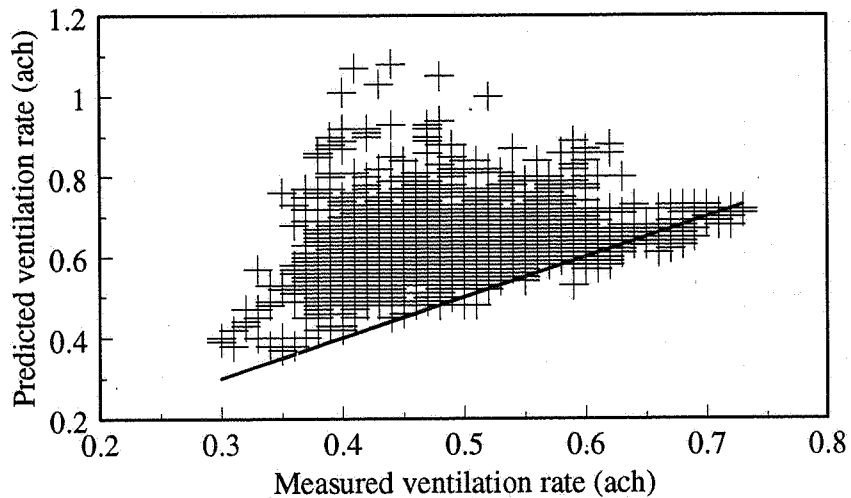
3a. Measured and predicted whole house ventilation rates
E&E Test house no.2, March 1995



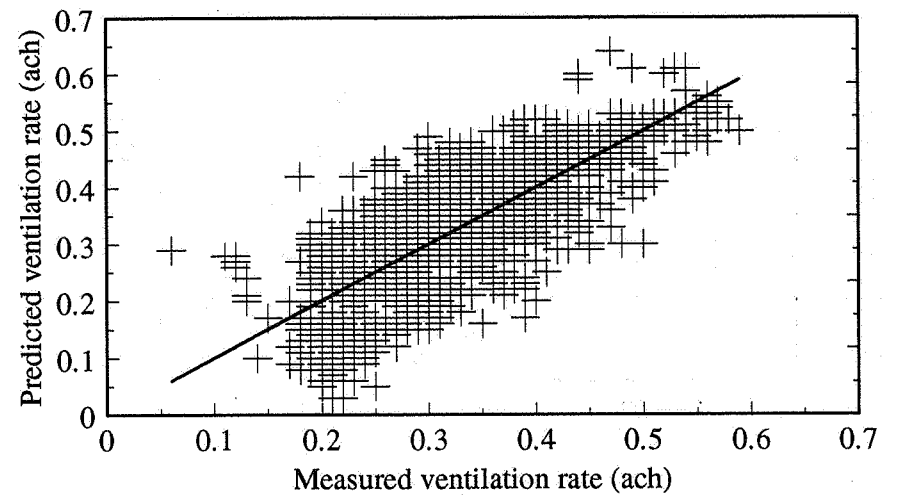
4a. Measured and predicted whole house ventilation rates
E&E Test house No.2, July 1995



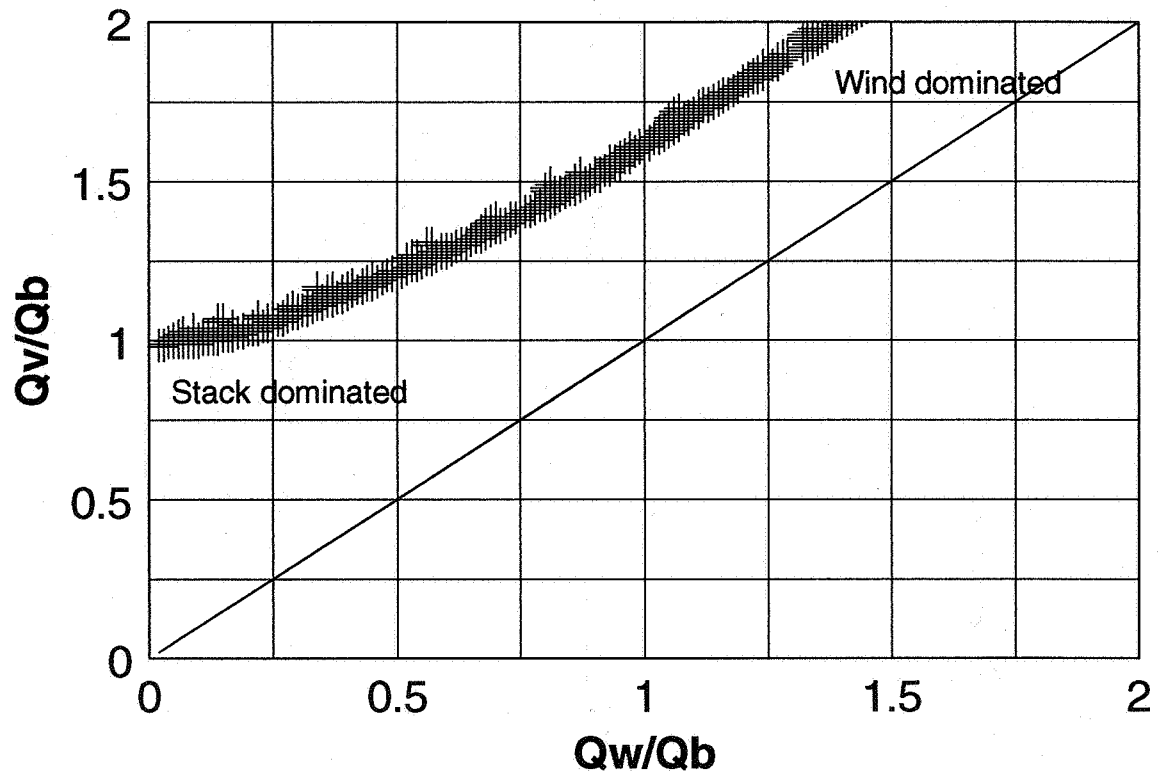
3b. Measured and predicted whole house ventilation rates
E&E Test house No.2, March 1995



4b. Measured and predicted whole house ventilation rates
E&E Test house No.2, July 1995

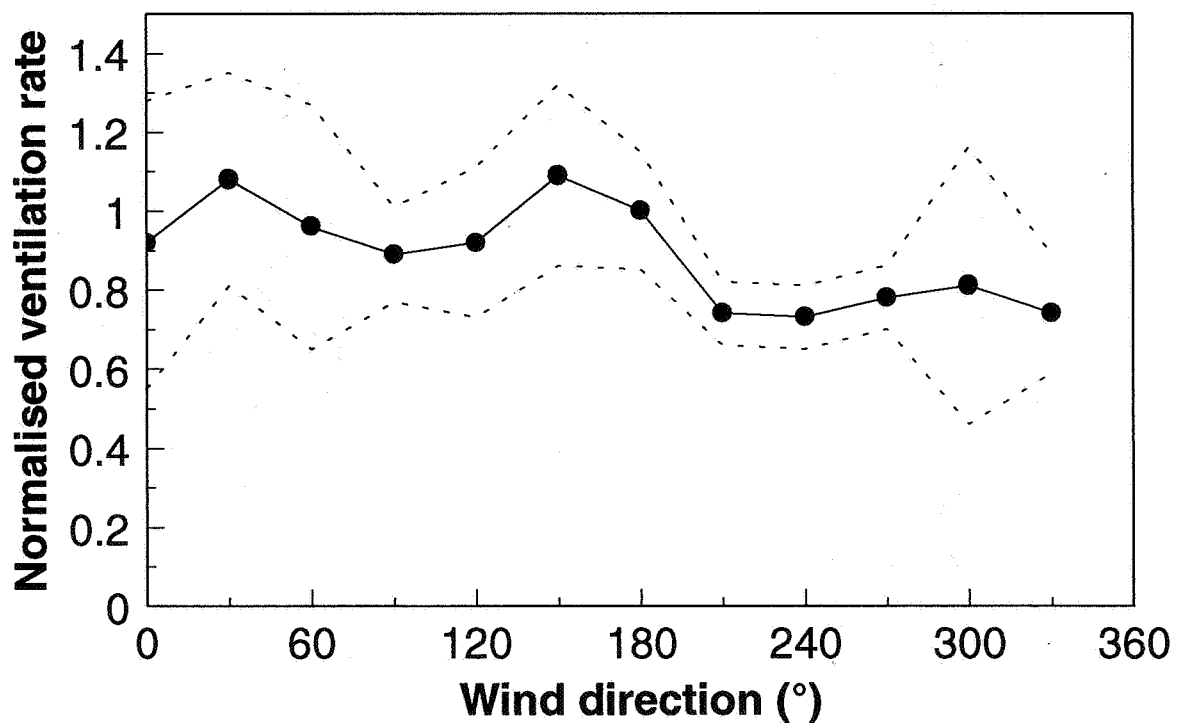


5. Dimensionless plot of ventilation



6. Effect of wind direction on ventilation rates

Test house No.2 (March and July 1995)



OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

(Title)

**EXPERIMENTAL STUDY OF CRACK FLOW WITH VARYING
PRESSURE DIFFERENTIALS**

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Experimental Study of Crack Flow with Varying Pressure Differentials

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Synopsis

Existing experimental techniques for calculating air flow through building cracks are usually based upon relationships derived from experimental studies employing relatively simple procedures. Typically, a fixed pressure difference, ΔP , is established across the crack of interest and then the air flow Q through the crack is determined. Most crack flow equations take the pressure differential ΔP to be steady-state. In reality, the wind forces which generate much of the driving pressures represent highly fluctuating signals. A basic problem is to know what effect a fluctuating ΔP has on the overall air flow through building cracks. The basic aim of this study was to determine the impact of these fluctuating pressures on crack flow equations. Experimental techniques generated sinusoidal pressure fluctuations across a range of fabricated, straight-through cracks, ranging from 1 to 5 mm in thickness. Fluctuation frequencies from 0.5 to 2.5 Hz were used in the investigations. The results suggest that the average flow under conditions of fluctuating pressure differentials does not differ significantly from the flow which would occur if the mean pressure differential were imposed. This implies that the instantaneous flow at any fluctuating value of the pressure differential is that corresponding to the pressure differential on the steady-state flow-pressure curve.

1 Introduction

Equations for calculating the air flow Q through a building crack experiencing a steady pressure difference across it of ΔP are normally presented in either a power law or quadratic format:

$$Q = a (\Delta P)^n \quad (1)$$

$$\Delta P = A Q^2 + B Q \quad (2)$$

where a , n , A and B are experimental coefficients. Examples of this type of approach can be found in the works of Baker *et al.* [1], Fleury *et al.* [2] and Yakubu and Sharples [3]. Potter [4] investigated the possible errors produced by the use of a 'steady-state' crack flow model when the driving pressure differences are actually fluctuating. A series of field measurements were made on a single room with windows in opposite walls, using a flow measuring device and tracer gas analysis. Significant errors were found between the ventilation rates predicted by the steady-state method and those measured in the room. Laboratory studies of ventilation generated by fluctuating pressure differentials were made by Sahin *et al.* [5] on a range of orifice and cylindrical tube shapes. This work did not deal explicitly with building crack geometries, and nor did a study by Rao and Haghighat [6], which used a bundle of straws as their specific flow opening. They developed a fluctuating air flow model which made use of spectral analysis and statistical linearisation techniques and obtained good agreement between the model and their experimental results. A more

detailed discussion of the model is given by Haghighat *et al.* [7]. Although some recent work has been done on fluctuating flows through large openings (see Haghighat *et al.* [8]) there is still a lack of fundamental experimental data for fluctuating flows through building cracks. Kronvall [9] has suggested that for cracks exposed to, say, a sinusoidal fluctuating pressure difference $[\Delta P \sin(\omega t)]$ then equation (1) could be modified to the form

$$Q = a[\Delta P \sin(\omega t)]^n \quad (3)$$

Theories such as that expressed by equation (3) had not been tested experimentally. The main aim of this project was to develop a measurement programme to obtain fluctuating flow and pressure data for building cracks.

2 Method

For the experimental measurements the basic requirements were for a means of generating fluctuating controllable air flows, a crack rig across which fluctuating pressure differences could be produced and instrumentation capable of measuring the range of flows and pressures to be observed in the experiments. The cracks used in the study were mounted in one side of a plenum chamber measuring 1x1x1m. The opposite side of the plenum was connected to a length of ductwork which housed a Furness Controls Laminar Flowmeter FCO96-2000L. A laminar flow device was chosen for measuring flow because of its robustness, high turn-down ratio and ability to measure reversing flows. Discussions with Furness Controls alleviated concerns about the ability of laminar flow devices to follow accurately the fluctuations in the flows. The pressure drops across the laminar flow device and the cracks were monitored by highly sensitive Furness Controls FCO12 digital micromanometers. Fluctuations in the air flow were produced by incorporating in the ductwork a rotatable damper linked to a servo drive motor which, in turn, was driven by a Feedback FG601 signal generator. Frequency fluctuations between 0.5 Hz and 2.5 Hz could be obtained with this arrangement. The duct-plenum was connected to four variable speed fans, which were arranged so as to allow the flow direction to be reversed for some of the experiments i.e. pressurisation or depressurisation of the plenum. The general layout is shown in Figure 1.

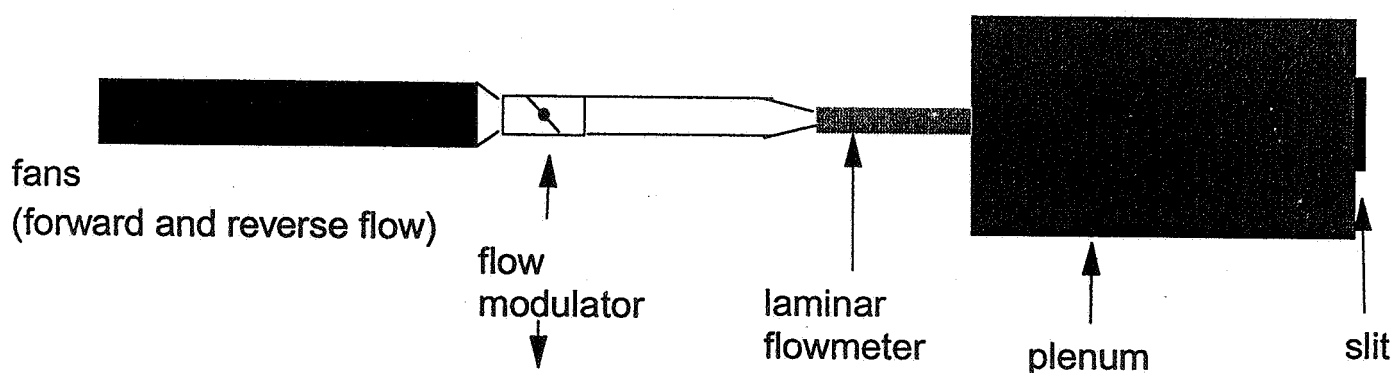


Figure 1 Layout of test rig for unidirectional flow

3 Experimental Measurements

Measurements were made on a range of crack sizes and geometries. Each crack was fabricated from Perspex, and was 500 mm wide and 50 mm deep. For simple parallel sided cracks the heights investigated were 1, 2, 3 and 5 mm. Figure 2 show the geometry of the cracks.

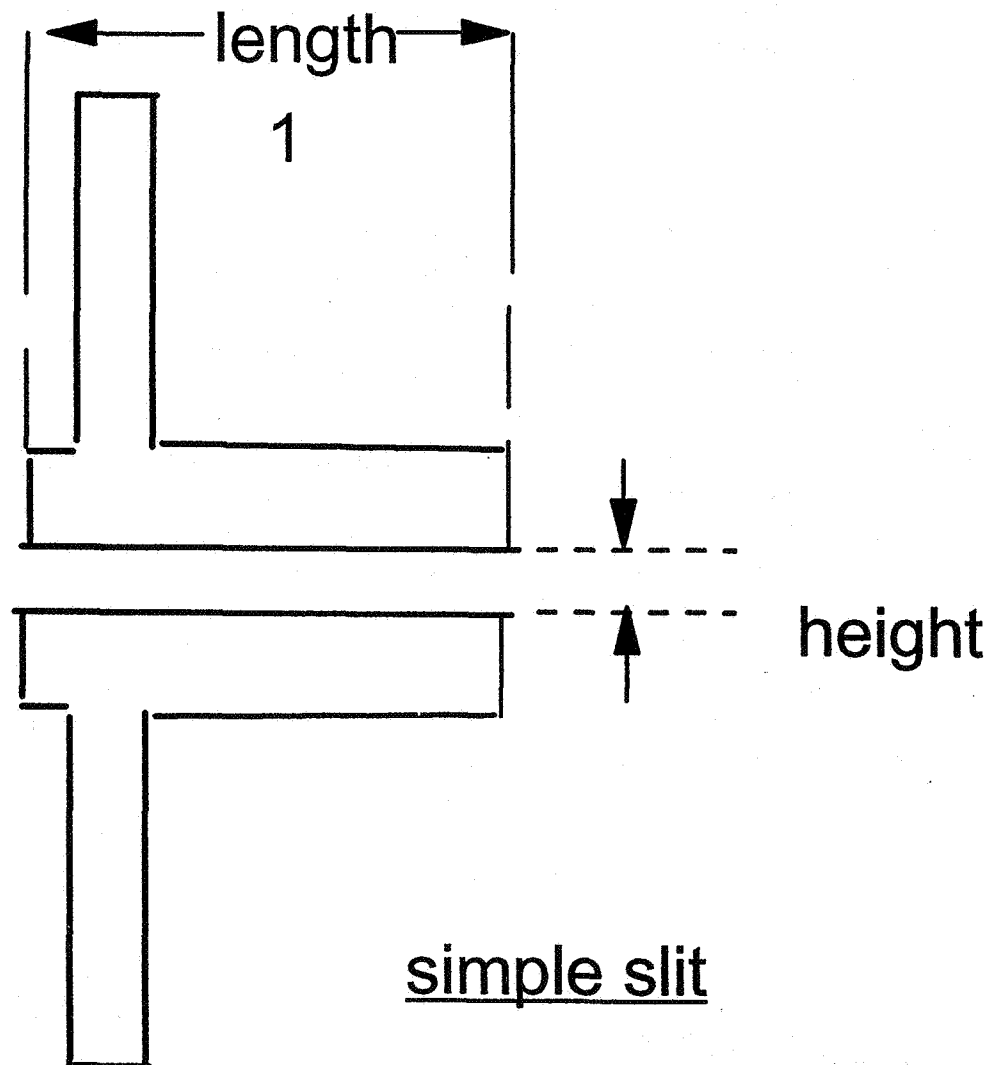


Figure 2 Geometry of simple straight-through crack

Voltage signals from the micromanometers were collected by two Pico analogue to digital converters and stored on computer files. Each measurement record consisted of a 20 second duration of data collection. Four thousand measurements per channel were obtained, and so the sampling interval was 5 milliseconds. It appeared from the measurements that in addition to the fluctuations imposed by movement of the flap valve

there were significant fluctuations at higher frequencies. There were indications of phase lags of pressure behind flow, at the imposed frequency, but the secondary signals and the non-sinusoidal form of the wave were sufficient to make any attempt to scale the amplitudes of the oscillations to allow phase relationships to be examined very imprecise. In order to examine such relationships a computer program 'Spectrum', available from Hensa Micros Archive, and originating from the American National Institute of Standards and Technology, was used (Hess[10]). The original purpose of the program was to observe the phase separation due to small disturbances in a flow system some distance downstream of the point where they were generated, and derive the phase lag despite significant background noise. Consequently the program is ideally suited to the present purpose. In use on reasonably extensive records of pressure fluctuations, the program can derive from simultaneous records of two outputs: amplitude spectra for both channels; power spectra for both channels and cross spectra and phase angle spectra.

4 Experimental Results of Laboratory Fluctuating Crack Flow Measurements

Experiments were carried out with flow both into, and out of, the plenum chamber through the slit, and for several slit sizes. The data were transferred to Lotus 123 spreadsheets, and the use of the 'Spectrum' software allowed phase lag corrections to the flow and pressure data to be applied. The time mean flows were obtained and compared with the pressure-flow curves for steady state operation. Figures 3 and 4 show, for 1 and 5 mm cracks respectively, the results for the time-mean flows imposed on the steady-state calibration curves.

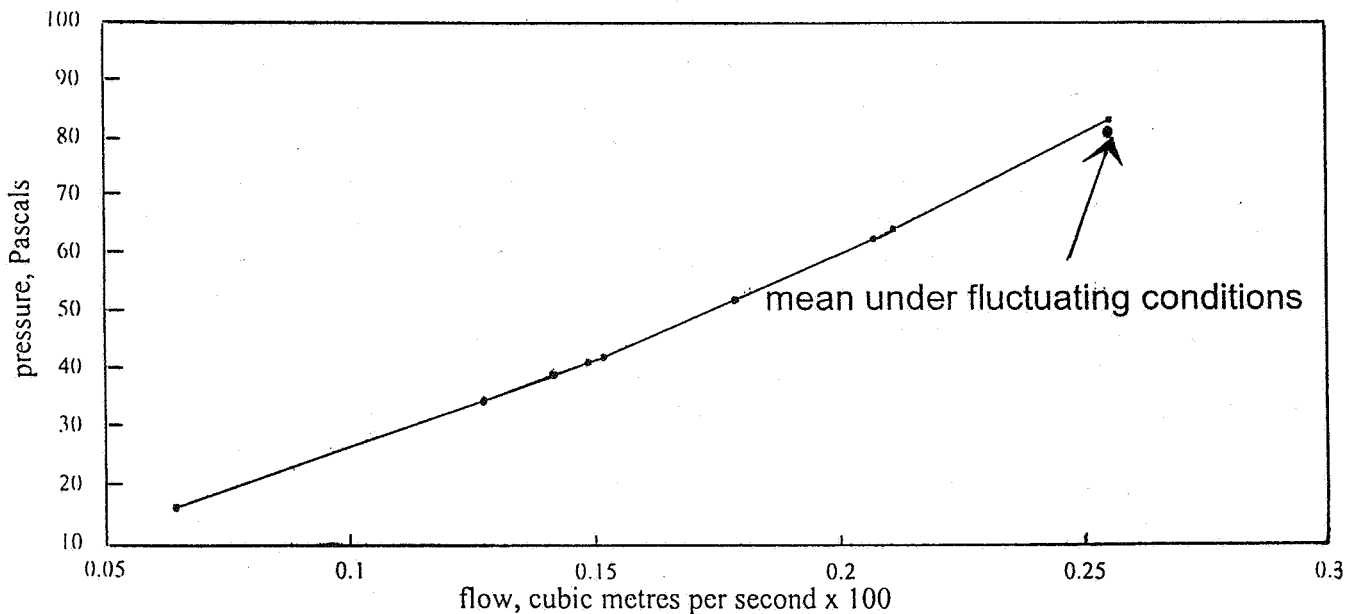


Figure 3 Steady state and fluctuating flow values for a 1 mm slit

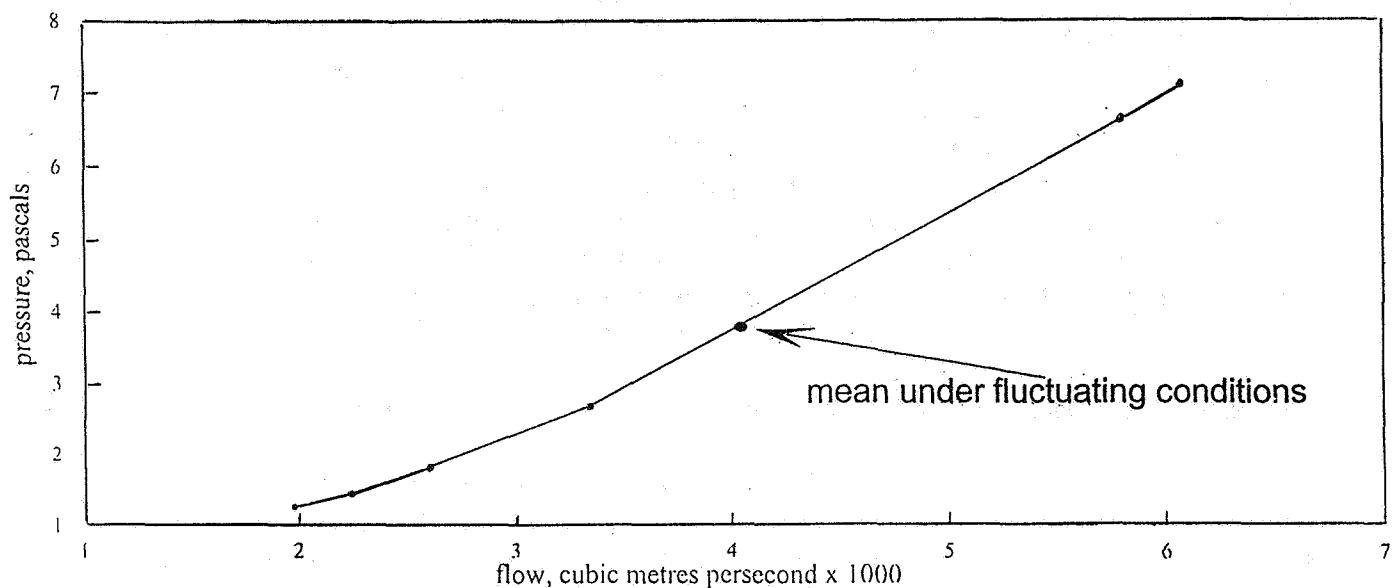


Figure 4 Steady state and fluctuating flow values for a 5 mm slit

For flow in either direction and for all slits, the time average flow under fluctuating conditions was found to be very close to the steady-state calibration curve. In order to examine the possibility that the phase lag was related to capacitive effects within the system, a series of experiments were carried out using a much smaller plenum chamber, with total volume 0.0156m^3 (compared to a plenum volume of 1.0m^3 in the original experiments). These experiments also indicated close agreement between steady-state flow and time-average of fluctuating flow. A further experiment was carried out to determine whether turbulence could influence the flow. A centrifugal fan was mounted in a second 1m^3 chamber and the open side of the chamber was moved close to the face of the other plenum in which a crack was mounted. The fan generated turbulence, and the flow through the crack was measured with constant suction, but with this secondary fan on and off. The flows were the same for the fan on and off within the limits of experimental error, and it is concluded that the large scale turbulence did not affect the flow through the smooth crack significantly. The results have shown that the average flow under conditions of fluctuating pressure differentials does not differ significantly from the flow which would occur if the mean pressure differential were imposed. This implies that the instantaneous flow at any fluctuating value of the pressure differential is that corresponding to the pressure differential on the steady state flow-pressure curve.

5 Experimental Results of Fabricated Crack Mounted in Real Window

A separate investigation was carried out to obtain simultaneous records of the pressure differential between the pressure adjacent to a slit mounted in a real window and the pressure in the laboratory, and the flow through the slit. The 2 mm, 500 mm wide crack was attached to a small plenum and fixed to a sheet of marine plywood which was then fixed into the space left by an open window, as shown schematically in Figure 5. The window was located on the 18th floor of a 19 storey tower block. In these experiments the low flow range of the flowmeter was used, i.e. the pressure tapings nearest the ends of the plates. As the laboratory was on the eighteenth floor of a building in one of the higher parts of Sheffield, a strong wind with minimal effect due to other buildings was readily available. To obtain results requiring analysis in depth it would have been necessary to seal all unspecified apertures in the laboratory and carry out confirmation that these were all sealed by pressure testing. The time allowed did not permit this, and consequently the project was limited to definition of an experimental layout suitable for future work, and confirmation that there was no obvious deviation from the results obtained with the earlier laboratory test rigs described earlier. Some apparent divergence in some of the data points was assumed to be due to either (i) activity in the building leading to pressurisation or depressurisation of the room via cracks around the doors (ii) switches in wind direction leading to a pressure change by enhanced or decreased ventilation via cracks in the wall at right angles to the wall in which the crack was mounted (the laboratory was at the corner of the building) (iii) gusting leading to flexing of the windows remote from the test crack and consequent volume changes not reflected fully in the pressure differential across the crack.

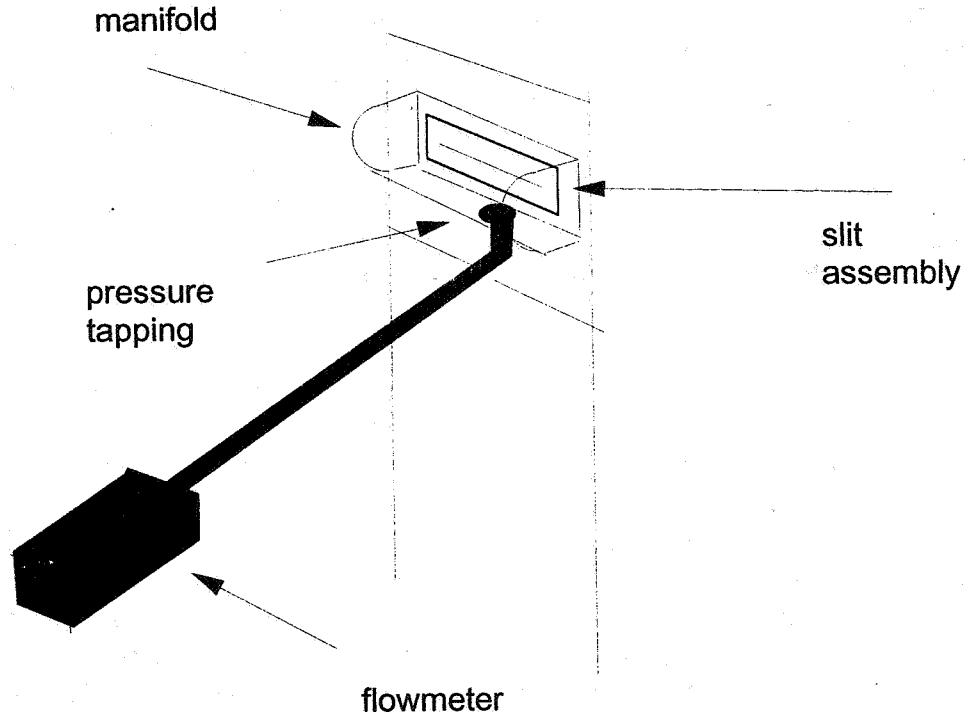


Figure 5 Crack flow arrangement for open window measurements

6 Conclusions

This investigation has been very specifically aimed at the definition of fluctuating pressure-flow relationships for a crack in the wall of a building. It has shown that the flow through such a crack is not materially affected by fluctuations in the local pressure differential, and that the flow over a period of time under conditions of fluctuating flow can be satisfactorily represented by the flow for the equivalent time-average pressure differential.

This conclusion contrasts with the observations by Potter [4], who found that room exchange rates were affected markedly by fluctuating pressures. His measurements were made with nitrous oxide tracer concentration decay as the monitor of exchange, and although he remarks that there are considerable potential uncertainties in the results, there are also potential reasons why his observations are valid reflections of the divergence between the volume exchange implied by the static flow calibration of the leaks and that implied by the decay in nitrous oxide concentration. Firstly, he notes that the spurious leaks were sealed with plastic sheeting: secondly, homogeneity within the room must inevitably take some considerable time when tracer-free air is admitted. The comment about plastic sheeting arises because such sheets could be flexible, and depending on the degree of flexibility and the direction in which the leakage path it closes lies, can oppose or support transfer of air through the test leak.

The delay in mixing can be discussed as a generality for any system: the air entering has momentum, and may indeed be hotter or colder than the air in the room and rise or fall - but it will retain some degree of unmixedness for some period, and will move away from the point of entry. If reverse flow occurs, the air leaving will be undiluted by the air entering in the previous half-cycle (and there is no reason to suppose that a stream of progressively mixed air can build up representing the mixing history over several cycles). The loss of nitrous oxide could well, therefore, exceed the rate predicted for perfect mixing. A further complication could arise if the air entering were cooler than the air inside: in this case warming and expansion lags may occur. If the admitted air is approximated as a series of expanding volumes then this could lead to a rising internal pressure which eventually reaches steady state, but could lead to faster exchange whilst doing so.

Overall, the results from the project are reassuring for air flow modellers. The pressure fluctuations which are known to occur in the natural environment need not be considered as a potential source of error when steady-state crack flow algorithms are used in models. It appears that the relatively high resistance to flow offered by narrow cracks effectively damps out the impact of the fluctuations on the observed flows, allowing time-averaged mean values (i.e. steady-state) to be used. The modelled and observed fluctuating flows through large openings obviously represent a physically very different flow environment to that of narrow cracks. It would be an area of future research interest to investigate how and under what geometric and physical conditions the transitions take place for fluctuating flows from the insensitive small cracks to the sensitive large openings.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

CONTAM96: A MULTIZONE AIRFLOW AND CONTAMINANT DISPERSAL MODEL WITH A GRAPHICAL USER INTERFACE

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(Full paper not available at time of print)

ABSTRACT

CONTAM96: A Multizone Airflow and Contaminant Dispersal Model with a Graphical User Interface

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The latest version in the CONTAM family of airflow and contaminant dispersal models, CONTAM96, has recently become available. CONTAM96 is an easily-used contaminant analysis program combining state-of-the-art algorithms for modeling airflow and contaminant dispersal in multizone buildings. CONTAM96 employs a graphical interface for entering data and viewing the output of a simulation. The interface is based on a SketchPad upon which the user draws building floor plans and places various symbols representing building features relevant to the calculation of airflow and contaminant dispersal. The SketchPad is used to establish the geometric relationships of the relevant building features, producing a simple illustration of the building that represents the underlying mathematical model. This illustration describes the connection between building zones using icons to represent the zones, openings between zones, ventilation system components, and contaminant sources and sinks. The SketchPad brings up data entry screens used to define the mathematical characteristics of various building features, e.g., leakage areas of openings and contaminant source strengths. After performing a simulation, the flows and pressure drops at each opening are presented on the SketchPad. Transient contaminant concentrations can be displayed as separate graphs. In contrast to previous versions of CONTAM, CONTAM96 has the following new features: the ability to perform year-long simulations and exposure analysis, and a duct model. CONTAM96 requires a 386-class (or higher) PC-compatible computer with a math coprocessor, VGA graphics and MS-DOS. The program can also run in a DOS window under Microsoft® Windows95.

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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VENTILATION PERFORMANCE EVALUATION USING PASSIVELY-GENERATED CARBON DIOXIDE AS A TRACER GAS

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Ventilation Performance Evaluation Using Passively-Generated Carbon Dioxide as a Tracer Gas

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Synopsis

Tracer gases are commonly used to evaluate the performance of ventilation systems. One way to reduce the time, complexity, and cost of such experiments is to use the carbon dioxide generated by occupants as a tracer gas. In this paper, a method for using the carbon dioxide generated by occupants as a tracer gas for determining the effective supply air flow rate to a zone or the relative air-change effectiveness of a zone is described. The approach is to make use of a model of the accumulation dynamics and a model of the way that occupants generate carbon dioxide. The carbon dioxide accumulation dynamics are modeled as a first-order perfect-mixing process with bypass. The rate at which occupants generate carbon dioxide is modeled as an integrated random process. Sudden changes in the rate at which carbon dioxide is generated in a zone that occur when occupants enter and leave are detected using statistical techniques. The detector is used to adjust the gains of an estimator that determines either the effective supply air flow rate or the relative air-change effectiveness and also the rate at which occupants generate carbon dioxide. This can be done even when the rate at which occupants generate carbon dioxide changes with time and when the concentration of carbon dioxide in the supply air changes with time. The performance of the strategy is demonstrated with experimental results on a room in a working building.

1 Introduction

Ventilation is the process of diluting contaminated air with uncontaminated air. The rate that contaminants may be diluted depends on the rate at which uncontaminated air is supplied, the air distribution pattern in the ventilated space, and the nature and location of the contaminant. This paper describes a new method for measuring the effective supply air flow rate or the air-change effectiveness, which is a measure of the air distribution pattern in a ventilated space.

Although duct flow rates in mechanically ventilated buildings may be measured with a probe or sensing apparatus, measurement of air distribution in a ventilated space generally involves the use of tracer gases. There are numerous methods and procedures for using tracer gases to evaluate the air distribution pattern in a ventilated space. Common techniques are the "step-up" or "step-down" tests [1]. The use of multiple tracer gases has been proposed in [2]. Another method involves introducing the tracer from a number of low-intensity constant emitters distributed throughout the space [3].

There are three characteristic problems with these techniques. The first is that most of the commonly-used tracer gases are either slightly toxic or environmentally unfriendly, so they must be used in low doses and their use may be resisted. The second problem with these techniques is that a single experiment takes a long time. A step test may take several hours to conduct, and a constant-emitter test may take several weeks. The third problem is that they are labor intensive. They required a skilled technician to conduct the test, and the constant emitter technique requires that the detectors be sent to a laboratory for analysis.

Carbon dioxide generated by building occupants has been proposed as a tracer gas (e.g., [4, 5]). The use of passively generated carbon dioxide eliminates the first characteristic problem because no foreign tracer gas is added to the air. However, these methods may take a long time to conduct, and some only evaluate the ventilation performance when the space is unoccupied. Also, they are labor intensive because the carbon dioxide source strength cannot be measured directly. Instead it is inferred from the number of occupants.

In this paper a new method for evaluating the performance of ventilation systems is proposed. It is a model-based technique that uses carbon dioxide generated by occupants as a tracer gas. The number of occupants need not be known because the carbon dioxide source strength is estimated as a by-product of this technique. This feature allows the method to distinguish between occupied and unoccupied conditions automatically. In comparison to other tracer gas techniques, this method is fast. It can also be easily automated, which means that it can be used to continuously monitor ventilation performance.

2 Model

The technique is based on a single-zone, perfect-mixing model of a ventilated space. This does not imply that the method only applies to single-zone buildings. It may be applied to one zone of a multi-zone building. Figure 1 shows a schematic diagram of a single zone. The bypass factor is used to model the effects of imperfect mixing. The differential

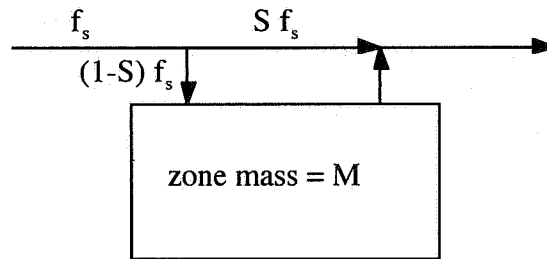


Figure 1: Estimated and actual carbon dioxide source strength.

equation describing the accumulation and dilution of carbon dioxide in this system is

$$M\dot{\omega}(t) = (1 - S)f_s(t)(\omega_s(t) - \omega(t)) + r(t) \quad (1)$$

Note that this model does not assume that the supply air flow rate is constant, nor does it assume that the supply concentration is constant. Therefore, this model may be applied to variable air volume (VAV) systems and to individual zones of a multi-zone system. It has been shown in [6] that the quantity preceding the supply air flow rate is the relative air-change effectiveness.

$$\epsilon_r = 1 - S \quad (2)$$

The effective supply air flow rate is the product of the relative air-change effectiveness and the supply air flow rate.

$$f_{s_e} = \varepsilon_r f_s(t) \quad (3)$$

In order to estimate the source strength and detect large changes in the source strength, a model of the behavior of the source is needed. It is assumed that the source is a random walk that jumps from time to time. The random walk accounts for small variation in carbon dioxide generation that occur as a result of slowly varying metabolic processes or slight changes in workload. The jumps primarily account for the fact that people will enter and leave the space, although they also model the large change in metabolic activity accompanied by changing from a strenuous to a non-strenuous task or vice-versa. Mathematically the source behavior can be described by the following differential equation

$$\dot{r} = w_r \quad (4)$$

where w_r is an uncorrelated random process that contains impulses.

In order to model slow time-variations in the flow parameter, it is assumed that the flow parameter is also a random walk. When the flow parameter being estimated is the effective supply air flow rate, then the mathematical model is

$$\dot{f}_{s_e} = w_f \quad (5)$$

When the flow parameters being estimated is the relative air-change effectiveness, then the mathematical model is

$$\dot{\varepsilon}_r = w_\varepsilon \quad (6)$$

3 Estimation Technique

The model described in the previous section describes the system behavior in continuous time. However, the estimation technique is implemented in discrete time with a micro-processor, so a discrete-time model must be derived from the continuous-time model. The following difference equations describe the accumulation dynamics and source strength in discrete time

$$\frac{M}{2T}(\omega(k+1) - \omega(k-1)) = f_s(k)[\omega_s(k) - \frac{\omega(k+1) + \omega(k-1)}{2}] + r(k) \quad (7)$$

$$r(k+1) = r(k) + Tw(k) \quad (8)$$

It is assumed that the mass of air in the space is known. Therefore, the estimation problem is to simultaneously estimate the source strength and either the effective supply air flow rate (when the supply air flow rate cannot be measured) or the relative air-change effectiveness (when the supply air flow rate can be measured).

3.1 Kalman Filter

A Kalman filter generates an optimal estimate of the state of a linear, stochastic system given all information up to and including the current time. It is beyond the scope of this paper to delve into the details of Kalman filtering theory. The interested reader is referred to [7] for a complete description of the Kalman filter and its application.

For a ventilation system and gas source such as described above, it has been shown [8] that equations 7 and 8 can be transformed so that the source strength and the effective flow rate or the effectiveness are state variables. With this transformation, the “measurement” is the mass multiplied by the discrete-time approximation of the derivative. This transformation allows the “state” of the system (i.e., the flow parameter and the source) to be estimated with a Kalman filter. Note that this is a slight abuse of the Kalman filtering theory because the parameters of the measurement vector are noisy, and because the “measurement” noise is correlated. However, it will be shown that even with this abuse of the theory, the results are satisfactory.

3.2 Detector

The Kalman Filter provides an estimate of the covariance of the state estimate. This statistical measure of the uncertainty in the estimate is used to construct hypothesis tests on the innovations (i.e., estimation error) sequence to detect jumps in the state (i.e., the source strength or flow rate or effectiveness).

Whenever the detector is tripped, the gain of the estimator is reset to a large value so that the estimation error is reduced rapidly. This resetting behavior allows the estimator to track variations in the ventilation performance over time in a continuous manner. However, unlike a sensor, the ability of the estimator to track variations in effectiveness, especially large, fast variations, is dependent on the presence of disturbances which will primarily be large changes in the source strength caused by the movement of people into or out of the space.

4 Experimental Results

In this section the system behavior is demonstrated with an experiment on an office in a conventional operational office building. The office has a 9.3 m^2 floor area and a 2.75 m ceiling height. Air is supplied to the office by a constant-volume multi-zone air handling unit with a hot and cold deck. The office is on the perimeter of the building, has a non-operable window on the west wall, and a perimeter heating coil below the window. Air is delivered to the office through slot diffusers on both sides of two fluorescent light fixtures in the ceiling. Air is exhausted from the room to an adjacent hallway through a grille in the door. The operation of the HVAC system was not altered for this experiment, and nothing was done to alter the natural air distribution pattern. During the experiment, the office was unoccupied. The office contained two desks, a bookshelf, and a gas bottle which was used to emulate the coming and going of an occupant. The bottle contained 4% carbon dioxide by volume and was metered with a computer-controlled precision flow control valve. During the “occupied” period, the source strength was a constant value of $1.4 \times 10^{-5} \frac{\text{kg}}{\text{s}}$ carbon dioxide which is the amount that a medium-sized male will generate when standing and performing a light task such as drafting.

The carbon dioxide concentration of the supply air and zone air were measured with a single non-dispersive infrared (NDIR) sensor using a mechanical multiplexing technique. The zone air was sampled at a height of about one meter in close proximity to the return grille in the door. The mechanical multiplexer was controlled by a computer. The time between each measurement was 30 seconds, so the time between measurements of the supply concentration was 60 seconds as was the time between measurements of the zone concentration.

The estimator was designed with the assumption that the flow rate and source strength are constant (i.e., $w_r = 0$, $w_f = 0$). The ability to track time-variations in the flow rate and source strength was provided by the detector tripping and resetting the

covariance of the estimator. With this estimator design, the estimates are the average values of the parameters since the estimator was last reset. The detector was designed so that it would completely reset the estimator when the number of trips was odd, and it would partially reset the estimator when the number of trips was even. A partial reset means that only the element of the covariance associated solely with the source strength was reset. This was done to guarantee sufficient signal excitation to estimate the flow rate, while at the same time allow the estimator to track variations in the flow rate.

Prior to the experiment, the supply air flow rate from each light fixture diffuser was measured twelve times with a flow capture hood. The mean and standard deviation of the volume flow rate from the west light fixture were $0.0370 \frac{m^3}{s}$ and $0.00465 \frac{m^3}{s}$ respectively, while the mean and standard deviation from the east light fixture were $0.0404 \frac{m^3}{s}$ and $0.00481 \frac{m^3}{s}$ respectively. Therefore, the mean and standard deviation of the total supply air flow rate were $0.0774 \frac{m^3}{s}$ and $0.00670 \frac{m^3}{s}$ respectively. The air density was $1.16 \frac{kg}{m^3}$, so the total supply air mass flow rate was $0.0898 \frac{kg}{s}$ and the standard deviation was $0.00777 \frac{kg}{s}$. The estimator was initialized with these values. The variance of the initial estimate of the effective supply air flow rate was set equal to the variance in the measured flow rate using the flow capture hood.

Figure 2 shows the supply and zone concentrations as a function of time during the experiment. The dip in the supply air flow rate 130 minutes into the experiment is due to the evacuation of the surrounding offices during lunch time. Figure 3 shows the

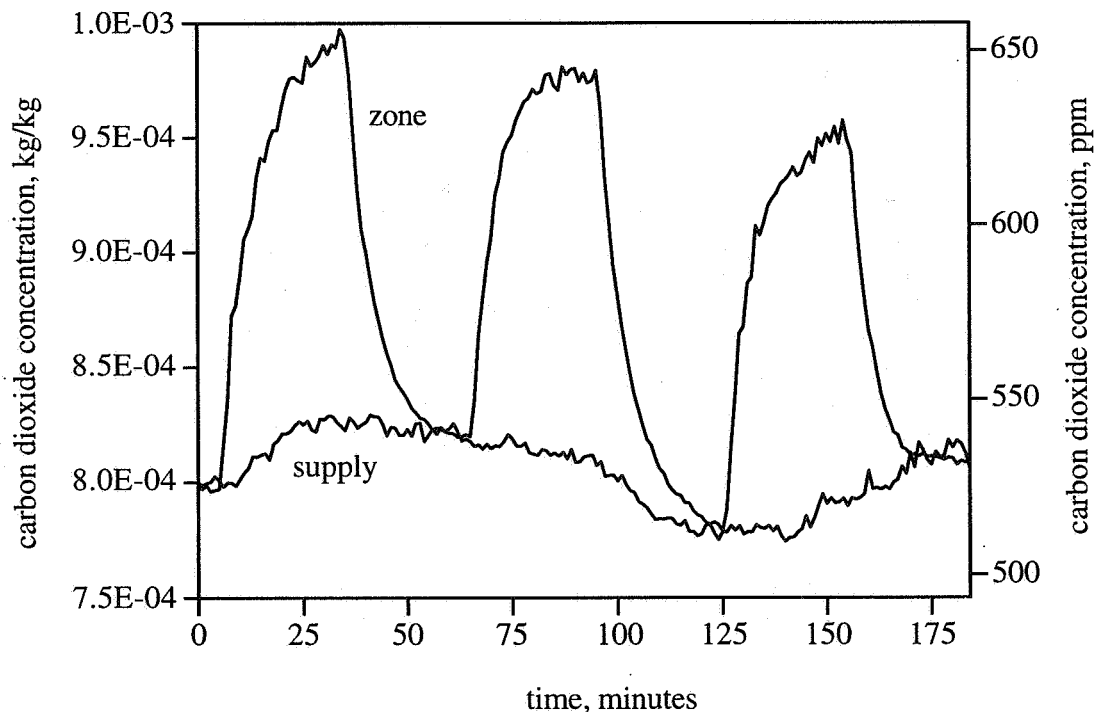


Figure 2: Zone and supply concentrations of carbon dioxide during the experiment.

estimated and actual source strength during the experiment. The figure shows the short lag in the estimate relative to the actual value which occurs because it takes two samples before the detector determines that a jump has occurred. After the detector is tripped, the estimates converge rapidly to a value nearly identical to the actual source strength.

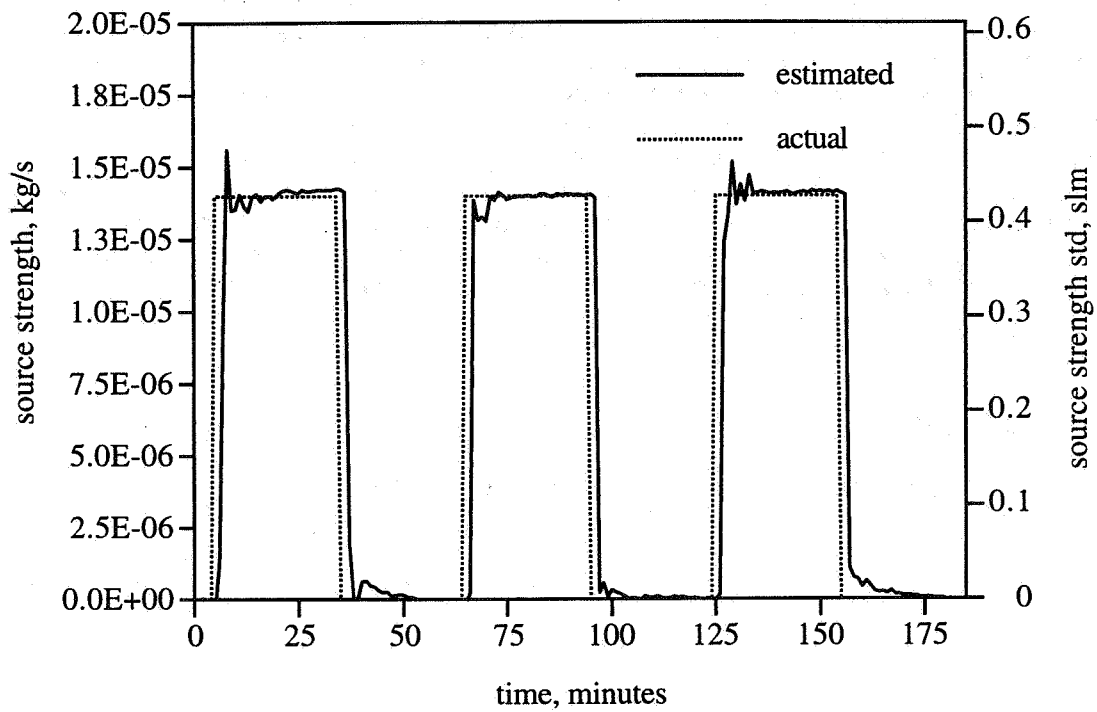


Figure 3: Estimated and actual carbon dioxide source strength.

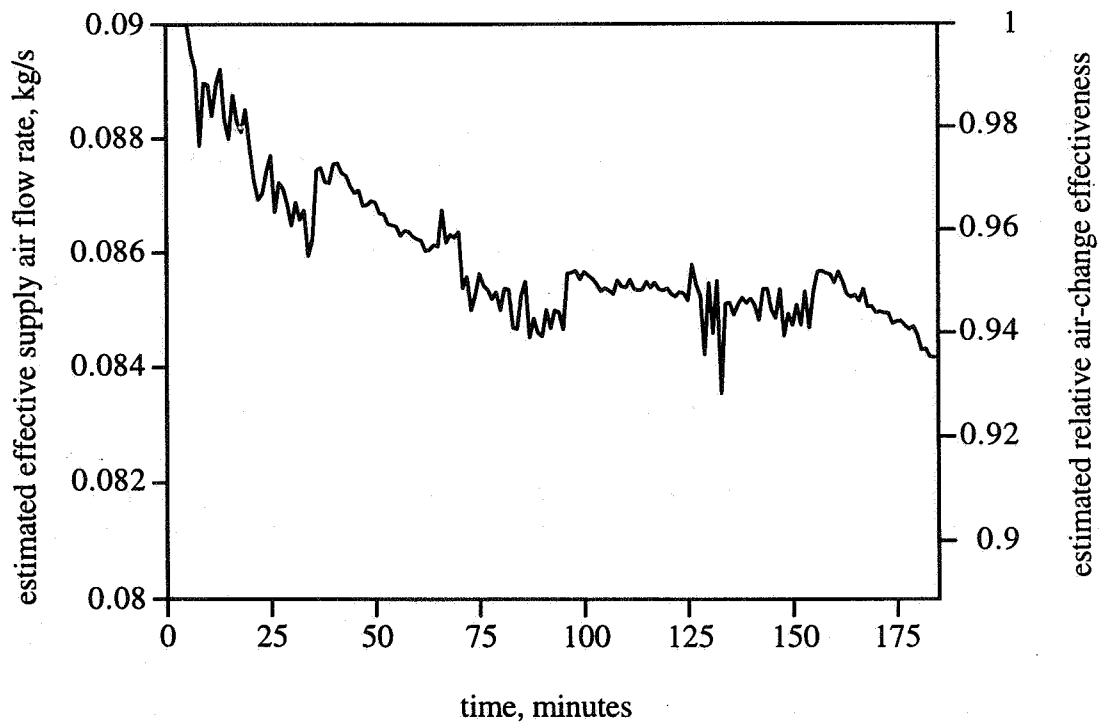


Figure 4: Estimated effective supply air flow rate, and estimated relative air-change effectiveness.

Figure 4 shows the estimated effective supply air flow rate. The figure also shows the estimated relative air-change effectiveness assuming that the supply air flow rate did not change during the test. Note that there is no way of knowing if the decrease in the effective supply air flow rate is due to a decrease in the flow rate or a decrease in the effectiveness. The data is presented this way to demonstrate that it may be possible to continuously monitor the effectiveness if the supply air flow rate is known. One feature of the estimates is that they are noisier when the source strength is non-zero than when it is zero. This is partly because the zone concentration measurement is noisier when the source strength is non-zero than when it is zero, and partly because the partial resetting which occurred at the transition to occupied did not increase the gain associated with the effective flow rate as much as it increased the gain associated with the source strength.

5 Conclusions

A new method of evaluating ventilation performance is described and demonstrated with an experiment. The method has the following advantages over other methods:

1. It uses carbon dioxide generated by occupants rather than a foreign tracer gas.
2. It can be automated, so it is not labor intensive.
3. It is fast.
4. It can distinguish between occupied and unoccupied conditions.
5. It can provide a continuous measure of ventilation performance.

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OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

**17th AIVC Conference, Gothenburg, Sweden,
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PREDICTING AIR FLOWS IN NATURALLY VENTILATED BUILDINGS WITH COMPUTATIONAL FLUID DYNAMICS

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(Full paper not available at time of print)

PREDICTING AIR FLOWS IN NATURALLY VENTILATED BUILDINGS WITH COMPUTATIONAL FLUID DYNAMICS

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Computational fluid dynamics (CFD) simulation software is being used increasingly to predict air flow patterns within buildings, and there is no doubt that it has a very important part to play in the optimisation of internal environments if it is used appropriately. It is now considered that the first option for a building design is to ventilate naturally, and for such buildings the designer must be carefully to provide a controllable environment which can avoid overheating in summer and excessive, uncomfortable ventilation in winter. CFD is therefore a very attractive tool in optimising the design of naturally ventilated buildings.

Like any other application area for CFD, simulation of natural ventilation poses its own challenges in CFD modelling. Users need guidance on how to avoid potential pitfalls in the modelling, particularly as CFD is increasingly being run by users who are not CFD specialists. Unfortunately there is little such guidance currently available. This paper describes the first stage of work carried out to address this need.

This work has compared results from various participants who used their chosen codes to predict the airflows in some simple test cases. By keeping the test cases simple, there is less room for confusion in interpreting and comparing the results. With help and advice provided by a steering group of CFD and building experts, the cases were designed to represent the major features influencing the ventilation airflows, without the detailed geometry of a real building. The initial trials were based upon an atrium-like geometry, with the first test having fixed airflows as boundary conditions, and the second allowing the CFD to calculate the boundary airflows as part of its solution procedure.

OPTIMUM VENTILATION AND AIR FLOW CONTROL IN BUILDINGS

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

(Title)

Energy Efficiency in Office Buildings An Energy and System Analysis Study

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SYNOPSIS

While the use of heat energy has decreased since the middle of the 1970's the use of electricity in the Swedish stock of commercial buildings has increased dramatically. In the average Swedish office building, roughly 30 % of all electricity is used for heating, ventilation and air-conditioning (HVAC). Another 30 % is used for lighting, 20 % for office machines, and about 20 % for other loads.

In order to study the use of electricity in Swedish office buildings in detail, the Swedish Council for Building Research initiated four monitoring and building simulation projects in 1989. The project buildings represent different ages and different HVAC technologies.

Based on a combination of hourly monitored data during four years, building simulations using the DOE2 program have been carried out. In two of the buildings, energy conservation measures have been implemented too.

An interesting aspect in our study is the so called load factor. Load factors for lighting and other electric loads are evaluated. Taking this into account, you can optimize the design of the air flow rates and the cooling systems and reduce the investment and operational costs for the HVAC systems. Thanks to a correct HVAC design you may also reduce the negative impact on the environment by less use of CFCs etc.

Our studies indicate that the load factor variation in Swedish office buildings, during a day or a week, diverge substantially from the schedule for lighting and office machines etc. according to the ASHRAE 90.1-1989 Standard ⁶. Regarding office machines etc., the installed power density also diverges to a great extent.

In order to classify ventilation systems with respect to energy efficiency, some different key values are defined. One is the so called SFP value (Specific Fan Power). In some Swedish projects, the goal has been to reach a level of 1.5 kW/(m³/s) or less. However, it is not easy to reach that level in existing buildings. SFP values for Swedish office buildings based both on rated and measured data are presented in this paper.

By implementing appropriate energy conservation measures (ECM), the use of electricity and heat energy may be considerably reduced and also lead to a better indoor climate. However, it is important to have an overall view of the building and its systems and to take all effects of interacting ECMs and systems into account.

1 INTRODUCTION

While the use of heat energy has decreased since the middle of the 1970's, the use of electricity in the Swedish stock of commercial buildings has increased dramatically. This increase was most pronounced in the 1980's.

The use of electricity in the Swedish commercial building sector amounts to about 25 TWh annually. In the average Swedish office building roughly 30 % of all electricity is used for HVAC. Another 30 % is used for lighting, 20 % for office machines, and about 20 % for other loads ¹.

In order to study the use of electricity in Swedish office buildings in detail, the Swedish Council for Building Research initiated four intensive monitoring and building simulation projects in typical office buildings in 1989. These buildings, owned by four Swedish insurance companies, represent different HVAC technologies. Two of them are representative of the early 1980's and the other ones of the late 1980's.

During this before-after study period, we also investigated some twenty other office buildings with respect to the energy use both in total and for different loads such as for lighting, office machines etc. The HVAC system, control system and lighting system were also studied. Several energy and power measurings as well as other kinds of measurements were carried out.

Detailed results from these studies have up till now mainly been published by the Swedish Council for Building Research ^{2,3}. However, the results have also been used recently in a licentiate thesis as well as in a doctoral thesis, both at Chalmers University of Technology in Gothenburg ^{4,5}.

2 LOAD FACTOR STUDIES

2.1 Definition

An often neglected thing when designing HVAC systems for office buildings is the so called **load factor**. This factor we define as the quotient between the electrical power used at a given moment and the total power installed (rated) e.g., for part of or the whole office building.

The load factor can be said to be compounded of two parts. The first part is the difference between rated power and real power in use for each machine etc. The second part is the fact that all machines and lighting etc. in an office floor etc. are never used simultaneously.

Another expression that will be used in this paper is the **installed power density**. When talking about installed power density for lighting or receptacle (office machines etc.) or both together, we mean the power installed (rated) e.g., for part of or the whole office building, divided by the floor area of immediate.

2.2 Correct Load Factor Consideration Leads to Environmental Benefits

When designing HVAC systems it is very important to consider the load factor in a correct manner. By taking the load factor into account, you can **optimize the design** of the air flow rates as well as the cooling systems. Thus you can reduce the investment and operational costs for the HVAC systems as well as the environmental influence.

You obtain environmental benefits in two ways. First, a correct designed HVAC system will use less energy than an oversized one. Secondly, this will lead to less use of CFCs etc., which will be demonstrated shortly in section 3.3.

2.3 Decreasing Load Factors

Our studies have shown that the load factors in office buildings are often much lower than most consultants and building engineering experts think. Furthermore, the load factors in Swedish office buildings have tended to decrease during the last few years. This fact has been possible to establish as a result of our monitoring project running for several years. There is no reason why our results would not be valid also for office buildings in other countries.

In **Figure 1** results from the four main buildings in our investigation are shown. The load factors in this figure are counted for single office floors and represent **maximum** values during several years of monitoring. The values include lighting and all electrical office machines etc. and are representative of about 700 m² in each building.

Office building	Lorensberg	Svaneholm	Stampen	Lagern
Year 1990/1991	48 %	60 %	40 %	47 %
Year 1995	22 %	33 %	Not investigated	Not investigated

Figure 1 Maximum load factors for single office floors in four typical Swedish office buildings

Office buildings in common are getting more and more equipped with electrical office machines. This is the main reason why the remarkable load factor decrease in the Lorensberg and the Svaneholm office buildings. The great drop for the Lorensberg office building is also a result of the installed occupancy sensor-controlled lighting system.

When the number of personal computers, printers etc. increase, the difference between the total summed up rated power and the power in use also increase i.e., the load factor becomes lower. One reason for this is that all machines and lighting are never used simultaneously.

Based on investigations in several office buildings during a number of years, we also state that the difference between the rated power and the real operating power for each single machine today is often greater than it used to be. Often the operating power is about 30 % of the rated one.

Our detailed studies in the four office buildings have also showed that there normally is a difference regarding the load factor level between summer and winter time. In the summer the load factors

often are somewhat lower, mainly due to the fact that all office staff does not need to use the lighting all day. Of course vacations also to some extent have an influence on the level of the load factor.

Considering load factors, one must also be aware that they will probably increase even more in the future, although most office buildings already today are very well equipped. This statement is based on two trends seen today. First, most office machines become more and more energy efficient in standby periods, thanks to power sleep modes, Energy Star demands, etc. Secondly, the use of office lighting controlled by occupancy sensors will probably become more frequent. Both these circumstances will increase the difference between the installed rated power and the real power demand, and through this the load factor, will be reduced.

Recently, a new project has been started in order to study load factors, not only for single office floors but also for entire office buildings. In the entire office buildings, the load factors are probably even lower than has been shown in this paper. We intend to find out the normal level of total load factors as well as their schedules.

2.4 Comparisons Regarding Installed Power Densities and Load Factors

In this section, results regarding installed power densities and load factors from our four building projects are compared with the ASHRAE 90.1-1989 Standard ⁶.

	Installed power density [W/m ²]				
Office building	Lorensberg	Svaneholm	Stampen	Lagerm	ASHRAE
Office room lighting ¹	16.9	22.1	17.0	12.6	19.4
Office machines etc	21.2	16.0	26.6	20.5	8.1

¹In new Swedish office buildings the installed power density for office lighting is often less than 10 W/m² incl. losses

Figure 2 Installed power densities for single office floors in four typical Swedish office buildings (before ECMs) compared with the ASHRAE Standard

From measured load factor schedules and the installed power density values in **Figure 2**, we can calculate what we may call **power density schedules**. An analogous calculation may also be done based on shedules and power density values according to the ASHRAE Standard.

In **Figure 3**, power density schedules for office machines etc. based on the four project offices, as well as based on the ASHRAE Standard for an average working-day may be seen. In **Figure 4**, a similar comparison is shown, but this time regarding density schedules for office lighting.

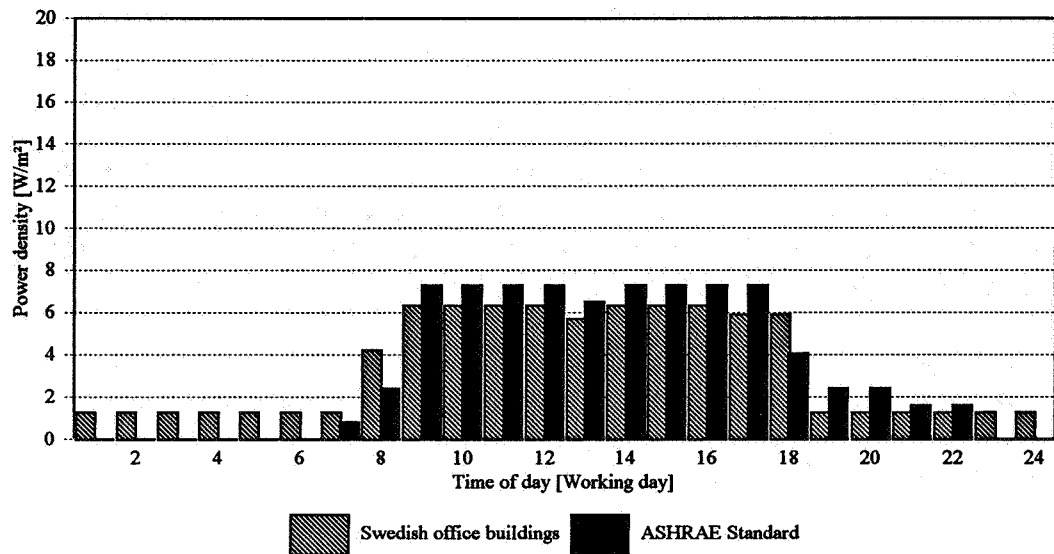


Figure 3 Power density schedules for office machines

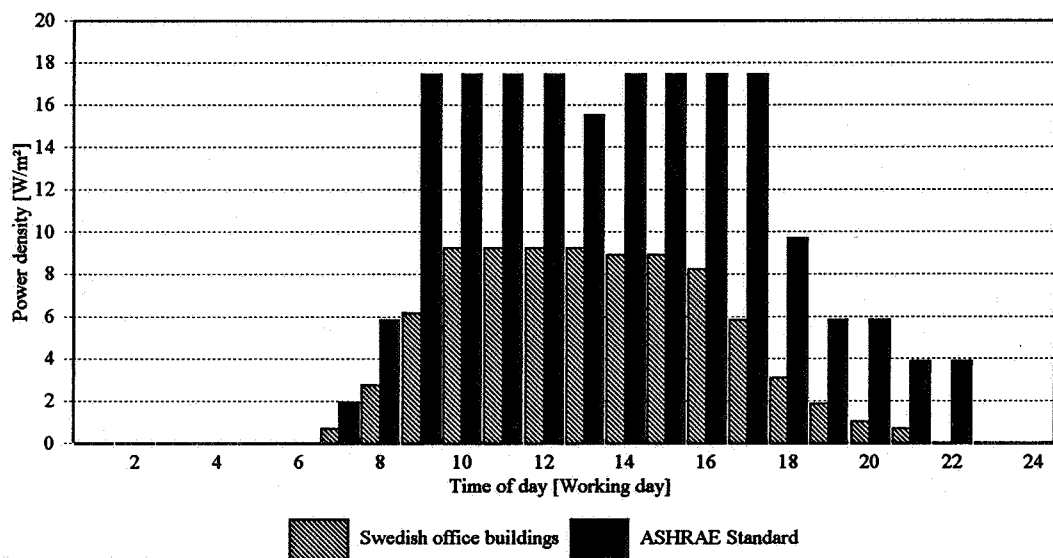


Figure 4 Power density schedules for office lighting

Two main conclusions may be drawn from the results shown in **Figure 3** and **Figure 4**.

Firstly, despite substantial differences regarding installed power density as well as the load factor schedules, the two power density schedules for office machines etc. become quite similar. The daily energy use will be nearly the same in both cases ⁵.

Thus, regarding **receptacles** (office machines etc.) the ASHRAE values may be used also for Swedish office buildings. Please note that in this case **both** the ASHRAE load factor schedule **and** the ASHRAE installed power density must be used.

On the other hand, the second main conclusion is that a utilization of installed power density, and schedule for **lighting** according to the ASHRAE Standard will result in a power density schedule that does not at all correspond to our experiences based on conditions in Swedish office buildings.

In Sweden therefore, a HVAC system fully designed and based on the ASHRAE Standard would probably become oversized.

3 DOE2 SIMULATIONS

3.1 Why Simulate?

When for instance an office building is to be built or rebuilt, it is very often advantageous to carry out computer simulations. Hereby different possible HVAC systems may be analyzed. This is a way to optimize the design and to reduce the life cycle costs for the system chosen. In a rather cheap way you will also get a prediction of the energy and power demands for the building.

3.2 Simulations for the Svaneholm Office Building and Measuring Results

There are several different building energy simulation programmes on the market today. We have used the well known and in many studies validated DOE2-program (version 2.1D) ⁷.

Based on a combination of hourly monitored data during four years, building simulations of the effect of different possible energy conservation measures (ECMs) have been carried out for the Svaneholm office building ^{1,2}. The monitored data represent the most important end-users of electricity such as ventilation, cooling, lighting, computers, etc.

One of our goals was to analyze the real result of the energy savings besides learning more about the system effect of different ECMs.

In **Figure 5**, the results of some of these simulations are shown. The use of electricity as well as of energy for heating could be considerably reduced if both studied measures were to be implemented. Measure no 1 stands for "Installation of energy efficient lighting", and measure no 2 for "More energy efficient HVAC system".

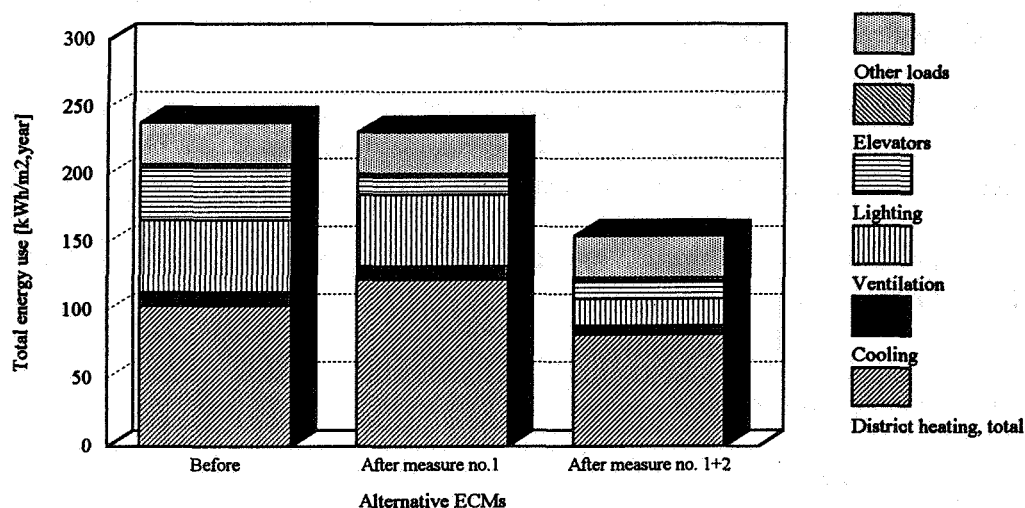


Figure 5 DOE2 simulations of energy use for the Svaneholm office building

These simulations were carried out based on real load factors, determined through the measurements mentioned above in combination with inventories. When simulating buildings to be built, of course you do not have any load factor information etc. for the building. This is one of the reasons why it is of great interest to establish what might be called schedules of normal load factor for different kinds of buildings.

Today in the Svaneholm building, the ECMs have been implemented to a great extent. By means of long time monitoring, we have been able to see that the accordance between simulations and real outcome is very good regarding energy use as well as power use. Although all ECMs are not yet completely carried out, we have reached about 90 % of the predicted energy savings.

In **Figure 6** and **Figure 7**, power profiles are shown for comparable weeks before and after the implemented ECMs. The y-axis represents power and the x-axis represents time. Consequently the different areas in the figures represent the energy use.

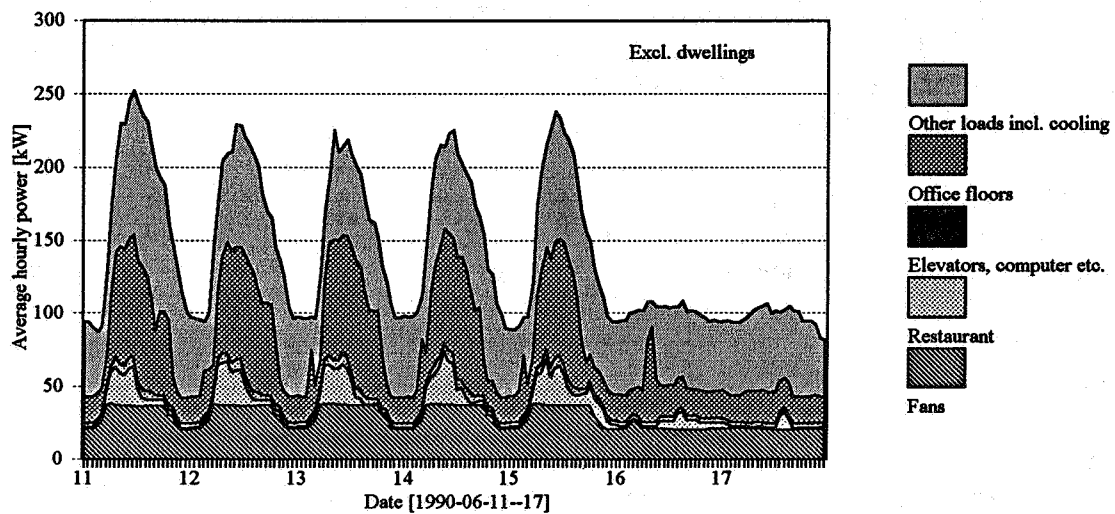


Figure 6 Power profile for the Svaneholm office building **before** implemented ECMs

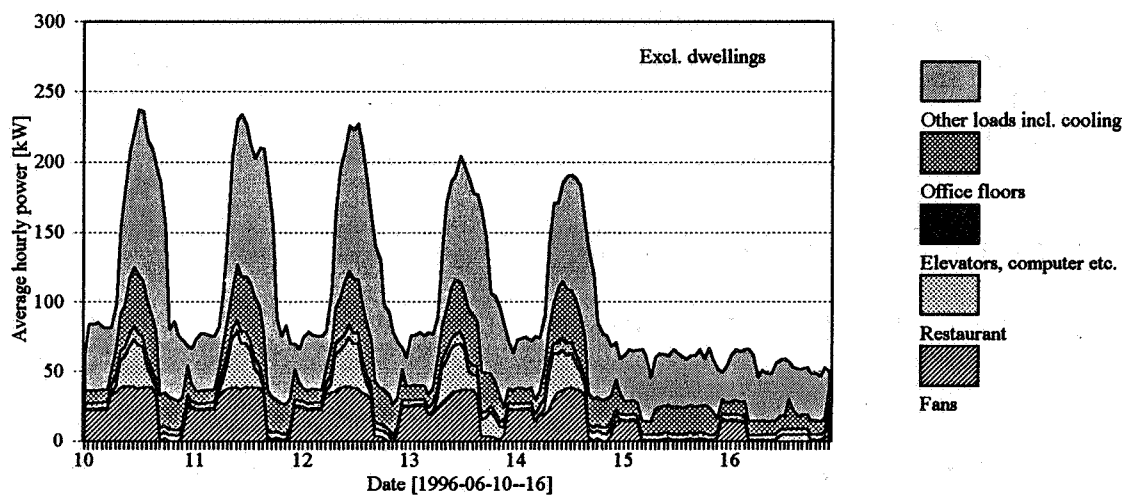


Figure 7 Power profile for the Svaneholm office building **after** implemented ECMs

A comparison between these two figures shows that the energy and power used for **Office floors** (i.e. lighting and plug loads as PCs etc) as well as for **Fans** are substantially reduced. However, it should be noted that all HVAC measures were not yet implemented when the power profile in **Figure 7** was plotted. Due to circumstances not connected to the project, the power use for **Other loads** has increased.

3.3 A Short Example of Load Factor Influence

To show the influence of the load factor level and its time variation some other DOE2 simulations for the Svaneholm office building were carried out. In the basic simulation (left column in **Figure 5**) a load factor schedule based on real measurements was used. In a second serie of simulations a load factor schedule according to the ASHRAE 90.1-1989 Standard ⁶ was used. In both cases the same criteria for satisfactory indoor climate was set.

The ASHRAE simulations indicated that more energy was to be used for ventilation as well as for the cooling system. The electric power demand for cooling equipment was about 10 kW higher according to the ASHRAE simulation. The HVAC system would probably become oversized if it was designed in accordance with the ASHRAE schedule. This example demonstrates the importance of using correct and relevant load factors when simulations are to be carried through and, above all, when HVAC systems are to be designed in reality.

As mentioned before, correct design of the HVAC systems does not only result in reduced investment and operational costs, but also in environmental benefits such as less use of CFCs etc.. The difference regarding the demand for cooling between the ASHRAE simulation and the basic simulation for the Svaneholm building corresponds to a difference concerning the CFC volume of about 10 kg.

Of course, a use of relevant load factor schedules is also very important when ECM simulations is to be done, as e.g. those shown earlier in **Figure 5**. Often, the same type of ECM lead to quite different energy savings if the simulations are based on different load factor schedules. The higher an office's internal loads (lighting, office machines etc.) are, the more important is a correct load factor assumption.

4 SPECIFIC FAN POWER MEASUREMENTS

In order to classify ventilation systems with respect to energy efficiency, different factors are used. One is the so called SFP value (Specific Fan Power). The SFP value is defined as the total fan motor power installed divided by the total air flow through the building.

In some Swedish projects the goal has been to reach a SFP level of 1.5 kW/(m³/s) or less. However, it is not easy to reach that level in existing buildings even when extensive renovation work is to be carried out.

In **Figure 8**, SFP values for the four buildings in our main project are presented.

	Specific Fan Power [kW/(m ³ /s)]			
Office building	Lorensberg	Svaneholm	Stampen	Lagerm
Rated data	3.7	3.6	5.2	2.9
Measured	2.2	2.6	4.6	2.8

Figure 8 SFP values for the four Swedish project office buildings before ECMs

A comparison between the figures above and Swedish (as well as American) limits regarding SFP values shows that most SFP values considerably exceed these limits. However, this result was also expected, since the four buildings are not built according to current standards.

Figure 8 also indicates a fact that we have seen in many offices that we have studied. There are often great differences between SFP values based on rated data (motor power and air flow) and SFP values based on measured data.

The above values may be classified as high compared to the levels that we discuss today. However, with the exception of the Stampen value the SFP values for the four project buildings are not extremely high. In Figure 9, SFP values for seventeen Swedish office buildings that we also have studied are shown as a comparison.

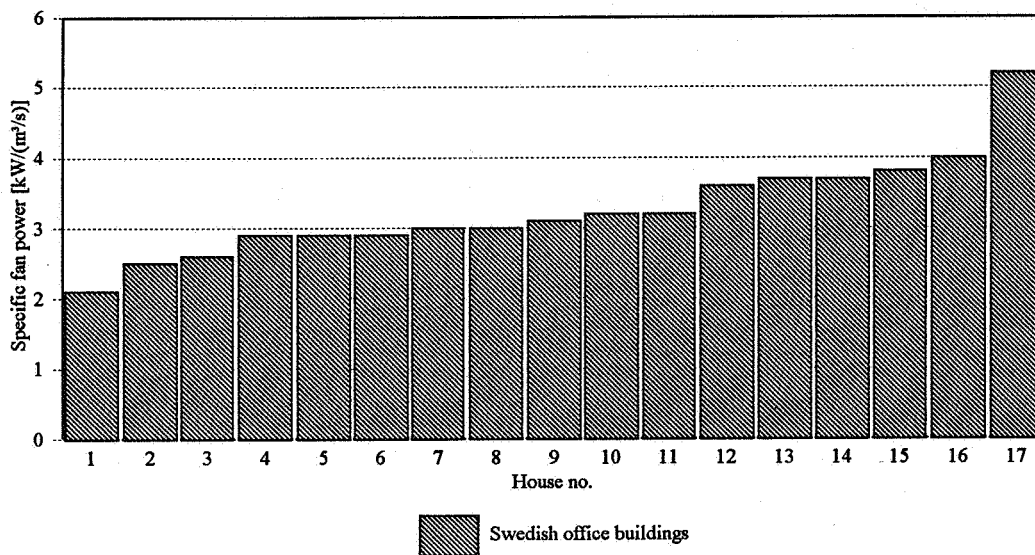


Figure 9 SFP values based on rated data in seventeen Swedish office buildings

5 AN OVERALL VIEW IS IMPORTANT WHEN PLANNING ECMs

By implementing appropriate ECMs the use of electricity and heat energy may be considerably reduced and also lead to a better indoor climate. However, it is important to have an overall view on the building and its systems and take all effects of interacting ECMs and systems into account.

The result of the DOE2 simulation shown in Figure 5 above demonstrates this. When only lighting measures (Measure no 1) were carried out, a great reduction of the use of electricity was obtained - but more energy had to be supplied for heating. However, when also HVAC measures (Measure no 2) were carried out, the heat energy use as well was reduced considerably.

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¹ A summary report based on reference 2 and reference 3 will be published in English in the autumn of 1996

