15th AIVC Conference
The Role of Ventilation

held at Palace Hotel, Buxton, UK
27-30 September 1994

Proceedings Volume 1

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 *):

I Load Energy Determination of Buildings*
II Ekistics and Advanced Community Energy Systems*
III Energy Conservation in Residential Buildings*
IV Glasgow Commercial Building Monitoring*

V Air Infiltration and Ventilation Centre
VI Energy Systems and Design of Communities*
VII Local Government Energy Planning*
VIII Inhabitant Behaviour with Regard to Ventilation*
IX Minimum Ventilation Rates*
X Building HVAC Systems Simulation*
XI Energy Auditing*
XII Windows and Fenestration*
XIII Energy Management in Hospitals*
XIV Condensation*
XV Energy Efficiency in Schools*
XVI BEMS - 1: Energy Management Procedures*
XVII BEMS - 2: Evaluation and Emulation Techniques
XVIII Demand Controlled Ventilating Systems*
XIX Low Slope Roof Systems
XX Air Flow Patterns within Buildings*
XXI Thermal Modelling
XXII Energy Efficient Communities
XXIII Multizone Air Flow Modelling (COMIS)
XXIV Heat Air and Moisture Transfer in Envelopes
XXV Real Time HEVAC Simulation
XXVI Energy Efficient Ventilation of Large Enclosures
XXVII Evaluation and Demonstration of Domestic Ventilation Systems
XXVIII Low Energy Cooling Systems

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 aggreement 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, Switzerland, United Kingdom and the United States of America.

15th AIVC Conference "The Role Of Ventilation"

Contents Volume 1

Contents Volume 2

The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Efficiency of Ventilation in Office Buildings

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SYNOPSIS

Inadequate ventilation is often cited as the cause of unhealthy air quality within office buildings, whilst excessive ventilation is similarly assumed to be the cause of discomfort and energy waste. However, the reality is that very little data is available to assess the significance of these problems on any large scale. The perfuorocarbon tracer (PFT) technique offers the potential for overcoming the problems of applying conventional tracer gas techniques to large or multi-roomed buildings. Methodologies are described for its application to measure ventilation in a selection of different office building types, based on the concept of homogeneous tracer gas emission. Local ventilation rates are measured in a multicell office building, with both mechanical and natural ventilation. These illustrate the distribution of ventilation and highlight implications for air quality and energy-efficiency. A multicell computer model is used to validate the field protocol and to compare predictions with measured results. A simplified PFT measurement system based on standard commercial equipment is described, to encourage wider use of the method.

INTRODUCTION

Inadequate ventilation is often cited as the cause of unhealthy air quality within office buildings, whilst excessive ventilation is similarly assumed to be the cause of discomfort and energy waste. However, the reality is that very little data is available to assess the significance of these problems on any large scale. Part of the reason for this is the lack of any suitable measurement method which can be easily and cheaply applied in large or multi-roomed buildings. Conventional tracer gas techniques suffer from many inherent drawbacks when applied to such buildings, particularly where they are naturally ventilated.

The perfuorocarbon tracer (PFT) technique offers the potential for overcoming these problems. Measurement equipment has been developed independently in the US[1] and Sweden[2], although this has not been taken up widely in Europe mainly due to the specialist nature of the technology. The application to office buildings has not been well developed. To address these issues, we describe work carried out to develop such methodologies and a simplified PFT measurement system, based on standard commercial equipment, to encourage its wider use. Local ventilation rates are measured, which illustrate the distribution of ventilation and highlight implications for air quality and energy-efficiency. The work was carried out under the DOE Construction Sponsorship Directorate's Energy-Related Environmental Issues research programme.

METHODOLOGY

Sandberg[3] presented a theoretical method for measuring the mean age of air in a single room using a uniform distribution of tracer source strength, or homogeneous emission. BRE has been developing a practical application of the method to multi-roomed office buildings[4,5] as follows. Passive tracer gas permeation sources are placed in rooms and corridors throughout the building, with one or two in individual rooms and several along corridors, broadly in proportion to floor area. Diffusion-type air samplers are subsequently placed in a selection of typical rooms, to collect the average concentrations, C, over several days. The local mean age, τ_j, within a room of volume V_j is then calculated from the following:

$$(S / V_j) \,.\, 1/C_j = 1/\tau_j \qquad (1)$$

where S is the total emission rate, and S/V is the emission rate per unit volume, assumed equal in all rooms. We can regard $1/\tau_j$ as a local ventilation rate. The mean age of air for the building overall can be estimated by calculating a volume weighted average for groups of rooms represented by individual results.

The approach can be applied equally to buildings with natural ventilation or forced air systems. The technique can also be applied to large open areas. In this case sources are deployed evenly throughout the area, and air samples are subsequently taken at a representative number of locations or wherever a measurement is required.

MEASUREMENTS IN MULTI-ROOM BUILDING WITH NATURAL VENTILATION

Field trials were carried out using the homogeneous emission technique applied to measure ventilation rates in the three-storey BRE low-energy office building with natural ventilation, as currently operated. We placed sources in individual rooms approximately one week prior to the measurements (Figure 1). Pairs of diffusion samplers in 17 of the 70 or so rooms and the corridors collected time-averaged air samples over a one-week period. After analysis, the average concentrations were used to calculate time-averaged 'local' (ie room) ventilation rates using equation (1).

Results

Room air temperature were continuously recorded at six locations and used to adjust the source rates. Internal doors and windows were 'as used', i.e. a mixture of open and closed, with very few windows slightly open. External doors were normally closed. The wind was approximately 4.5m/s from 240° clockwise from north, and the average external air temperature approximately 7 °C, from a continuous on-site record.

Table 1 shows the local ventilation rates. A striking feature is the fall off in ventilation rates across the table, as you go from ground to second floor, over a range of 0.12 h^{-1} to 0.76 h^{-1} (a factor of six). This may be surprising in such a low rise, cellular office building. Note that these measurements take account of internal air exchanges from other rooms and floors and include only that portion which can be considered as the equivalent of 'fresh' air from outside ('purging' flow rate). The room (purging) flow rates were in some cases, on the second floor, less than half the minimum requirements as recommended by CIBSE[6], ie 5 l/s for single occupancy, with only a slight improvement on the first floor. Ventilation rates on the ground floor all exceeded this minimum requirement.

A previous study of controlled background ventilation for office buildings[7], based on the same building, predicted a range of ventilation rates between approximately 0.05 and 0.5 h^{-1}, and took the value exceeded 50% of the time, 0.12 h^{-1}, to represent acceptable ventilation. Clearly, however, on a significant number of occasions the ventilation rate will be lower than standard requirements. This suggests two issues to consider in drawing conclusions from these results. Firstly, we should bear in mind that occupants can open windows if they are dissatisfied with the indoor air quality. Secondly, we may need to reconsider how to interpret the ventilation guidelines; should they be an absolute minimum, or a time-average minimum in some sense - if so, over what length of time?

MODEL VALIDATION OF FIELD PROTOCOL USING MULTIZONE AIR MOVEMENT MODEL

The likely validity of the field protocol proposed above was assessed using a multi-zone computer model to simulate the bulk air exchanges between rooms and contaminant concentrations in a multi-room office building. The model was used to predict the sensitivity of the measured local ventilation rates to deviations from homogeneity of tracer gas source strength. This addressed a practical problem since, for field work, it would be convenient to use only a single source strength, with single sources placed in most rooms and small (integer) multiples in larger rooms. However, variations in room sizes will inevitably result in variations in source strength from room to room.

Multizone air flow and contaminant transport model

The model used, BREEZE, is commercially available and has been described elsewhere[8]. Briefly, the model takes input data on the distribution of external background leakage, and on connections between rooms and outside, ie openings, and their characteristic pressure and air flow relationships. External wind pressures are taken in the form of wind pressure coefficients and wind speed at a reference height, and stack pressure in the form of internal and external air temperatures. The model solves for the internal pressure consistent with air flow mass balance between the cells (rooms). Standard algorithms also estimate two-way air exchanges through large openings due to buoyancy and turbulence. In addition, contaminant source rates can be specified to calculate contaminant movement within the building.

Input data

The building modelled was the BRE Low Energy Office building at Garston. Wind pressure data were taken from previous physical model scale measurements[7] in a boundary layer wind tunnel. Calculations were carried out with different source strength distributions, and with internal doors open and closed in different combinations. All external doors and windows were closed. The background leakage for this building has previously been measured as 'tight' for UK office buildings[7,9]. In all cases the wind speed and direction were set at 5m/s from 240° clockwise from north, and air temperatures 18.5°C internally and 7.3 °C externally. These correspond approximately to the average conditions of the full scale measurements described above. The interzone air exchanges and room (cell) concentrations were determined and, together with the local source strength, used to manually calculate local ventilation rates (from local mean ages of air) using equation (1).

Results

The reference condition was taken to be the case with rooms containing single or integral multiples of sources, 'unity sources', with all internal doors open. The results for all rooms in the centre section of the ground floor are shown in Table 2. A series of further cases were modelled in which the source strength per unit volume was set equal, or 'normalised', to that in an average sized room (cell 15) for increasing numbers of cells in the centre section. No significant change in the local mean age was observed. Similarly, no change was observed when the source strength was normalised in adjoining sections of building on the same floor level. Two further cases were modelled in which five of the doors to rooms in the centre section were closed, but allowing for a small crack area 1.5 cm x 75 cm, and then with perfectly sealed doors. Again, no significant difference in local rates was observed between unitary sources and uniformly distributed source strength although, as expected, the room ventilation rates themselves were significantly altered in each case.

COMPARISON OF FIELD MEASUREMENTS WITH MODEL PREDICTIONS

The multicell model of the test building, described above, was used to compute mean ages from time averaged tracer concentrations over six days, to compare with the measured data. This was done by dividing up the six days into ten distinct sets according to wind speed. The average wind speed and external temperature were then determined for the duration of each of these sets. The wind direction remained approximately constant over the whole test. The model was then run for each of the ten 'weather' conditions, resulting in ten sets of local mean age for specified rooms, chosen to correspond to the field measurements. Time-weighted average values were calculated to compare with the measured results. Area weighted averages of these were computed to estimate of the mean age for each storey, and for the whole building, in the same way as for the field results.

Predicted and measured results are compared for each storey and the whole building in Table 3. The measured value of whole building ventilation rate (reciprocal of mean age), 0.26 h^{-1} compares reasonably well with the predicted value of 0.19 h^{-1}, based on the limited selection of rooms. The latter can be compared with the time-weighted nominal air change rate (inverse of nominal time constant) for the whole building, 0.16 h^{-1}, based on the predicted total inflow to all rooms. The values for individual stories do not compare so well. The reasons for this may be difficult to identify, with possibilities including differences in the model data and the real building regarding leakage between floors, pressure coefficients, internal door opening, and external door and window opening.

MEASUREMENTS IN A MULTI-ROOM OFFICE BUILDING WITH MECHANICAL VENTILATION

The above building was also equipped with a mechanical ventilation system supplying full fresh air, incorporating a cross-flow heat exchanger. With this operating, the local ventilation rates were measured in six rooms and the three corridors, using pumps to take air samples over a 30-minute period in each room.

Results

The internal doors and windows were in general as for the previous test. The wind was 4.5 m/s and the external air temperature was 8.5 °C. Table 4 shows that the ventilation rates were, on average, a factor of about two greater than for natural ventilation. Although the absolute values are greater, from 0.5 to 2.8 h^{-1}, they vary over a factor of six, coincidentally the same as with natural ventilation. However, as expected, the room rates do not vary with floor level. Local flow rates are just over one to under three times the CIBSE recommended value of 8 l/s per person. In this building, it is important that the installed cross-flow heat exchanger should be effective in offsetting some of the potential ventilation heat-losses which would result from these high supply rates.

By placing PFT sources in the supply duct and measuring the concentration downstream at the terminals in several rooms, the measurement technique was also successfully applied to measure the overall supply rate of 1.31 m^3/s and, including infiltration, 1.53 m^3/s, These results compared well with similar reference measurements of 1.26 and 1.45 m^3/s respectively, carried out using SF6 tracer gas. The build-up of SF6 tracer was also continuously monitored in the exhaust duct, and used to calculate the overall mean age $<\tau>$ and ventilation efficiency $<\varepsilon_a>$ of the ventilation system, using the following expressions[10,11]:

$$\langle\overline{\tau}\rangle = \frac{\int_0^\infty t.\,(1-\frac{c(t)}{C_e^\infty})\,dt}{\int_0^\infty (1-\frac{c(t)}{C_e^\infty})\,dt} = \frac{\mu^1}{\mu^0} \qquad (2)$$

$$\langle\epsilon_a\rangle = \frac{\tau_n}{2.\langle\overline{\tau}\rangle} = \frac{\tau_n}{2}.\frac{\mu^1}{\mu^0} \qquad (3)$$

where $C(t)$ is the instantaneous concentration in the exhaust duct, and C_e^∞ is the final steady-state concentration. The numerator is defined as the 1st moment (μ^1) and the denominator the 0th moment (μ^0)about the origin[12]. τ_n is the nominal time constant, defined as the building volume divided by the ventilation supply rate.

This gave a ventilation efficiency of 54%, which is close to the theoretical value for a well-mixed system (50%). This indicates that the air supply is well distributed throughout the building, with no significant short-cuts. In this case, if we assume occupancy to be evenly spread, the required local ventilation supply rate per unit volume (ie local ventilation rate) is the same throughout the building, and equal to the overall building ventilation rate. Table 4 compares the local rates (h^{-1}) to the overall ventilation exhaust rate (1.2 h^{-1} assuming an internal volume of about 4500 m^3), as an indicator of efficiency. Values greater than 50% (perfect mixing) in many rooms indicate that more air is supplied than needed for uniform occupancy.

SIMPLIFIED 'PASSIVE TRACER' GAS ANALYSIS EQUIPMENT

For a few years now, technology has been available to measure ventilation rates using perfluorocarbon tracer gases (PFTs) emitted from small permeation sources, and collected using diffusion samplers. However, this has only been taken up in a few countries in Europe, in part due to the 'non-standard' nature and relative complexity of the laboratory-based analysis equipment. To overcome this problem BRE has collaborated with a manufacturer of gas chromatographic analysis equipment (Perkin Elmer (UK) Ltd), with guidance from Jan Krisstenssen of the University of Stockholm, to develop a simplified technique, based on commercially available equipment. This is based on the Perkin Elmer ATD50 thermal desorber and compatible gas chromatograph (gc) with ECD detector. Tracer separation on the gc is achieved with a silica PLOT column (25 m) at 120°C. Tracer gas samples are trapped in the samplers on an adsorbent bed of Carbopack B and desorbed at 200°C.

Validation tests
The passive tracer system has been used to measure the ventilation rates in both a test chamber and a naturally ventilated office room. In the case of the chamber, results compared well with reference measurements of the constant supply flow rate of 60 litres/sec. In the room reference measurement were made by monitoring the concentration of SF_6 tracer gas continuously introduced at a constant, metered, flow rate, in parallel with measurements made using the SBI passive tracer gas samplers and sources (PMCP - peflouromethylcyclopentane). Good agreement was found between the SBI and BRE analyses, for both pumped and passive sampling, but both over estimated the ventilation rate by 25% compared to the SF6 measurement. Further validation and inter-laboratory tests are planned to investigate this.

Field measurements
The equipment was used for measurements in five buildings, in parallel with work carried out as part of an EC collaborative programme on a 'European Audit Project to Optimise Air Quality and Energy Consumption In Office Buildings' [13]. All five buildings had forced air supply systems. In these situations it was considered necessary to use pumps to take air samples for a short period during the afternoon, to allow sufficient time for steady state tracer concentrations to be achieved following system start-up in the morning. Measurements in one of these buildings have been described in a previous paper[5]. The areas studied were all open plan, and in four of the buildings these received recirculated air from other parts of the building not tagged with tracer. This made it difficult to determine either the fresh-air supply or infiltration rates without additional measurements. Although such additional measurements were carried out, the analyses are rather complex and inappropriate for inclusion here. In the fifth building, two independent air supply systems supplied air only to the test area, but these switched on and off alternately at an interval of about 15 minutes, which may be expected to have affected the ability of the PFT tracer to reach steady state.

However, in four cases, results showed a relatively even distribution of tracer and, consequently, local ventilation rate. An uneven distribution was measured in the fifth building, part of which was confirmed to operate with full recirculation. This application to large open areas would benefit from some form of validation, particularly on source placement. A sensitivity analysis could usefully be carried out using computational fluid dynamics (CFD).

CONCLUSIONS
The perfuorocarbon tracer (PFT) technique offers the potential for overcoming the problems of applying conventional tracer gas techniques to large or multi-roomed buildings. Methodologies were described for its application to measure ventilation in a selection of different office building types, based on the concept of homogeneous tracer gas emission.

Field trials were carried out using the homogeneous emission technique applied to measure ventilation rates in a three-storey, naturally ventilated office building. The local ventilation rates in some rooms were less than half the minimum recommended rate of 5 l/s for single occupancy. It was noted that occupants can open windows if they are dissatisfied with the indoor air quality. However, since the occupants are satisfied, this suggests the possible need to reconsider how to interpret the ventilation guidelines; should they be an absolute minimum, or a time-average minimum over some specified time period?

A multi-cell prediction model was used to validate the field procedure, which indicated that local mean ages calculated from tracer concentrations were not significantly insensitive to uniformity of tracer source placement (homogeneity). The model was also used to compute ventilation rates from tracer concentrations in a limited selection of rooms, for conditions corresponding to field measurements over a six day period. The measured whole building ventilation rate (reciprocal of mean age), 0.26 h^{-1}, compared reasonably well with the predicted value of 0.19 h^{-1}.

The technique was also applied to measure local ventilation rates in a multi-room office building with mechanical ventilation. Room ventilation rates were found to be just over one to under three times the recommended value of 8 l/s per person, which emphasises the need for the installed cross-flow heat exchanger to be effective in offsetting some of the potential ventilation heat-losses which would result from these high supply rates. The measurement technique was also successfully applied to measure the overall supply rates. A separate tracer gas showed the ventilation efficiency to be close to the theoretical value for a well-mixed system.

To underpin the wider use of the method, a simplified PFT technique, based on commercially available equipment was also described, along with some limited validation work. This was applied in five buildings, with forced air supply systems and open plan layouts. As expected, results generally showed a relatively even distribution of tracer, and consequently local ventilation rate. An exception to this was identified, which corresponded to a building with a malfunctioning supply system.

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Location:	021	022	008	001/2	007	C0	122	123	113	112	105	106	C1	224	225	213	212	204	205	C2
Vol m3	49	52	49	300	54	111	40	48	42	47	65	63	112	41	166	48	53	77	65	133
Up (l/s)	10.3	5.2	9.5	23.3	8.7	12.0	3.0	2.1	2.8	2.0	3.1	2.8	9.0	1.8	6.5	3.1	2.5	2.6	2.2	10.0
τ (h)	1.3	2.8	1.4	3.6	1.7	2.6	3.7	6.3	4.2	6.7	5.9	6.3	3.4	6.3	7.1	4.3	5.9	8.3	8.3	3.7
$1/\tau$ (1/h)	0.76	0.36	0.70	0.28	0.58	0.39	0.27	0.16	0.24	0.15	0.17	0.16	0.29	0.16	0.14	0.23	0.17	0.12	0.12	0.27

Key: Up = room purging flow rate; τ = local mean age; $1/\tau$ = local ventilation rate; Cn = corridor level n

Table 1 . Ventilation rates in the naturally ventilated office building

Cell number	Room number	Volume V (m^3)	Conc (pl/l)	S/V [1] (nl/h)/m^3	Infiltration (l/s)	age τ (h)	rate $1/\tau$ (h^{-1})
10	4	70	199	52	317	3.85	0.26
11	corridor	96	165	38	0	4.39	0.23
13	6	47	279	39	117	7.23	0.14
14	5	71	155	51	417	3.03	0.33
15	8	47	201	39	167	5.21	0.19
16	7	55	90	33	367	2.72	0.37
17	10	47	896	39	0	23.03	0.04
18	9	39	115	46	283	2.48	0.40
19	11	47	169	39	183	4.39	0.23
20	18	68	3,423	27	0	128.2	0.01

[1] S/V = 39 nl/h per m^3 for uniform homogeneous emission

Table 2. BREEZE prediction for centre section of ground floor, with 'unity' sources.

	$r = 1 / \tau_{avg}$ (h^{-1})	
	Measured	**Predicted**
Second floor	0.20 (0.05)	0.22
First floor	0.22 (0.05)	0.13
Ground floor	0.47 (0.2)	0.28
	( ) = approx std dev.	
Whole building	0.26 (0.05)	0.19

Table 3. Predicted and measured ventilation rates calculated from concentrations in selected rooms

Location:	001	008	C0	113	112	122	C1	213	C2	inlet	exhaust
Vol m3	300	40	110	40	45	27	112	40	133	(4500)	
Up (l/s)	51.7	20.2	19.6	10.4	11.3	21.0	15.6	17.0	24.8	1309	1480
τ (h)	1.6	0.5	1.6	1.1	1.1	0.4	2.0	0.7	1.5	1.0	1.2
1/τ (1/h)	0.62	1.82	0.64	0.94	0.90	2.80	0.50	1.53	0.67	0.95	0.84
Effic (%)	26	77	27	40	38	118	21	65	28		
Key: as Table 1; Effic = 100 * τ (exhaust) / 2.τ (room)											

Table 4. Measurements with mechanical ventilation

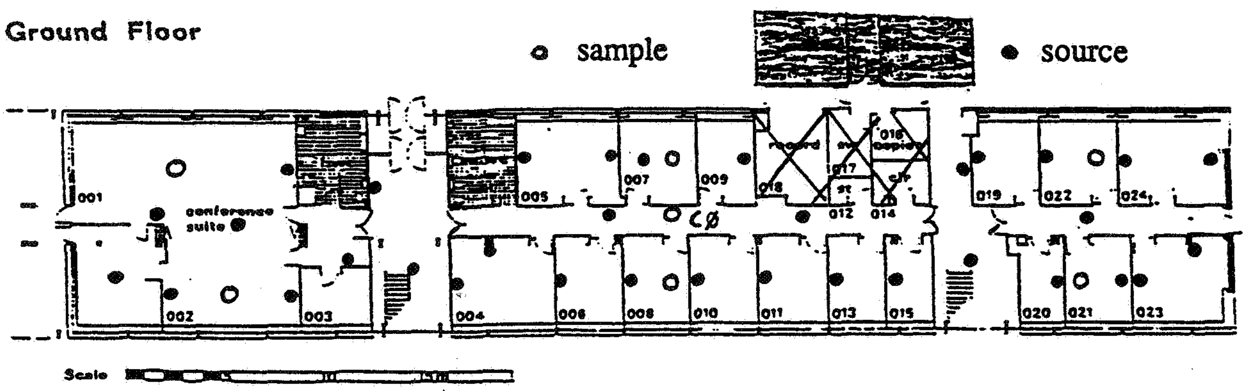

Figure 1. Multi-roomed office building; typical source and sample locations

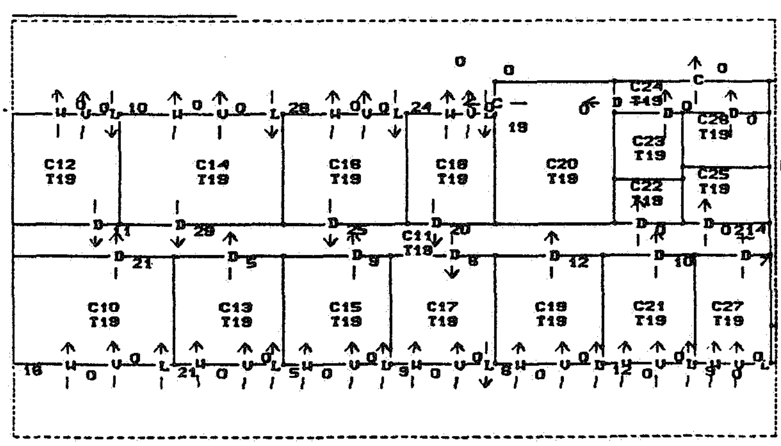

Figure 2. BREEZE coding of centre section of ground floor, showing air flows (m^3/h) through fabric leakage (L). No flow through windows (W) and vents (V).

The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Annex 27 - Domestic Ventilation , Occupant Habits' Influence on Ventilation Need

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Annex 27 - Domestic Ventilation.
Occupant Habits´ Influence on Ventilation Need.

Synopsis

The Annex 27 (A27), Evaluation and Demonstration of Domestic Ventilation Systems, is given a genaral introduction. The habits varys a lot between individuals, the dwellings are of various sizes with various numbers of occupants being at home for longer or shorter times. Those facts needed to be collected in the beginning of the annex.

In this paper background data will be given to make it possible to discuss the varied need for outdoor air supply in dwellings. Data for the parameters have been collected from many sources. Often the original purposes of the studies were quite another than discussing the required outdoor air supply. Sometimes the sources can be found outside the research field of the built environment.

Statistical data will be given on dwellings such as size, number, construction year, persons/dwelling, and also concerning the use of the dwelling and the occupants´ habits. The variation can be over the duration of the building. For the short time variation we can find some sources on time spent in dwellings, shower habits, water consumption, cooking time, use of appliances in the dwelling, number of pot plants, window airing and smoking habits.

1 Introduction

This paper consists of two parts. In the first part is briefly given a presentation of the newly started Annex 27 (A 27), Evaluation and Demonstration of Domestic Ventilation Systems. The main aim is to give the frame of the work and the direction of how to proceed. During forthcoming conferences results from this annex will be presented.

The second part gives data from various sources on occupants´ habits. Most of those habits are influencing the needed ventilation rate. As there is a great variation between individuals, avail-able data has been collected to give both average and extreme values. In order to establish real cases for the simulations, data has been collected from as many countries as possible, also outside the participating countries. When comparing the data from the individual countries some common trends can be found and also that some countries have similarities.

2 Annex 27 - Domestic Ventilation Systems

The new annex, A 27, started in April 1993 as a result of discussions in conjunction with AIVC-meetings. The work planned will use the results of several of the finished and ongoing annexes and put the parts together in order to find tools for practitioner to evaluate domenstic ventilation systems also at the desk facilitating a better jugement of the expected outcome of a selected system. The official participants in the A 27 are Canada, France, Japan, The Netherlands, Sweden, UK, and USA. As observers have acted Belgium, Finland, Norway, and Switzerland. Those countries have a total number of 225 million dwellings with a useful floor space of about $27\,000 \bullet 10^6$ m^2.

2.1 Background

The rate of outdoor air supply as well as comfort aspects associated with air distribution and the ability of the systems to remove pollutants are important factors to be considered at all stages in the building lifecycle. As distinct from a work place, occupants in dwellings can vary across a wide span from an allergic infant to a well trained sportsman, from active outgoing people to elderly confined to a life indoors.

During the lifetime of a building its dwelling occupational pattern vary. This results in a varying need for supply air to obtain acceptable indoor air climate and to avoid degradation of the fabric. Emissions from building materials are also time dependent. When the building is new or recently refurbished it may be necessary to dilute the emissions by extra supply air. In stand-ards and codes the supply air needed in a dwelling is generally based on the maximum number of persons living in the dwelling, defined by the possible number of beds contained therein.

Dwellings represent about 25 - 30 % of all energy used in the OECD countries. In the near future domestic ventilation will represent 10 % of the total energy use. Thus even relatively small reductions in overall ventilation levels could represent significant savings in total energy use. Improvement of residential ventilation is of concern in both existing and future buildings. The functioning of the ventilation system may deteriorate at all stages of the building process and during the lifetime of the building. Research in the recent years and in particular the IEA annexes now makes it possible to formulate methods to evaluate domestic ventilation systems.

2.2 Objectives

The objectives of A 27 are: **to** develop tools, for evaluating domestic ventialtion systems; **to** validate the methods and tools with data obtained from measurements; **to** demonstrate and evaluate ventilation systems for different climates, building types, and use of the dwellings

The methods, tools, and systems are intended for existing and future residential buildings that require heating. The target group is composed of standard and policy makers, developers in industry, and ventilation system designers.

2.3 Scope of the annex

The annex is divided into three subtasks described briefly in table 1. Most of the work in Subtask 1 is done and the main efforts are now spent on Subtask 2.

Table 1 Scope of the annex

Subtask 1 ***State of the Art***	**Subtask 2** ***Development and Validation of Evaluation Methods***	**Subtask 3** ***Evaluation, Demo, and Application of Current and Innovative Vent Systems***
1. Give an overview of typical and frequently used systems **2.** Background and reasons for exist-ing systems and standards **3.** Review exist. evaluation method Report Nov 1994	**1.** Define evaluation parameters **2.** Select methods **3.** Develop tools **4.** Validate methods and tools Report mid 1996	**1.** Use the methods and tools developed for a set of variables (climates, building types, users, constructions, new, renovated, and existing buildings **2.** Demo good performance of priniply different ventilation systems **3.** Demo innovative systems Report mid 1998

3 Occupants´ habits

The main objective bringing together detailed information about the occupants´ pattern in dwellings is to identify most of the cases, say 90 %, that have influence on the needed outdoor air supply. When these boundary conditions are identified, realistic cases can be set up and assumptions made for occupants´ habits so that simulations can be made. In the first section is shown statistical data on housing in as many OECD countries as possible and in the second users´ habits. The facts has been collected both from technical reports but also from quite another studies such as market surveys. Some studies are only made in one country and can be said to be week but on the other hand indicates a pattern that is general. The study also aimed at giving evidences to the starting point that the needed outdoor air rate varys over the time during the lifecycle of a dwelling illustrated in figure 1.

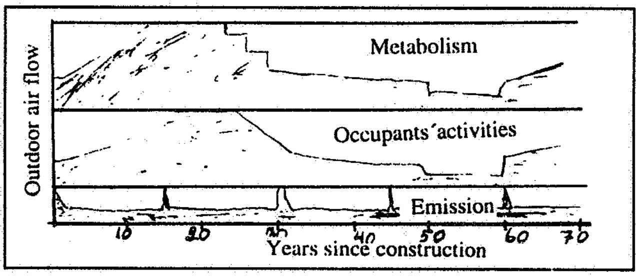

Figure 1. Long time ventilation need

3.1 Data on housing

With data on housing it makes it possible to see that many dwellings are not that densed populated that the codes and standards have assumed when making the rules. When studying severel countries´ data it makes it possible to give an estimation of the trends that might be possible e.g. the growing number of elderly people gives more households with only one person. The data given here are the useful dwelling space alone, as it is the more proper figure for discussing needed ventilation.

The 14 OECD countries, see table 1, with a population of about 700 million, have approximatly 280 million dwellings with a useful floor space of 32 000 million m^2. However, there is a great variation of the useful floor space from country to country. The largest dwellings are in North America (134 m^2 - 152 m^2) whilst the smaller are in Japan (89 m^2)and Europe (65 m^2 - 110 m^2). When looking on the construction years we can get an opinion of the future need for new dwellings or if it needs to be refurbished. From data four different groups can be identified, see table 1. Another factor to be considered is if the dwellings are in blocks of flats or single family houses. Studies have indicated that people living in flats tend to make more complaints than in single family houses. One interpretation given is, that people in flats have a feeling they can not manage the situation, and here in particular the ventilation system.

Table 1 Construction years of dwellings. Ref. 1, 2, 21, 22, 23

Before -1945 40 - 50 % constructed	1946 - 1970 ≈ 50 % constructed	1971 - and later ≈ 50 % constructed	Even distributed with 1/3 in each period
Belgium, Denmark, France, UK	Germany, Italy, Sweden	Canada (?), Finland, Japan, Netherlands, USA	Norway, Switerland,

The distribution between flats and single family houses varys greatly. UK has 83 % of the dwellings in single family houses whilst the opposite is Switzerland with only 31 %. Japan and North America have about 65 % whilst most of the European countries have figures around 50 %, except Belgium, Denmark, and The Netherlands with figures closer to that of North America. Ref. 1, 3, 10, 11, 13, 22, 23, 27.

3.2 Dwelling usage

In this section will be given figures on how the dwelling is used both the population density and the variation of activities and time spent in the dwelling. As can be seen in table 2 the number of persons/dwelling (p/d) vary from 3.2 p/d to 2.1 p/d and most countries around 2.5 p/d. The area/person can vary from 27 m^2 to 61 m^2 giving a much larger volume for persons in some of the countries.

If the ventilation rate is going to be more adapted to the individual demand and the activity at hand the ***number of persons/household*** is of interest. If this is coupled to the size of the dwelling we can get an opportunity to give the boundaries for the needed ventilation rates. As can be seen in table 2 there is only 2 persons in nearly 60 % of the dwellings with the exceptions of Japan and Italy. The trend is that with a growing number of an elderly population the fewer persons/household. Families with 5 persons or more are not very frequent, less than 10 % of the households, with the extremes, 5 % of the households, in Denmark, Germany, and Sweden. In a study in Sweden it was found that only 1/4 of the households had children.

Table 2 Number of persons/household, % of all dwellings Ref. 1, 2, 3, 13, 21, 23, 24.

Country	Persons/ dwelling	Area m^2/person	Number of persons/household (distribution %)				
			1	2	3	4	5 -
Belgium	2.7		26	30	18	16	9
Denmark	2.2	49	34	33	15	13	5
Finland	2.5	30	31	29	17	15	8
France	2.7	32	25	28	19	16	12
Germany	2.5	35	33	29	18	14	6
Italy	3.2	29	22	24	23	21	10
Japan	3.2	28	18	20	18	23	21
Netherlands	2.6		27	30	15	19	9
Norway	2.5	43	35	26	15	15	8
Sweden	2.1	47	40	31	12	12	5
Switzerland	2.6	34					
United Kingdom	2.7	27	24	33	17	17	9
Canada	2.8		21	30	18	19	12
U.S.A.	2.3	61	25	33	16	17	9

From table 2 can be seen that ***1-person-household*** is more frequent in Europe, about 1/3 of all households. In Japan and North America about 1/5 of the households consist of one person. The trend in Europe since 1950 has been an increase from about 1/5. In USA the trend was the opposite. 1950 was 1/3 a 1-person-household and 10 years later 13 %. The general trend has been that the number of one-person-household doubled during 40 years, but in Japan the increase was 4 times. Ref. 1, 14, 24.

The number of persons/household can also be compared with ***persons/bedroom*** giving an estimation of the size of the dwelling. This value indicates if outdoor air is "needed". As a general trend most of the households with 5 or more persons have more bedrooms because the frequency of more than 2 persons/household is very low except in Finland, France%, and UK. Those countries have the same frequency as for 5 person/household. The frequency of less than 1 person/bedroom varys from less than 20 % in Finland, France, and The Netherlands; 30 - 35 % for Germany, Sweden, Switzerland, UK, and Belgium (40 %); finally about 50 % Denmark, Japan, Norway, Canada, and USA. Ref. 1, 23.

It is well know by planners that the number of children is on its maximum about 10 years after completion of a housing development. When the children have grown up and moved the number of persons living in each dwelling becomes fewer. This situation is illustrated in figure 2 showing that in single family houses constructed after 1971 there are 3.5 persons/house, while in the older houses there are only 2.5 persons/house. The long time perspective of the needed outdoor air flow rate can also be seen when studying the moving frequency, see figure 3. Highest frequency has young people moving away from their parents. Then they settle a family, it grows and a larger dwelling is needed. At the age of 35 the family is stable and the final size of the dwelling has been reached. The interest to move away from the dwelling that a family had at the age of 45 is very small.

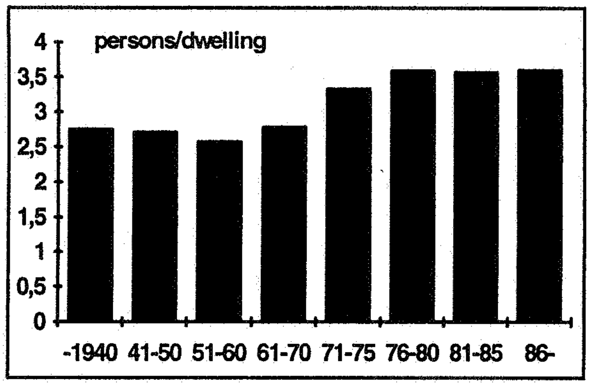

Figure 2 Number of persons living in single family houses, various construction years, Ref 18

The ***time spent in a dwelling*** varys within a great range. Elderly people are spending more time at home than young adults. About 30 % of employed men have their lunch at home. For women about 60 % are at home most of the daytime, except singles. The time spent in the kitchen varys both with the area of the kitchen and with cultural habits. French women spend twice the time in the kitchen compared to women in USA and it does not matter if the woman is a housewife or employed. Housewifes are working doubled time in the kitchen compared to employed women. Housewifes are working in the kitchen from 1.6 h to nearly 3 h and men about 5 min each day. Ref. 5, 26.

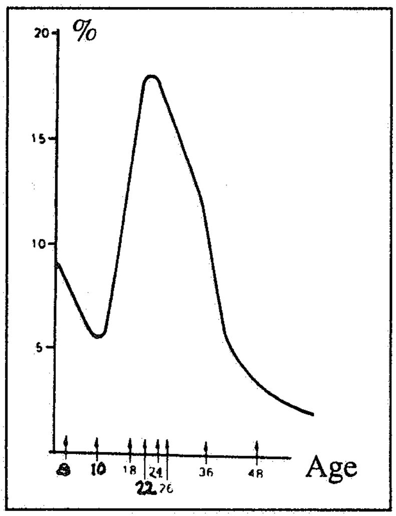

Figure 3 Moving frequency for different ages, in Sweden. Ref. 18.

There is a large span in the average ***indoor temperature*** between countries. Also between single family houses and flats there is a difference, higher in flats, ref 29. The deviation from the average temperature can be large and in some surveys it is found to be ±7 °C, ref 5, but more common is ±3. This might be influenced by the dressing habits and the long tradition of various indoor temperature in various rooms. However, having a great span in the temperature within the same dwelling causes discomfort.

In dwellings, where heating is necessary, the ***water vapour generation indoors*** is the most severe risk for the building because of the risk for condensation causing mould and house dust mites growing. How much water vapour, that is generated, depends on the number persons, their activities, and the use of water in the dwelling. The humidity can also be increased by the cooking habits.The seasonal variation of outdoor water vapour content increases the needed outdoor air for removal of the indoor generated water vapour.

The ***equipment*** in the dwellings indicates the possibility to produce moisture and other pollutants. In very close to 100 % of the dwellings there is a bath/shower. Another moisture source is the washing machine. About 85 % of the dwellings have such a machine. There are some doubts about this figure, which might indicate "access to a washing machine". Dishwasher is only used in 1/4 of the households except in USA, 60%. The percentage of central heated dwellings indicates the possible use of single room heaters, which can give pollutants directly to the room. In the Nordic countries and USA nearly 100 % are central heated, in Switzerland 50 %, and nearly no central heated dwellings at all in Japan, 3 %. Ref. 1, 2, 3, 4, 14, 23, 25.

The ***water consumption*** varys within a large range but average figures from Japan, The Netherlands, and Sweden are from 130 l/(person, day) to 190 l/p,d. The most interesting is the use of water for showering/bath and figures found goes from 45 l/p,d to 60 l/p,d. The water is used most frequently in the morning, 7 - 10 h, and in the evening 19 - 22 h.

When ***washing the clothes and drying*** it the water vapour content will be increased indoors. The most common way to dry the clothes is still to do it in the air. A study of 12 household (4 persons/household) measured the frequency of 4 times/week. A questionnaire to 1000 persons showed that more than 60 % washed 1 - 6 times each week. Ref. 7, 16, 17.

Most of the water vapour produced in a dwelling, besides from the metabolism, originates from ***body washing*** and very much depending on the habit to take a shower or a bath.. The variation in use of hot tap water can be in the ratio 1:20. A shower, today most used, gives a water vapour production of about 2600 g/h, ref 8. Another study, ref 15, shows that even with an exhaust fan, 100 % relative humidity will be reached after only 5 min of showering. In the above mentioned questionnaire 1000 persons, ref 16, 17, were asked there body-washing habits. The results shown in figure 4 and 5 can be summarized in the following points:

1. About 85 % of all are showering. Retired persons 70 %.
2. Men and women have the same showering habits.
3. Once a day 65 % are showering.
4. Twice a day 10 % are showering.
5. Nearly 20 % of the men aged 16 - 29 are showering twice a day and 10 % of the women.
6. Women takes shorter showers than men and is recognized especially for the busy period of the life at the age of 30 - 49 with work and children .

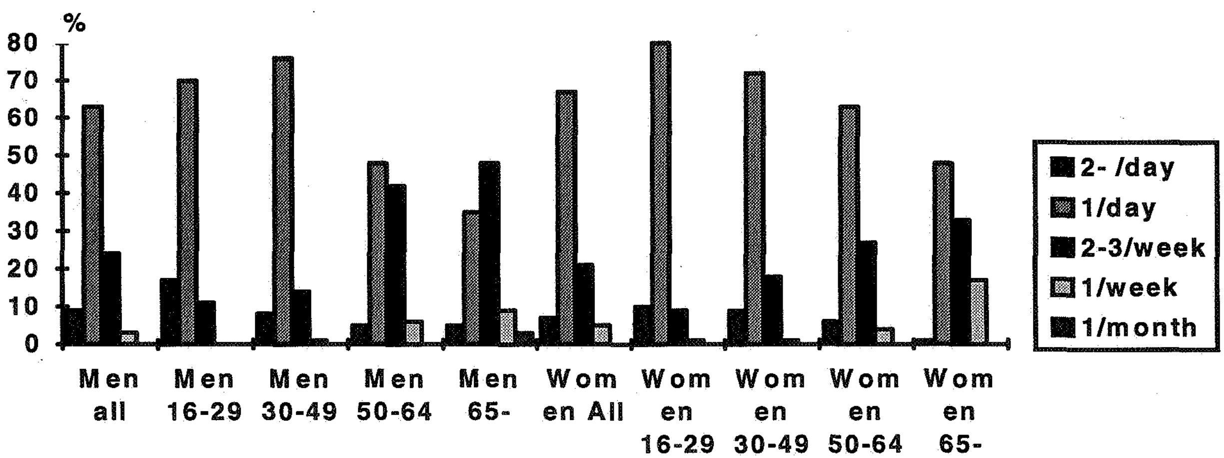

Figure 4 Showering frequency. Ref 16, 17.

Figure 5 Duration of showering. Ref 16, 17.

The number of ***pot plants*** is very depending on national habits. A market research, Ref 10, gave that Germany, The Netherlands, Switzerland, and Sweden are the countries in Europe with most pot plants in their dwellings. The result gives that pot plants are placed at the whole length of all the windows. A calculation applied on Sweden gives 0.25 -0.30 pot plants per m^2 dwelling area and need the air change of about 0.1 h^{-1} to exhaust the water vapour.

Even if there is some uncertanty about the calculation of air flow through large openings a much bigger uncertanty will be the time when the windows are opened and the width of the opening. Other parameters are the constructions of the window, indoor temperature, and wind speed. Some conclusions found about ***window opening habits*** are:

* The same daytime and at night
* Proportional to the outdoor temperature
* Proportional to the inverted value of the wind speed
* Windows are not left opened when no person is present in the dwelling.
* Doubled when tobacco smoking is allowed.
* Regulate the temperature
* Depending on housewives´ habits when making up beds and cleaning the dwelling.
* Less when higher indoor temperature is prefered.
* Less amongst elderly people.
* No socio-economic correlation
* Increased when the room has direct solar radiation
* More when sunny outdoors

Actually it was found in ref 19, that the number of windows multiplied with the temperature difference was constant $n \bullet \Delta T = 2.2 \pm 0.4$ (n=fraction of windows opened; ΔT= temperature difference) to be used at a temperature difference >7 ^{o}C. However in a study on 85 single family houses was found $n \bullet \Delta T = 0.3 \pm 0.1$. The airing with fully opened windows were reported to be 5 - 10 min. (8 min according to questionnaires, 11 min according to interviews), ref 20.

Smoking habits varys both between countries and within a country. Today about 30 % of the inhabitants are smokers and the smoking habit is nearly the same for men and women, with the exception of Japan where men are more and women less frequent smokers. Especially women that are smoking during pregnacy and staying at home with their infants can cause oversenitivity amongst their children. Adults´ own smoking habits will also cause oversensitivity that will lead to a demand for more outdoor air supply. Ref 28.

Other indoor pollutants that is depending on the occupants´ habits are ***nitrous gases*** from gas fired appliances. Here the building regulation goes towards a standard for direct exhaust by a chimney. However, for stoves and ovens it might be difficulties to exhaust all especially from the pilot flame as the cooker hood don´t work when there is no cooking going on.

Periodically ***volatile organic compounds (voc)*** will occur indoors. One is coupled to the cleaning habits and gives emissions weekly with some extra voc:s a few times yearly. The other type is voc from furniture, surface covering, and building fabric emitted once the building is constructed and then periodically when redecorating with an interval of 10 - 15 years, and major refurbishments after 30 - 40 years. Both types requires an increased outdoor air flow. In addition to the periodically emission types a daily emission occurs giving a base ventilation need by the use of deodorants (used by 50 % of men and 65 % of women), hair-spray and also daily use of detergents, soap etc and remaining emissions from furniture and building fabric.

To get an indoor climate that satisfy the individual need, also ***body odour*** has to be removed. The traditional tracer gas is CO_2 that is totaly harmless at levels in dwellings (<5000 ppm). The adaptation to the body odour is very quick and is nearly only sensed by people entering a room. The level is also a matter of how the dwelling is used. If the door to the bedroom is closed the level is more rapid increased. Most people have the bedroom door opened.

4. Conclusions

All the various habits may result in a demand for a variable outdoor air flow rate. Together with the great differences in dwelling size and number of persons living in the dwellings, it leads to a significant difference in the needed outdoor air flow rate if the rate/person is supposed to be the same for all.

The floor area varys from 65 m^2 to 152 m^2 giving a volume of 150 -365 m^3 with a ceiling height of 2.4 m. With the today´s outdoor air change rate standard of 0.5 h^{-1} the flow will be 75 - 180 m^3/h. If combining the volume and the number of persons/dwelling the outdoor air flow goes from 6.5 l/s,p to 24 l/s,p if all dwellings had mechanical ventilation adjusted to the standard. A base ventilation is also needed of at least 0.1 h^{-1} for the exhaust of daily voc and from pot plants. For weekly or monthly emissions window airing is one way to solve it.

The WHO European guidelines for indoor air quality require the dissatisfaction within a range from 10 % to 30 %. In a workshop at the Indoor Air conference 1993 it was recommended to lower the level to a nuisance threshold level placed at a dissatisfaction of 5 % of the occupants not more than 2 % of the time. The technical way to fulfil such a requirement (wish, recommendation) is to install a demand controlled ventilation system.

All these variables and habits led us in A 27 to make a set of assumptions when doing the simulations. Instead of trying to find an average pattern, which might have lead to a family that does not exist we made up cases to be studied. The variables for users´ behaviour are then used for making simulations. The assumptions that are needed and the parameters to be developed for tools are shown in table 4 to be simulated for different climates. In addition there are measurements planned on cases to be used in the annex for validation.

Table 4. Activities in A 27- Domestic Ventilation		
Design assumptions	**Occupants´ behaviour**	**Parameters (responsible country)**
• Example dwellings, 10 types with 1 - 5 rooms • Ventilation systems, 7 types	• Standard families, 6 cases • Combination families and ex. dwellings; crowded, average, and spacious • Time at home and in individual rooms • Window airing pattern • Internal doors, temperature, metabolism	• Air quality (NL) • Thermal Comfort (J) • Energy (USA) • Noise (NL) • Life Cycle Cost (UK) • Reliability (S) • Moisture (F)

All the variations in habits will result in a range of air flow rates for different situations. This paper gives the background for the hughe variation of the occupants´pattern in dwellings that is the major difference from workplaces giving a very hard task to solve. The detailed simulations will give the consequences for parameters concerning indoor air quality, energy, and life cycle cost at the selection of a certain ventilation system with certain occupants´ habits.

6 Acknowledgements

The Swedish Council for Building Research that is funding the work, the official participants in the annex 27, and the IEA exco for Energy Conservation in Buildings and Community Systems that accepted the annex.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Case Studies of Passive Stack Ventilation Systems in Occupied Dwellings

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CASE STUDIES OF PASSIVE STACK VENTILATION SYSTEMS IN OCCUPIED DWELLINGS.

by Lynn M Parkins

SYNOPSIS

A possible alternative to mechanical extract ventilation for kitchens and bathrooms is passive stack ventilation (PSV). BRE has carried out work on this type of system in a test house under controlled conditions. To find out how well they worked in practice, four occupied dwellings were monitored over a period of 2 - 3 weeks each. Each dwelling had two ventilation ducts.

Air flow rates within the ventilation ducts were measured, together with humidities, temperatures and climatological data.

The results show that the risk of problems due to condensation can be reduced by the use of this type of ventilation system.

The systems were found to have been poorly installed and where possible the faults were corrected as part of the study. Nevertheless the systems successfully kept down moisture levels below 70% RH for all but a small proportion of the time. The design and performance of the systems is discussed and advice given on how these could be improved.

This study demonstrates the need for clear and simple guidance on PSV systems to enable them to work to maximum efficiency.

1. Introduction

BRE has carried out a programme of work in a test house on the BRE site to monitor the performance of various Passive Stack Ventilation systems (PSV) under controlled conditions[1]. To determine how well this type of system performs under normal occupied conditions, four Local Authority dwellings in North London were monitored during the heating season. Each dwelling was monitored for a period of approximately 3 weeks.

The objectives of the study were:

1) To measure the flow rates obtained in the stack ducts and relate these to the humidity within the dwelling.

2) Determine whether the PSV systems would keep the relative humidity at sufficiently low levels to minimise the risk of condensation.

3) Assess the design and installation of each system and give advice on possible improvements.

2. Description of dwellings and PSV systems.

2.1 Dwellings

The four dwellings which were monitored were all 2-storey, end-terraced maisonettes above flats, thus occupying the second and third storeys of 3-storey blocks. The dwellings were built in the mid 1970s, originally with flat roofs and wooden-framed, single-glazed windows. They were refurbished approximately 2 years before testing, with the addition of pitched roofs, double glazed windows and PSV systems for the kitchen and bathroom. The position of the PSV serving the kitchen is somewhat unusual in that it is situated on the landing at the head of the stairs, the kitchen being at the foot of the stairs. In 2 of the dwellings (C and D) the kitchen was open plan with the hall and in the other 2 it was separated from the hall by a door.

2.2 PSV systems

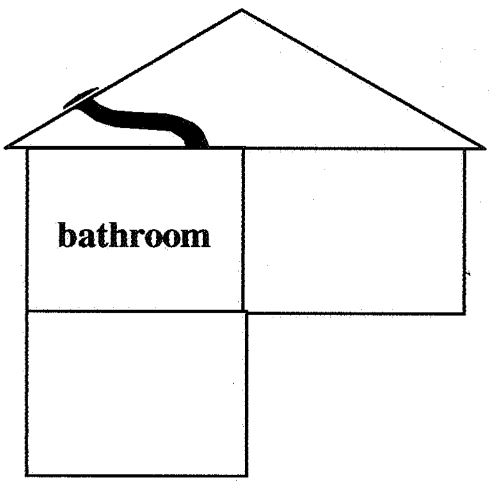

Figure 1: Design of bathroom PSV

The ducting used for the PSV systems was an insulated, flexible type, with 155 mm diameter used for the landing and 100 mm diameter for the bathroom. The landing duct terminated at a ridge ventilator whilst the bathroom one had been designed to terminate at a tile vent, situated low down on the roof, which was little higher than the bathroom ceiling (figure 1). The ceiling outlets were of the circular "valve" type with a central adjusting section to regulate air flow. On most of the systems tested these inlets were initially fairly tightly shut, thus restricting the air flow, most probably because this is the way they had been delivered from the manufacturer and had not been opened properly when fitted.

On inspection, the PSV systems were found to be badly installed with tight bends, much excess ducting and no supporting framework. In the first dwelling the landing duct had, in fact, become detached from the ridge ventilator and was lying on the floor of the loft. The bathroom ducts were of very poor design, having little difference in height between the ceiling outlets and the tile vent terminals on the roof slope.

3. Monitoring programme.

In view of the poor installation of the PSV systems, it was decided to monitor them for a period of time 'as found' then improve them as much as possible by taking out excess ducting, straightening bends and opening up ceiling outlets. Monitoring would then be carried out over a further period to determine the effects of the modifications. In this way some measure of how a systems' performance could be affected by bad design and/or workmanship could be assessed. It should be said that even in the improved state these systems would fall well short of current guidance given by BRE[(2)]. In the case of the first dwelling, where the ducting had become detached, no 'as found' condition was tested. Table 1 shows the different conditions monitored for each dwelling.

Dwelling	Condition 1	Condition 2	Condition 3
A	As found *	Vents opened fully	-
B	As found	Ducts shortened	Vents opened fully
C	As found	Vents opened fully, ducts shortened	-
D	As found	Ducts shortened	Vents opened fully

* *Landing system was disconnected when found, this was reconnected before monitoring commenced.*

Table 1: Monitoring conditions in test dwellings

When monitoring had been completed, the air leakage of the dwelling was measured using the 'Fan pressurisation ' technique as described by Stephen [3]. In brief, the air leakage is measured by sealing a fan into an external doorway and measuring the pressure difference between inside and outside simultaneously with the air flow through the fan. This procedure is repeated for several different pressure differences. The air leakage can then be calculated and is given as the air change rate obtained at an applied pressure difference of 50 Pa.

4. Monitoring equipment.

Each duct had equipment installed to measure :

air flow velocity,
humidity at inlet,
temperature at the top end.

In addition, temperatures were measured in the bathroom and landing immediately below each stack. Local wind speed and direction, and external temperature were measured at a mast attached to the gable end of the block approximately 2m above the eaves. This location was less than ideal so wind data was also obtained from the London Weather Centre, which is the nearest meteorological station to the test site.

All instruments were scanned once every 10 seconds and half-hourly averages recorded on 'Squirrel' data-loggers.

5. Results

5.1 Increases in flow rates after modifications

The data sets obtained during each monitoring period were analyzed in the same way as the previous sets from the BRE test house[1]. The air flow rates monitored in the landing stack of dwelling C were not used in the analysis as, due to instrument malfunction, they were not considered sufficiently reliable.

Regression analysis was carried out of duct air flow rate with the square root of the temperature difference and wind speed. From the results of this analysis curves were drawn of stack flow versus temperature difference up to 20° C for a typical wind speed of 4 m/s. Figure 2 shows the results obtained from dwelling A, which are typical of the increase in flows achieved in three of the four dwellings. In each plot the lower line is the 'as found' condition and the higher one, after modifications.

In dwellings A, B and C the flow increased considerably after modifications had been made to the systems. There were, however, no significant increases in the flow rates measured in dwelling D, possibly because there was very little excess ducting to remove and the ventilators were opened fairly well in the 'as found' condition.

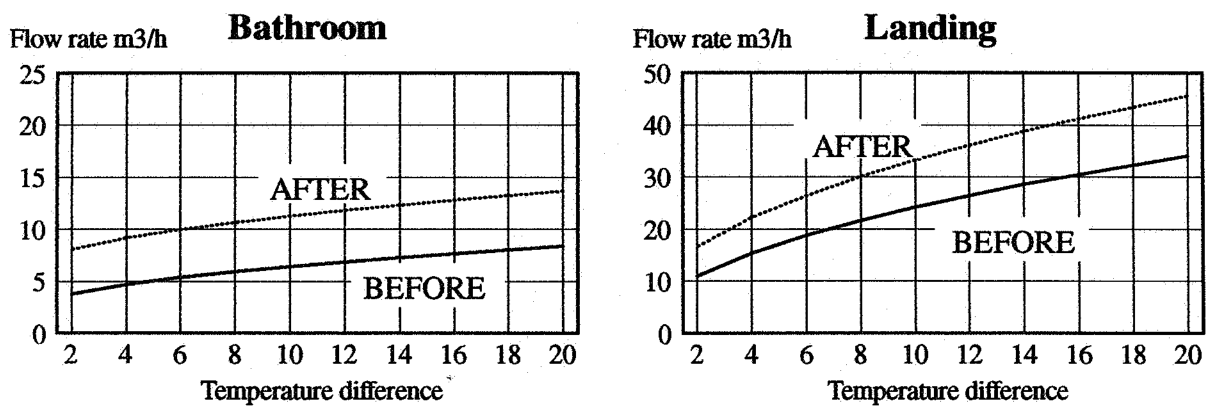

Figure 2: Typical increase in duct flow rates after modifications to PSV systems.

5.2 Moisture removal

To determine how well the PSV systems coped with moisture removal, cumulative frequency histograms were drawn of humidity levels in bands of 10% RH and the percentage of time that each level was exceeded. Figure 3 shows the results in the bathroom and landing of dwelling A after modifications. In the case of the bathroom, the humidity was always above 50% and the landing, 40%. The height of the 70% bar indicates the proportion of the time that the relative humidity is above 70%. In dwelling A 70% was exceeded 10% of the time in the bathroom and 0% on the landing. Table 2 gives the percentages for all four dwellings. If 70% RH is exceeded for lengthy periods then mould growth may occur[(4)]. In dwellings A,B and C the humidity levels measured are unlikely to give rise to problems of mould growth. In dwelling D, the percentage of time when 70% RH was exceeded is slightly higher in the 'as found' condition (24% of time in the bathroom) and there was indeed some evidence of mould growth. This was, however, a dwelling which was inadequately heated and where wet washing was hung indoors, not only in the bathroom but also in the kitchen, hall and landing areas. The decrease in RH above 70% shown in table 2 for this dwelling is not attributable to any modifications carried out as the was no increase in flow rates, as stated earlier. The internal temperatures, however, were higher during the measurements made after modifications thereby reducing the RH levels. It is probable that with the internal temperature maintained at the higher level, the mould growth found in this dwelling would not increase.

The design of the system was, however, still well below the standard of the new guidance.

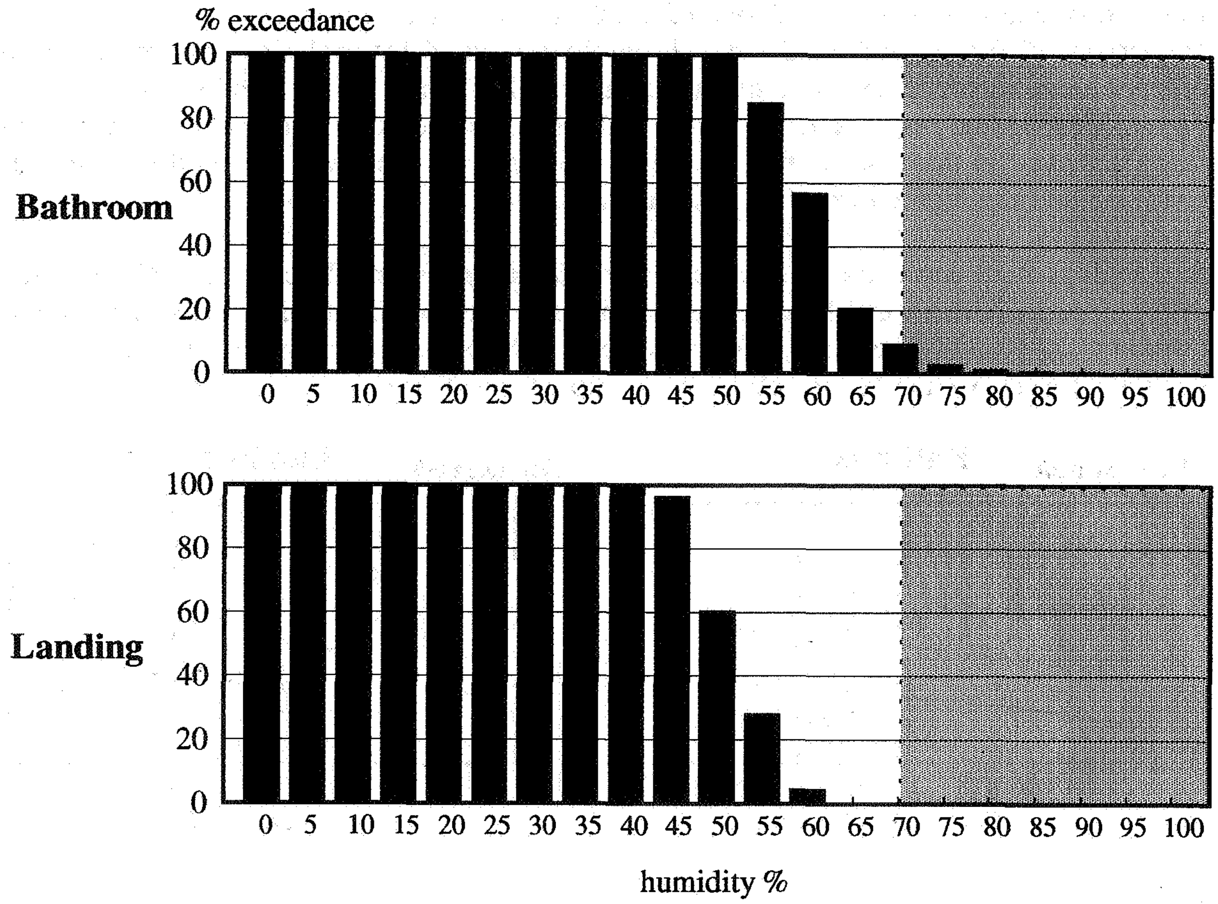

Figure 3: Cumulative frequency histograms for dwelling A

Dwelling	Bathroom		Landing	
	Before mods.	After mods.	Before mods.	After mods.
A	16.6%	9.3%	0.9%	0%
B	2.7%	2.1%	0%	0.2%
C	0.6%	0%	0.9%	0.7%
D	24.0%	8.5%	7.9%	0.1%

Table 2: Percentage of time RH greater than 70%

6. Air leakage measurements

Table 3 gives the results of the air leakage tests carried out in each dwelling with passive stack vents open, and shows that they lie within the range 10 - 12 air changes at 50 Pa. Analysis, by BRE, of a sample of U.K. dwellings show that the median air leakage at 50 Pa is in the order of 14 air changes per hour[5], the dwellings used in this study are, therefore, more airtight than average.

BS 5250[4] recommends ventilation rates of between 0.5 and 1.5 air changes per hour for the control of condensation. By applying the 1/20 rule (natural ventilation rates are approximately 1/20 of the air leakage rate measured at 50 Pa), to the air leakage rates of the test dwellings, it can be seen that they equate to natural ventilation rates of just over 0.5 ach. The dwellings should, therefore, have sufficient ventilation for control of condensation.

Dwelling	air changes per hour (+ve pressure)	air changes per hour (-ve pressure)	air changes per hour (mean)
A	11.5	10.8	11.2
B	12.1	11.7	11.9
C	11.2	10.5	11.9
D	10.4	9.8	10.1

Table 3: Results of air leakage measurements with passive stack vents open

7.Reverse flow

In certain circumstances reverse flow may occur within the PSV systems. This was detected by observing the stack temperatures in relation to the room and external temperatures. In the landing stacks temperatures generally followed the same pattern of variation as the temperature in the room which they served, but occasionally, in house B, dropped towards the external temperature, indicating that cooler air from outside was flowing down the stack. This phenomenon occurred when the wind blew from particular directions and was possibly due to the influence of adjacent buildings. Greater periods of reverse flow were detected in all the bathroom stacks, due to the bad positioning of the roof terminal referred to earlier in section 2. An example of the deviations in temperature pattern can be seen in figure 4,

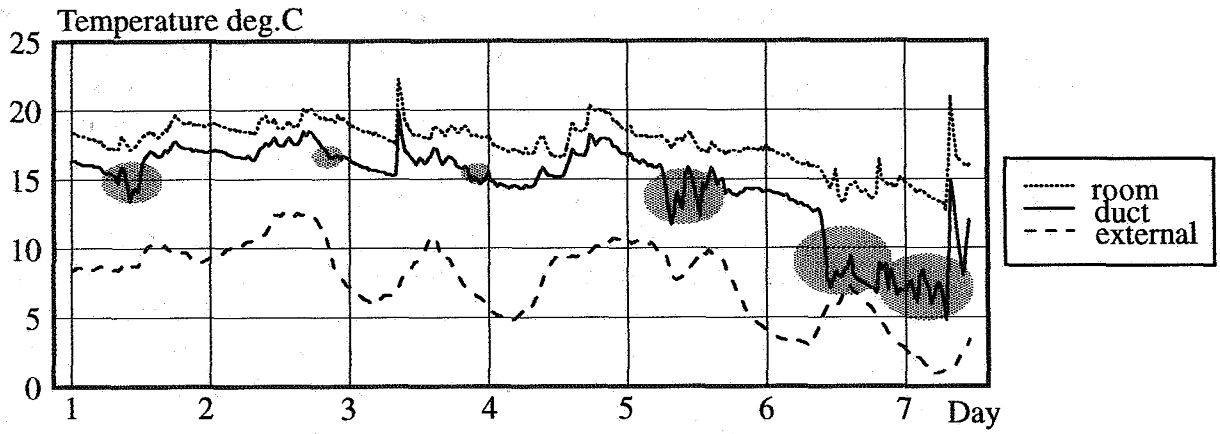

Figure 4: Temperatures in room, duct and outside, showing periods of possible flow reversal

where the top line is the room temperature, the bottom line the external temperature and the centreline, the temperature in the duct. The periods where the duct temperature deviated from the pattern followed by that of the room temperature show when reverse flow was possibly occurring and are indicated by shading.

Even in this extreme case however, flow reversal only occurred a small proportion of the time and was clearly limited in amplitude and had little effect on the average ventilation rate.

8. Discussion

The design of the PSV systems monitored in this study was poor.
The systems serving the kitchen were obviously installed with low cost in mind and would have been more effective if the duct had run from the kitchen. The way in which they were installed i.e.. the outlet from the landing, resulted in air being extracted from other areas of the dwelling as well as the kitchen, so although the flow rates would appear to be adequate it should be remembered that not all the airflow was from the kitchen.

In the case of the bathroom systems, the duct should have been taken straight through the roof and terminated at ridge height to give a greater height difference between inlet and outlet and to avoid flow reversal. This would also have eliminated the right angle bends in the ductwork which reduce the flow rate. An alternative solution could have been to terminate the ducting at the ridge although this would have necessitated using longer lengths of ducting so would have been slightly more expensive. Figure 5 shows the existing design (a) and alternatives (b) and (c), which would reduce the possibility of reverse flow to a minimum.

If we look back at the air leakage results, a whole house infiltration rate in the region of just over 0.5 air changes per hour under normal climatic pressures, is indicated. If we add the flow rates measured in the landing and bathroom stacks of each dwelling, at a temperature difference of 10° C and a wind speed of 4 m/s, an average flow rate of around 45 m^3/h is obtained. This represents a contribution to the whole dwelling ventilation rate of 0.25 air changes per hour. In simple terms this suggests that the PSV systems contribute almost half the total ventilation. However, to determine the interaction between PSV and whole house ventilation it is necessary to use a single cell ventilation model such as BREVENT [6].
As previously mentioned, these dwellings are slightly tighter than the UK average, one could thus assume that a significant amount of water vapour is removed, not by excess infiltration through the fabric of the dwelling but by the flow through the PSV ducts.

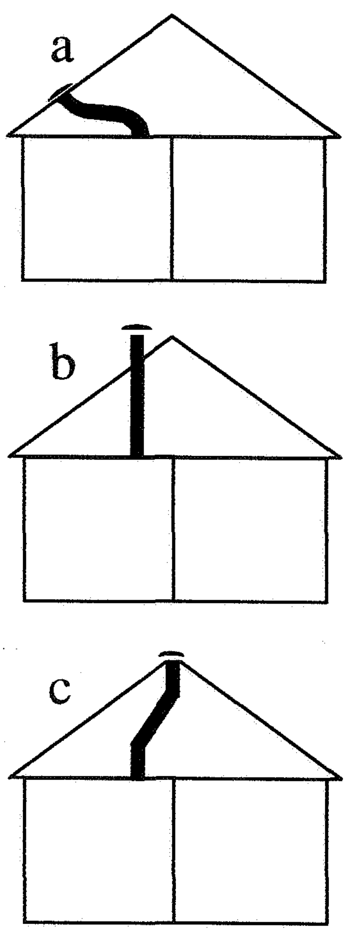

Figure 5: Existing bathroom PSV design and alternatives for improved efficiency.

In the 'as found' condition, moisture was kept to an acceptable level in all but one of the dwellings. This was, as stated earlier, a dwelling where drying of washing indoors seemed to take place for a large proportion of time.

9. Conclusions

1) Four occupied dwellings, with passive stack ventilation systems, were monitored over a period of weeks to determine how well the PSV systems performed.

2) The PSV systems in the bathrooms had been designed so that the ductwork ran parallel to the loft floor and had sharp bends, restricting the airflow.

3) The majority of the PSV systems monitored had been badly installed with much excess ducting, too many bends and no support.

4) The systems gave better airflow performance when excess ducting had been removed and any bends straightened out as much as possible.

5) In spite of poor design and installation the systems coped well with removal of moisture and kept the relative humidity levels below 70% for all but a small percentage of the time.

6) Care should be taken when designing and installing PSV systems to ensure that :
   a) Ductwork is as straight as possible,
   b) stacks terminate at or near the ridge,
   c) tile ventilators are not used as they can cause reverse flow in the system, this may give rise to occupant discomfort and reduced overall flow rates up the duct and
   d) terminals are opened sufficiently so that they do not restrict air flow

7) This study demonstrates the need for clear and simple guidance on passive stack ventilation systems as contained in the Reference 2, but notwithstanding unsatisfactory design and installation these systems performed satisfactorily in terms of keeping down relative humidity below 70%.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Passive Ventilators in New Zealand Homes: Part 1 Numerical Studies and Part 2 Experimental Trials

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Synopsis

New Zealand homes have traditionally been ventilated through open windows and by background infiltration. In recent times, new materials and construction practices have led to more airtight buildings, and open windows are seen more and more as a security risk. These trends call for new ventilation options that are inexpensive and consistent with home security, weathertightness and draught control. This paper is part one of a study of passive ventilation options for NZ homes. It explores numerically a range of ventilator sizes and locations in typical homes modelled in the climate of major New Zealand cities. Part two offers experimental verification of the ventilator performance data calculated here.

A numerical multi-zone air flow model was used to calculate the effect of adding stack and window type passive vents to houses of a range of airtightness levels. Wind pressure was found to be the dominant driving force of air flows delivered by window-mounted passive ventilators. Stack ventilators reduced the strong dependence of window ventilator air flows on wind speed when both types were present in a building, but when the ventilation system made small changes to the overall airtightness of the house, the role of the stack ventilator was less obvious. A simple linear function linking ventilator opening area with average added ventilation rates is presented for wall-mounted passive ventilator systems in NZ buildings.

1. Background

In older homes in New Zealand, air infiltration alone has been sufficient to meet most ventilation needs [1] and the practice of opening windows may have been rarely necessary for critical contaminant control. The airtightness of New Zealand homes has, however, steadily changed [2] and those of more recent construction are more airtight than older houses; mainly because sheet interior linings which eliminate joints between materials, and more accurately gauged materials and fittings have become more widely used. Natural air infiltration will still provide useful ventilation, (particularly for larger more complex designs) but for more simple designs (often low cost housing [3] it is desirable to add a further 0.2-0.3 air changes per hour (ac/h) of secure and reliable background ventilation.

Ventilation provisions for new homes in NZ are now defined by Approved Document G4 of the New Zealand Building Code [4]. One acceptable solution to achieve this (G4/AS1) is to provide openable window areas of at least 5% of the floor area in each room. The aim of this work is to develop new ventilation solutions that are compatible with trends in the construction and occupancy of new homes.

2. Modelling passive ventilator performance

A numerical multi-zone airflow model developed by Walton [5] at the National Bureau of Standards was used to determine the marginal change to natural ventilation in New Zealand homes caused by adding passive ventilation openings. Air leakage and wind pressure coefficient datasets were developed for six buildings covering a range of airtightness levels seen in NZ houses. Each dataset corresponds to a real house and is based on measured airtightness and ventilation performance data. Agreement between measured and calculated natural ventilation rates in the living spaces of these houses has already been established in earlier work, with houses C,D and E in this study being the same houses as B,E and C in the earlier study of inter-zone leakage paths [6]. Figure 1 shows the spread in airtightness of houses A to F compared with the histogram of airtightness for houses constructed in New Zealand since 1960 [2].

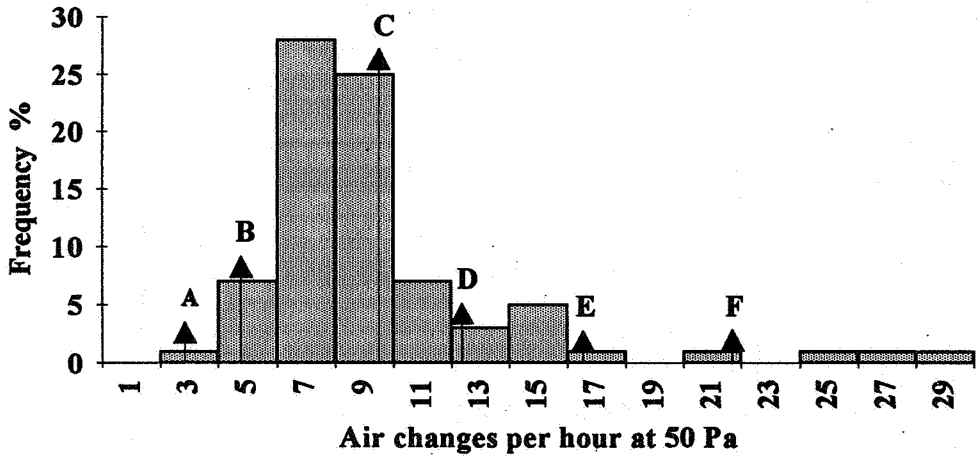

Figure 1: Airtightness of model houses compared with histogram of houses constructed since 1960.

3. Air leakage details of six model houses

The six residential buildings modelled in this study are labelled A - F. All had suspended floors but houses A, B, C and F were clad in lightweight timber or fibre cement materials with insignificant leakage paths directly linking subfloor and roof space zones. Houses D and E, on the other hand, were brick clad with large leakage paths through the wall cavity connecting subfloor with roof space zones. These two groups of houses were modelled in much the same way [6] but with an extra leakage path linking subfloor with roof space for houses D and E. Figure 2 gives the node diagram for house C.

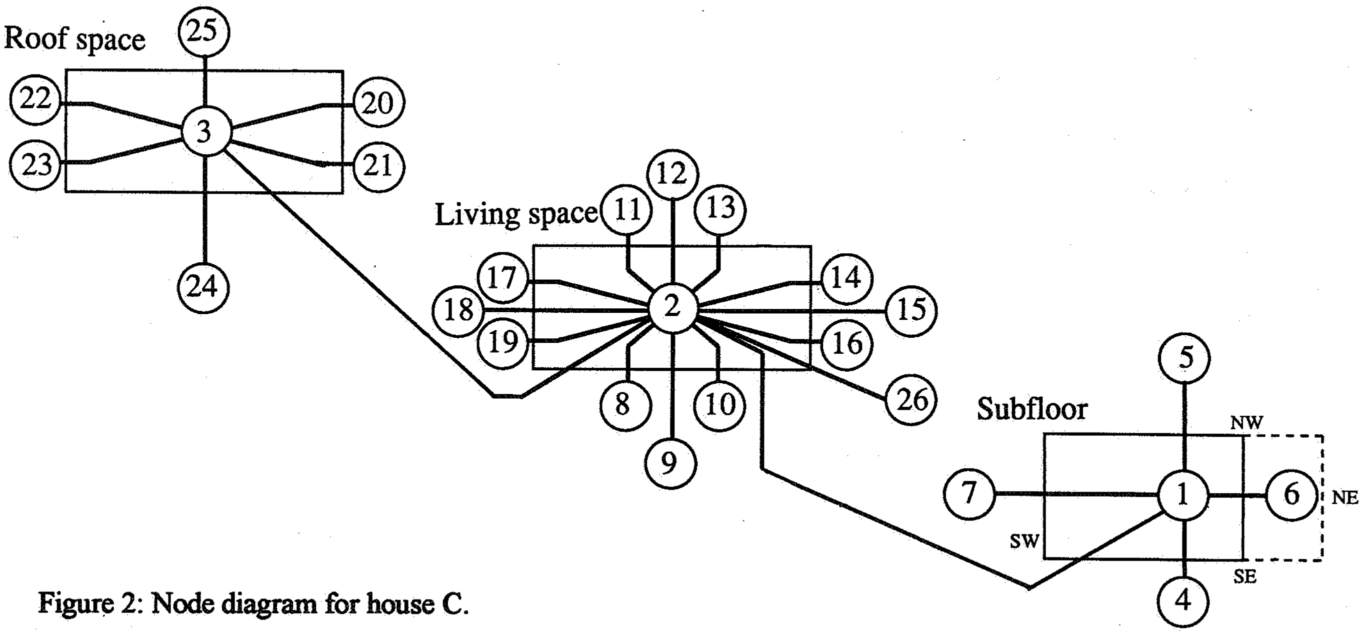

Figure 2: Node diagram for house C.

The basic geometry and airtightness characteristics for the 6 houses are given in Table 1, where N50 is the air leakage rate at 50 Pa expressed in air changes per hour, and the coefficient and exponent are those from the normal exponential equation linking air leakage rate Q with applied pressure ΔP.

$$Q = C\,\Delta P^n \qquad \text{where} \quad Q = \text{Air leakage rate} \quad \mathrm{m^3/s}$$
$$\text{and} \quad \Delta P = \text{Pressure difference} \quad \mathrm{Pa}$$

	House					
Factor	A	B	C	D	E	F
Floor area m^2	94	73	88	95	98	121
House volume m^3	255	175	210	229	234	276
Storeys one/two	one	one	one	one	one	two
Wind exposure	sheltered	sheltered	sheltered	sheltered	sheltered	sheltered
N50 ac/h	3.0	4.7	9.4	12.7	16.2	21.8
Coefficient C	0.0168	0.0174	0.0370	0.0522	0.0860	0.166
Exponent n	0.65	0.66	0.69	0.70	0.64	0.59

Table 1: Dimensions and airtightness characteristics of six example houses

Airtightness tests have given the overall leakage characteristics for each zone but have not located the specific leakage sites and their sizes over the building envelope. The distribution of leakage sites over the building and the wind pressure coefficients used, are described by Bassett [6].

4. Results of simulations

4.1 Natural infiltration in the example houses

The range of infiltration rates for houses A-E located in the Wellington winter climate is given in Figure 3. Here the median and quartile infiltration rates are calculated assuming medium heating (heating schedules indicated below) and plotted against the airtightness coefficient . The median and quartile have been chosen to represent the spread of the highly skewed distribution of natural infiltration rates. Houses A and B are the two most in need of added ventilation. Houses C,D, and E have median infiltration rates marginally less than 0.5 ac/h and house F is unlikely to need ventilation from any other source.

High heating	- Indoor temperature at 20 °C at all times.
Medium heating	- Indoor temperature floating 6 °C above outdoors with time lag and a minimum of 12 C°.
Low heating	- Indoor temperature floating 6 °C above outdoors with time lag.

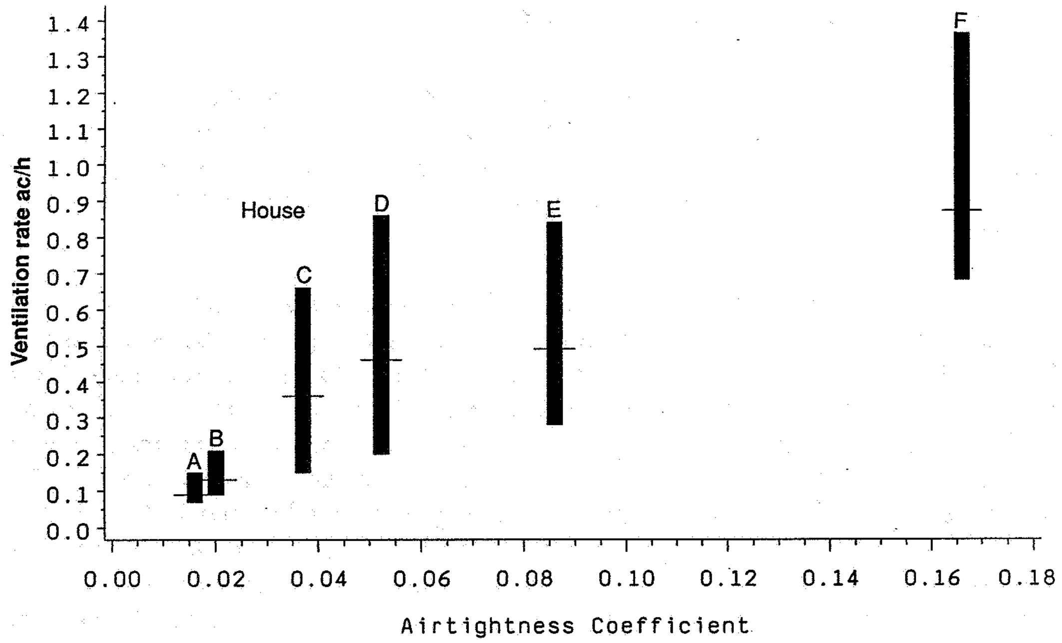

Figure 3: Median and Quartiles of hourly infiltration rates in houses A - F in the Wellington winter with a medium level of heating.

Turning now to the influence of indoor and outdoor climate. Figure 4 gives median and quartile infiltration rates plotted against winter heating degree days, for house C in the four climates and with High, Medium and Low levels of heating. The level of heating is clearly less important than location. Most of the location effect is considered to be a wind speed effect illustrated by the median infiltration rates increasing from Christchurch to Invercargill to Auckland to a high in Wellington in line with the same trend in mean wind speeds of 2.20, 2.22, 3.16 to 3.73 m/s respectively.

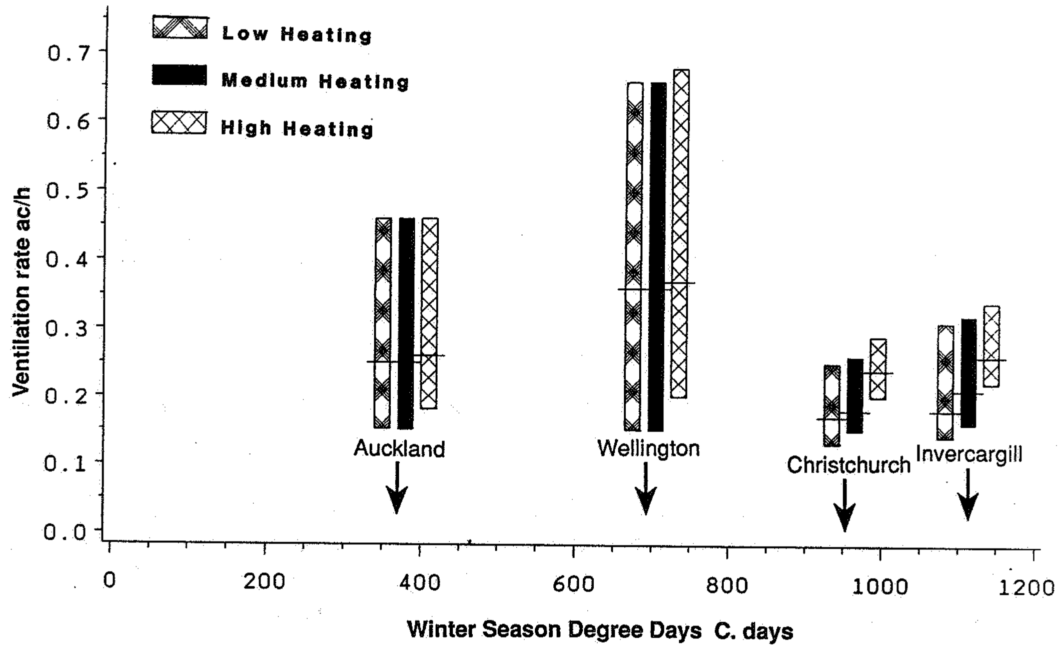

Figure 4: Median and Quartile ventilation rates for house C modelled in four climates and with three levels of heating.

4.2 Marginal changes attributed to ventilation openings

The marginal change in natural ventilation attributed to window ventilators and stack shafts has been determined for all six buildings in a matrix of climate, ventilator mix and ventilator size. Window vents were modelled as a single opening at mid stud height in proportion to wall area for each wall orientation, and stack ventilation was modelled as a single shaft from ceiling level to an external wind pressure coefficient of -0.15.

Adding three combinations of window and stack ventilators to house A in the Wellington climate changed both the range and median of the natural ventilation rate. Figure 5 shows the data plotted against the total building airtightness coefficient. Because the building has been modelled in the Wellington climate, the range of ventilation rates indicated here by the quartiles will be the extreme of the four available urban climates. Adding a single stack vent of increasing size has marginally increased the ventilation rate in the early stages but beyond this, the internal pressure has been dominated by the vent and the air pressure difference across the other leakage openings has changed no further with increasing vent size. An equal mix of vent type or window vents alone has increased the median ventilation rate in approximate proportion to the total house airtightness coefficient. As the effect of the ventilators begins to dominate the leakage openings in the house envelope, a significant difference between a window only and a mixed ventilation system emerges. The mixed system (equal window and stack vent size) achieves a more uniform ventilation rate than a window vents only system.

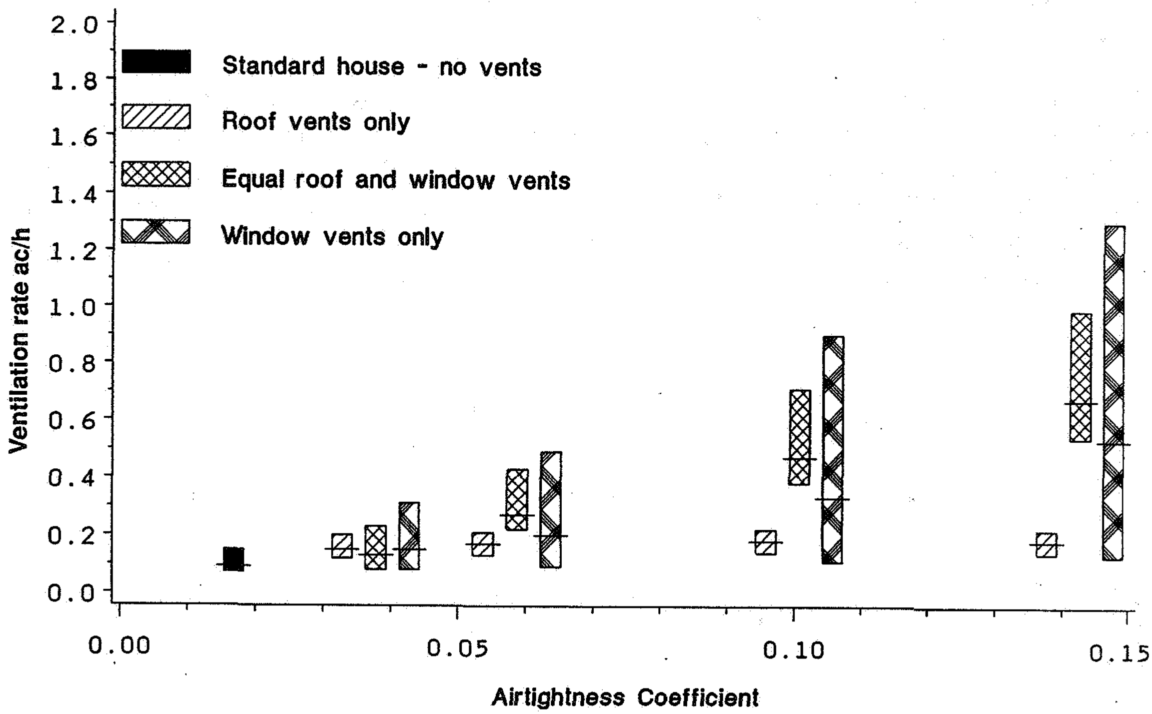

Figure 5: Ventilation rate trends in house A in the Wellington winter climate with increasing passive ventilator size.

For house C modelled in the Wellington climate and with the same ventilator options, a similar pattern emerges. Figure 6 shows that where a major change has been made in the building airtightness level with passive vents, then the spread of ventilation depends strongly on the mix of stack and window vents. For less extreme changes there is less difference between a mixed window and stack system and a window vents only system.

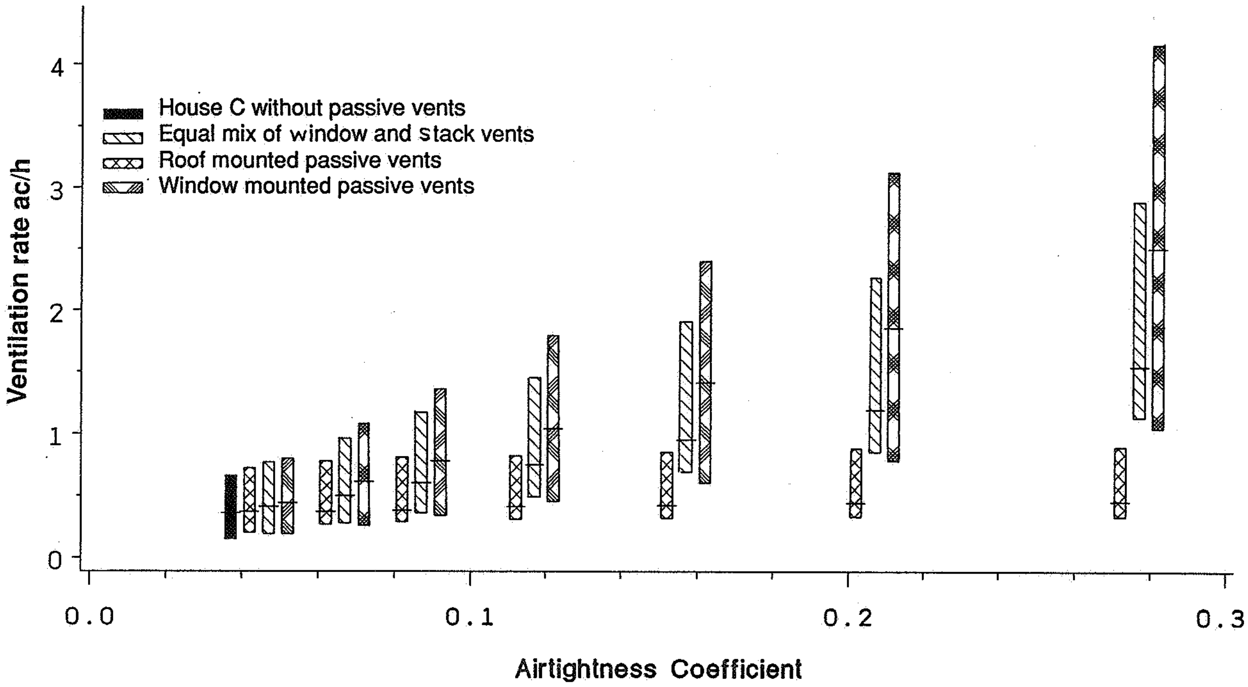

Figure 6: Changes to median and quartile natural ventilation rates in house C fitted with different ventilator options.

A more detailed look at the distribution of ventilation rates with the three ventilator mixes modelled in Figure 6 is given in Figures 7 and 8. In Figure 7 the effect of a relatively significant (100% or 0.03) change to the airtightness coefficient of house C is examined. Compared to the pattern of natural air infiltration calculated for the building, the addition of window vents and mixed vents has increased the frequency of ventilation rates above 0.5 ac/h at the expense of the lower ventilation rates. At times of low wind speed, stack driven airflows have provided ventilation rates of around 0.3 ac/h resulting in a more compact distribution of ventilation rates being delivered by the mixed stack and window vent system.

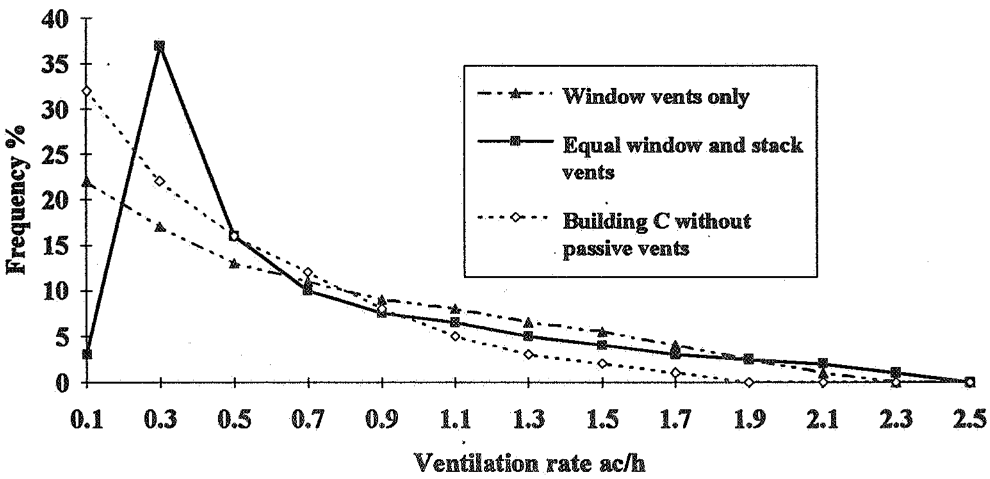

Figure 7: Distribution in hourly ventilation rates for house C without passive vents and with two different ventilator systems.

With larger vents in house C, (4.6 times the original airtightness coefficient, ie. 0.17) Figure 9 shows that the trends in Figure 7 have strengthened in this extreme case where the ventilation system dominates infiltration. Very low ventilation rates (less than 0.5 ac/h) were essentially shifted into the 0.5 - 1.0 ac/h range by the combined window and stack system. Very high ventilation rates driven by high wind speeds are also less common with the mixed arrangement of ventilators.

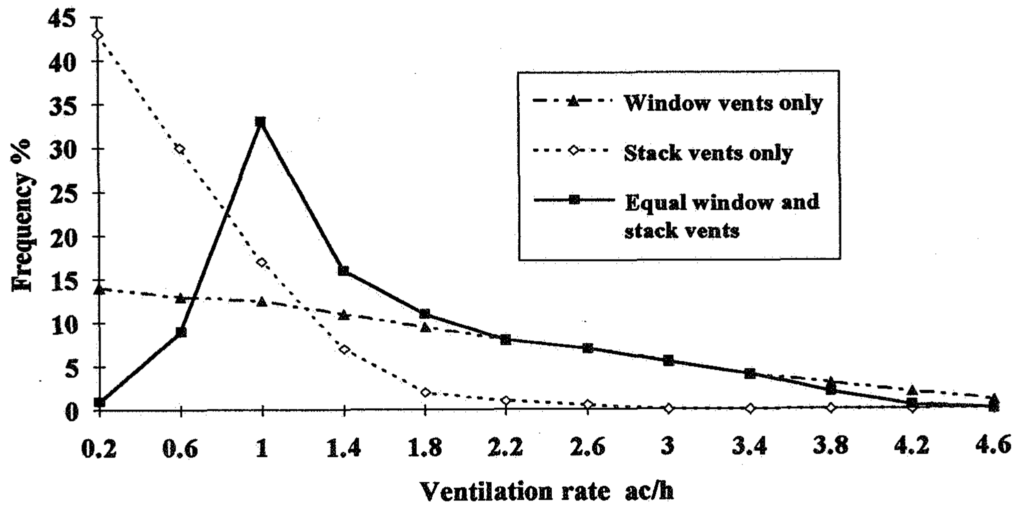

Figure 8: Distribution of ventilation rates in house C with three ventilation strategies contributing to an overall airtightness coefficient of 0.17.

4.4 Generalised guidance on ventilator sizes

Although the detailed performance of a ventilator system will depend on a large number of factors such as building geometry, location and wind exposure, an indication of the likely effect of the following two most promising passive ventilator configurations in common house types can be determined.

1 Window ventilators distributed around the building pro-rata with wall area.
2 An equal mix of window and stack ventilators.

A further series of simulations has been completed for each of these two ventilation strategies applied to houses A,C,E and F located in each of the four city climates and with typical urban sheltered exposure to wind (tracer gas measurements of air infiltration rates in houses has shown that most houses in urban subdivisions can be described with the sheltered level of wind exposure [6]). The marginal change in the average natural ventilation rate was determined for changes in building airtightness coefficient in the range 0.03 to 0.1. An example of the effect of this is given in Figures 9 and 10 for a 0.05 change in overall building airtightness coefficient. In Figure 9 the marginal increase in the average ventilation rate is averaged over the four houses and plotted against mean wind speed. In Figure 10 the ventilation rate data is climate averaged and plotted against the building (before addition of passive vents) airtightness coefficient.

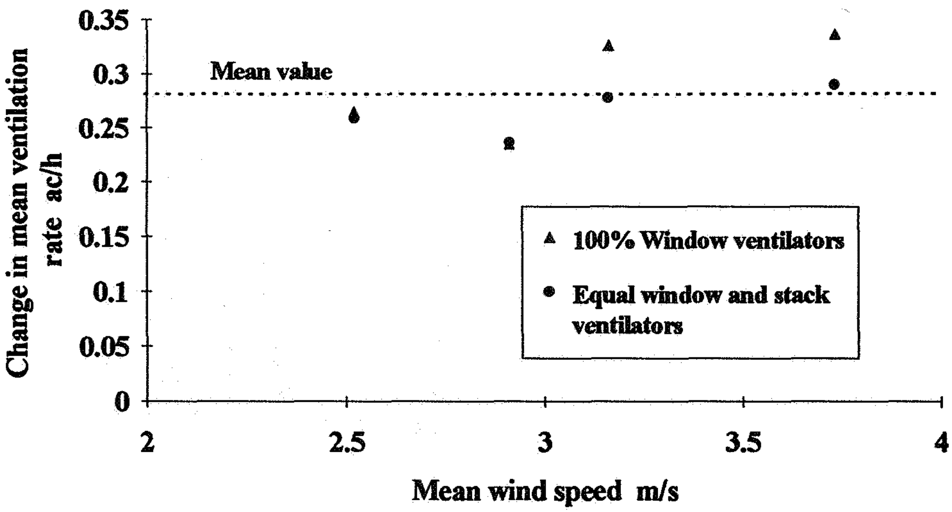

Figure 9: Marginal changes in the average ventilation rate in houses A,C,E and F modelled in four city climates and plotted as a building average against mean wind speed.

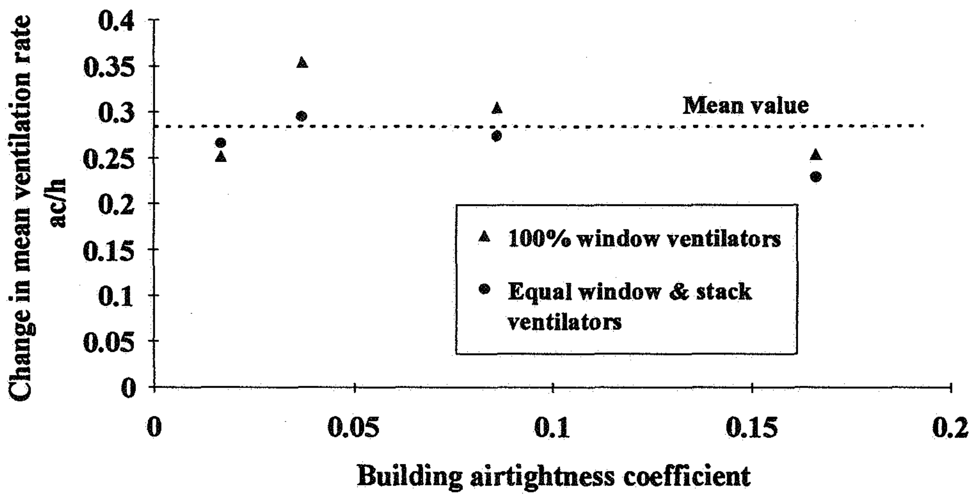

Figure 10: Marginal changes in the average ventilation rate in buildings A,C,E and F modelled in four city climates and plotted (climate averaged) against building airtightness coefficient.

The average marginal changes to the ventilation rate (building and climate averaged) have then been plotted in Figure 11 against the airtightness coefficient of the ventilator system.

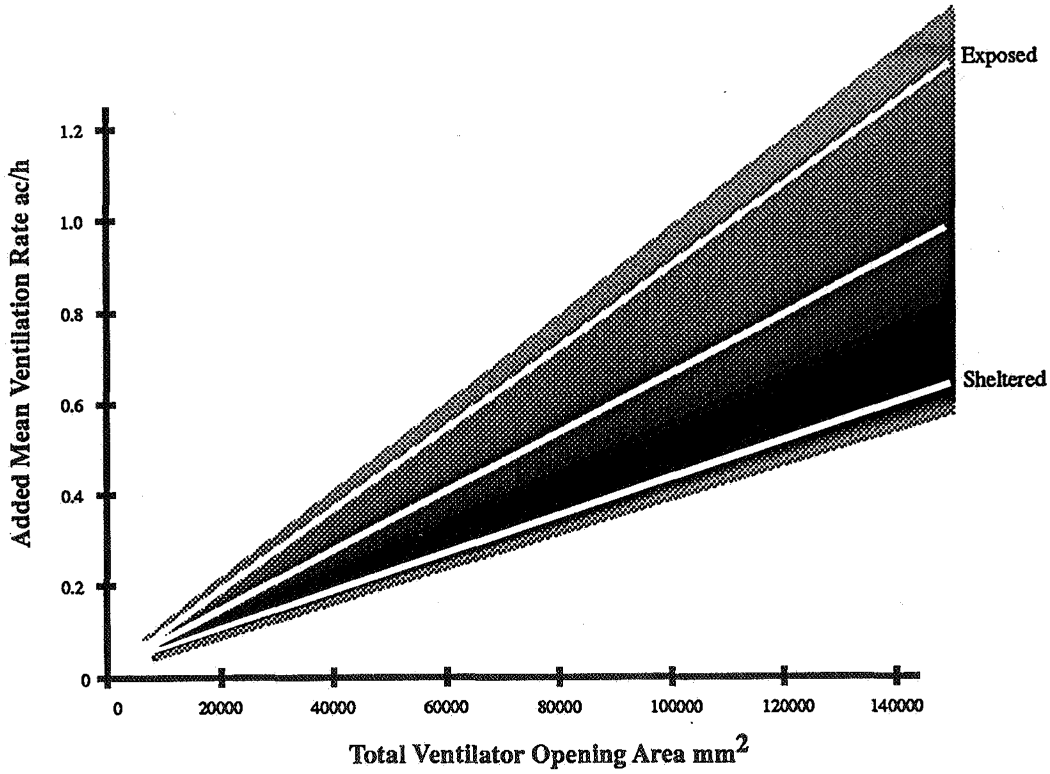

Figure 11: Average change in ventilation rate as a function of ventilator airtightness coefficient for a limited range of climate and building variables.

Figure 11 gives an indication of the average change in natural ventilation that can be expected from the addition of window ventilators or an equal mix of window and stack vents. A band has also been included to indicate the ranges appropriate to most buildings in urban sheltered locations (shown in dark shading) and for less common exposed locations (shown in lighter shading). These data are based on limited simulation of four buildings in four New Zealand city climates and experimental support for these conclusions is given in part two [7] of this paper. Earlier in this paper it was indicated that approximately 0.2 - 0.3 ac/h of additional ventilation would usefully boost natural ventilation in the more airtight types of homes to closer to the 0.5 ac/h level recommended for adequate indoor air quality. The addition of 0.2 ac/h would require passive ventilation with an area of 40,000 - 60,000 mm². This is comparable with the (DoE, 1990) requirements [8] which define passive ventilator sizes to provide background ventilation in habitable rooms in the UK. Trickle ventilators must have an open area of not less than 4,000 mm² in each room.

5. Conclusions - passive ventilators for NZ homes

A numerical study of the performance of passive ventilation systems in New Zealand homes has been completed. This simulated the performance of a variety of window and stack ventilator configurations in six houses sited in four New Zealand city climates. Changes to average ventilation rates and to the distribution of hourly ventilation rates were examined and the following conclusions drawn:

1. Air flow rates through passive vents in typical New Zealand building and climate combinations were shown to be primarily wind driven. The time averaged performance of ventilator systems containing stack components were little different from window only systems unless the ventilation system made large changes to the airtightness of the building.

2. The average ventilation rates delivered by passive systems containing a mix of stack and window ventilators were comparatively insensitive to indoor temperatures. A single stack ventilator on its own made little change to average ventilation rates in the four example houses, even when this dominated the airtightness of the house.

3. The distribution of hourly ventilation rates depended on the proportions of stack and window ventilators. Mixed window and stack ventilation systems came the closest to delivering an ideal (0.5 to 1.0 ac/h) distribution of hourly ventilation rates, particularly when added to the more airtight house types.

4. When the passive ventilation system made a small change to the airtightness of the house, the difference between mixed stack and window vent systems and window only vent systems was small. This would generally be the case unless the house is particularly airtight.

5. An approximate window ventilator sizing guide has been proposed consisting of a linear relationship between the ventilation rate added to a house and the airtightness coefficient of the ventilation system.

A further report [7] describes the measurements of air flows delivered in a number of passive ventilated houses as verification of the results calculated here. There are additional considerations such as acoustic isolation and possible effects of passive vents in fire still to consider.

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The Role of Ventilation

15th AIVC Conference, Buxton, England
27th-30th September 1994

Passive Ventilators in New Zealand Homes
Part 2 - Experimental trials

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Synopsis

This paper is part two of a study of passive ventilation options for NZ homes. The first part explored numerically, a range of ventilator sizes and locations in typical homes modelled in the climate and wind conditions of urban New Zealand. This paper offers experimental verification of the ventilator performance data calculated earlier.

Passive ventilators were installed in the window systems of three houses in Wellington. Air tightness characteristics and wind speeds were measured and used to predict ventilation rates for "vents open" and "vents closed" conditions. These predictions compared favourably with ventilation rates measured with an automated tracer gas dilution method, offering support for the earlier numerical determination of passive ventilator performance. A simple linear function linking ventilator open area with average ventilation performance has been supported.

1. Background

The numerical study of passive ventilation options [1] has drawn the following conclusions linking passive ventilator performance with as far as ventilator size and location is concerned:

1. Air flow rates through passive vents in typical New Zealand building and climate combinations were shown to be primarily wind driven. The time averaged performance of ventilator systems containing stack components were little different from window only systems unless the ventilation system made large changes to the air tightness of the building.

2. The average ventilation rates delivered by passive systems containing a mix of stack and window ventilators were comparatively insensitive to indoor temperatures. A single stack ventilator on its own made little change to average ventilation rates in the six example houses, even when this dominated the air tightness of the house.

3. The distribution of hourly ventilation rates depended of the proportions of stack and window ventilators. Mixed window and stack ventilation systems came the closest to delivering an ideal (0.5 to 1.0 ac/h) distribution of hourly ventilation rates, particularly when added to the more airtight building types.

4. When the passive ventilation system made a small change to the airtightness of the building, the difference between mixed stack and window vent systems and window only vent systems was small. This would generally be the case unless the building is particularly airtight.

5. An approximate window ventilator sizing guide has been proposed consisting of a linear relationship between the ventilation rate added to a house and the airtightness coefficient of the ventilation system.

2. An experimental study of ventilator performance

The ventilation performance of window mounted passive ventilators has been measured in three houses to offer direct support for the numerical results described in part one of this paper [1]. An automated concentration decay procedure [2] has provided hourly records of ventilation rates, together with average wind speed, wind direction and zone temperatures for each house. Data were taken over periods when the houses were passively ventilated and, at other times, where the ventilators were closed. The measured ventilation rates delivered by the ventilator system are compared with ventilation rates expected from the numerical study.

2.1. Wind exposure details

The three houses used in this study were all single family detached homes located in Newlands, a developing suburb high on the hills to the north of Wellington harbour. The subdivision is on rising terrain exposed to winds from N through W to S. Because of this, the terrain has been classified as "Exposed rural" for the purpose of calculating wind speeds at roof height from site measured wind speeds. Measured wind speeds (with H_{Met}= 10m above ground level) were converted to wind speeds at roof height as follows:

$$V_{Roof} = V_{Met} \frac{\alpha_{Roof}\left[\dfrac{H_{Roof}}{10}\right]^{\gamma_{Roof}}}{\alpha_{Met}\left[\dfrac{H_{Met}}{10}\right]^{\gamma_{Met}}}$$

Where V =Wind speed at met or roof height m/s
α and γ are terrain parameters, 1 and 0.15 respectively.
H =Height of roof or meteorological measurements m

Wind pressures at the leakage openings were calculated using standard wind pressure coefficients given in Air Infiltration Calculation Techniques - an Applications Guide [3] together with data for subfloor ventilators taken from Bassett [4]. House A was isolated from other buildings at the time ventilation measurements were made so that "Exposed" case wind pressure coefficients were appropriate. Buildings B and C were located close to other buildings and "Sheltered" condition wind pressure coefficients have been assumed.

2.2. Air tightness of ventilators and envelope of three test houses

Two of the houses, A and C, were fitted with commercial window ventilators that could be closed with a sliding plastic cover. In house B, the windows were modified to simulate vents by securing the windows in a cracked open position. Airtightness characteristics of the houses with ventilators closed were measured and the results are given in Table 1. Airtightness characteristics are expressed as the volume air changes of air leakage with 50 Pa of applied air pressure difference (N50) and as an exponent and coefficient in the following equation linking air leakage rate Q with applied pressure difference ΔP.

$Q = C\,\Delta P^{n}$ where Q = Air leakage rate in m^3/s
and ΔP = Pressure difference in Pa

Buildings A and C were newly constructed timber framed homes but house A was a more complex two storey design. This difference in complexity may explain the observation that house C was twice as airtight as house A. House B was also light timber framed and on a suspended floor. It was as airtight as expected for this style and age of building.

Factor	House A	House B	House C
Floor area m²	97	99	83
House volume m³	228	239	199
Storeys	two	one	one
Airtightness details with vents closed			
N50	11.5	7.2	5.8
Coefficient C	0.0646	0.0454	0.0183
Exponent n	0.62	0.60	0.73

Table 1: Airtightness characteristics of three test houses

Table 2 gives the leakage characteristics of each house lot of ventilators, and shows how these were distributed around the building. For the vents in house B the leakage characteristics were determined from the difference between airtightness tests. For houses A and C, measurements in the laboratory have provided more accurate measurements of the ventilator air leakage characteristics than could be determined from house airtightness tests. (House B Ventilator 2) is simply another variation of (House B Ventilator 1) with the windows cracked open a little further. The ventilators in house C were fitted with a baffle that progressively closed with increasing wind pressure; reducing air flow at higher wind speeds. The result of this (* in Table 2) is an exponent of 0.35 and less dependence on wind speed than would be physically possible for fixed ventilators. The baffle restricts inward airflow more than outward air flow, and therefore the leakage characteristics are flow direction dependent. The effective leakage areas (ELA) are calculated assuming a flow coefficient of 0.6.

Whole House Passive Ventilator Leakage Characteristics					
	House A	House B1	House B2	House C	House C
Air flow direction	both	both	both	out-in	in-out
N50	2.1	5.3	9.5	0.9	2.1
Coefficient C	0.0185	0.0494	0.0860	0.0120	0.0160
Exponent n	0.51	0.50	0.51	0.35*	0.5
ELA mm²	24,000	64,000	110,000	-	21,000
Proportions of Ventilator Opening on each Building Face %					
N/NE	8	51	51	0	0
E/SE	36	8	8	25	25
S/SW	39	26	26	12	12
W/NW	17	15	15	63	63

Table 2: Leakage characteristics of passive vents in test houses A, B and C

In all houses, the ventilators were distributed around the walls on the basis of window sizes and not wall area. This means that in practice, ventilation area is unlikely to be uniformly distributed around a house and that ventilation delivered by a passive ventilator system will depend to some extent on wind direction.

2.3 Measured infiltration and ventilator performance

A sample of the measured ventilation data for each of houses A,B, and C, plotted against wind speed, is given in Figures 1,2 and 3 respectively. The plotted data are representative of one wind direction where data for both vents open and closed conditions are available.

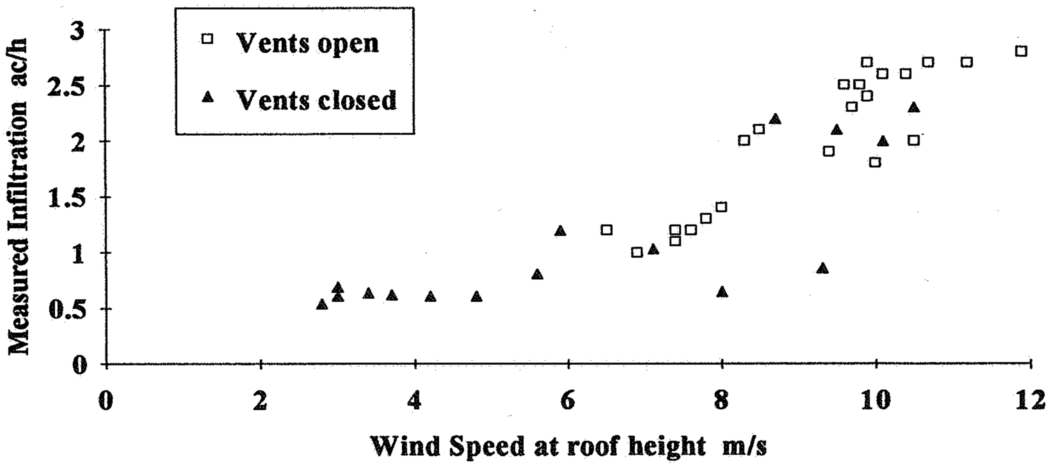

Figure 1: Infiltration in house A against wind speed at roof height with vents open and closed

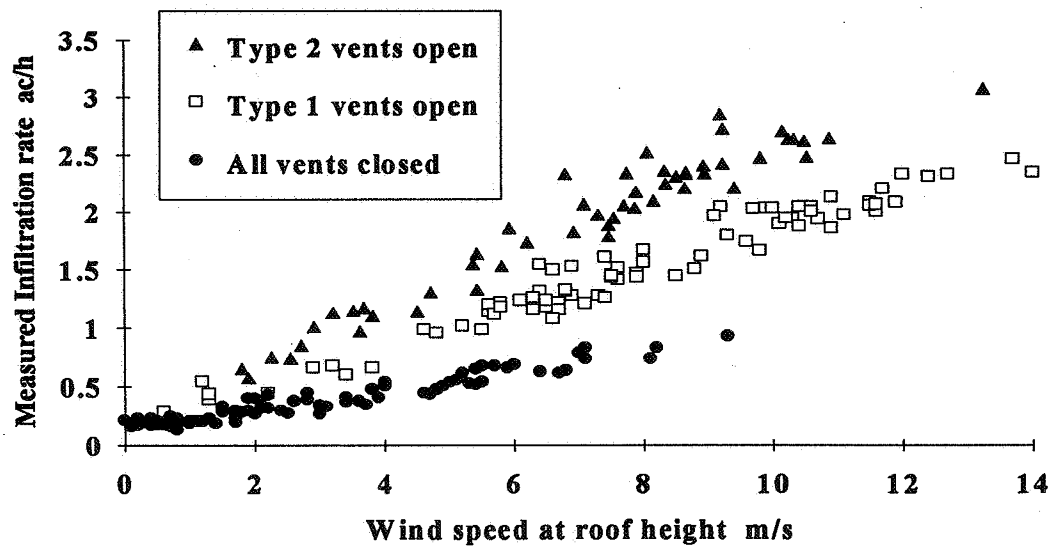

Figure 2: Infiltration in house B against wind speed at roof height with vents open and closed

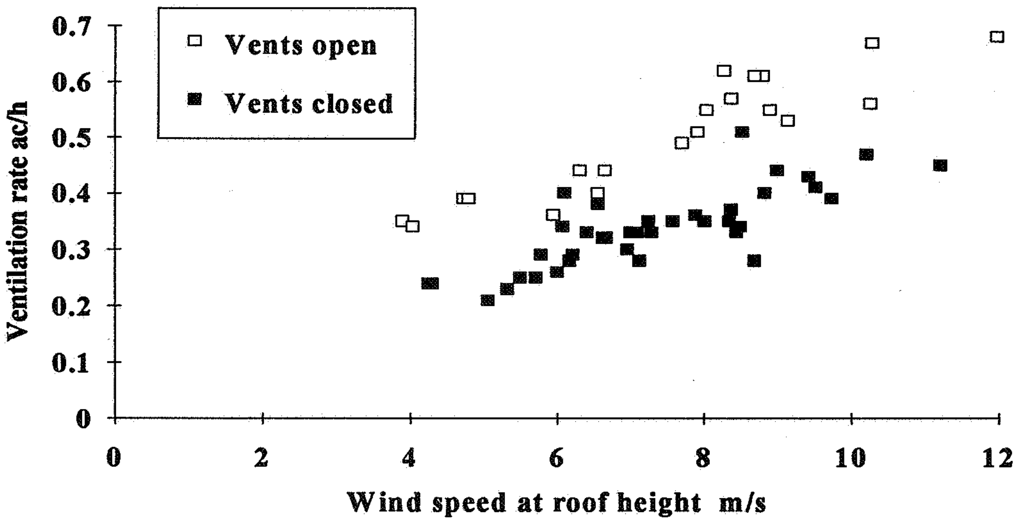

Figure 3: Infiltration in house C against wind speed at roof height with vents open and closed

2.4 Comparison of measured and calculated ventilator performance

Ventilation rates have been calculated for wind and temperature conditions recorded in houses A, B and C using the same numerical methods used in the numerical study [1]. Figures 4, 5 and 6 compare calculated results with measured ventilation rates for houses A, B and C, indicating that reasonable agreement has been achieved. In the case of house A the comparison is less clear because of frequent wind direction changes and therefore comparatively little data.

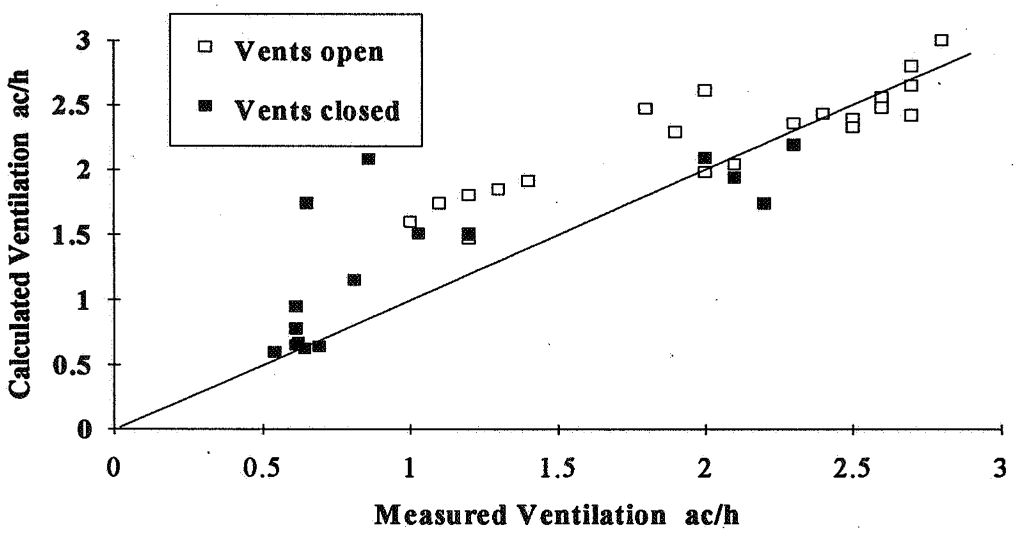

Figure 4: Comparison of measured and calculated infiltration in house A with passive vents open and closed.

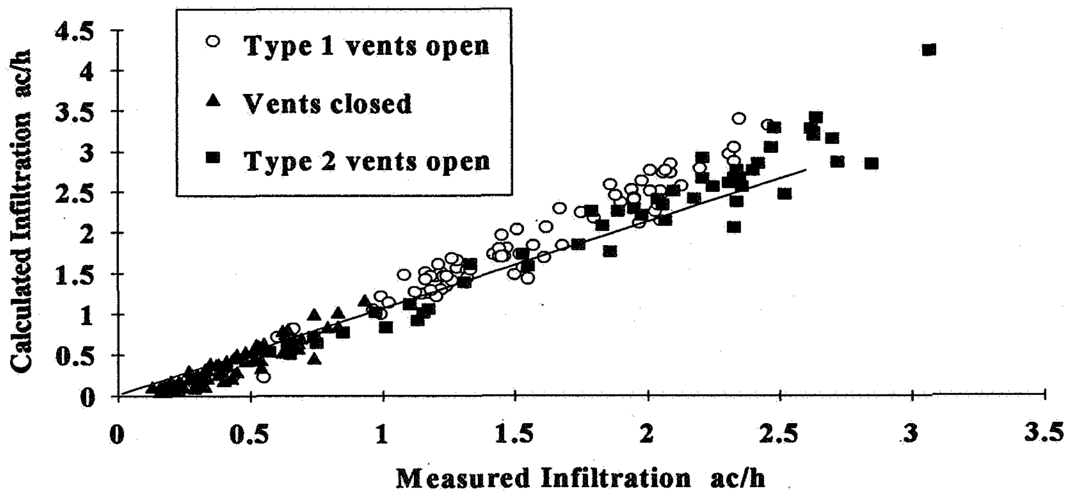

Figure 5: Comparison of measured and calculated infiltration in house B with two types of passive vents in open and closed states.

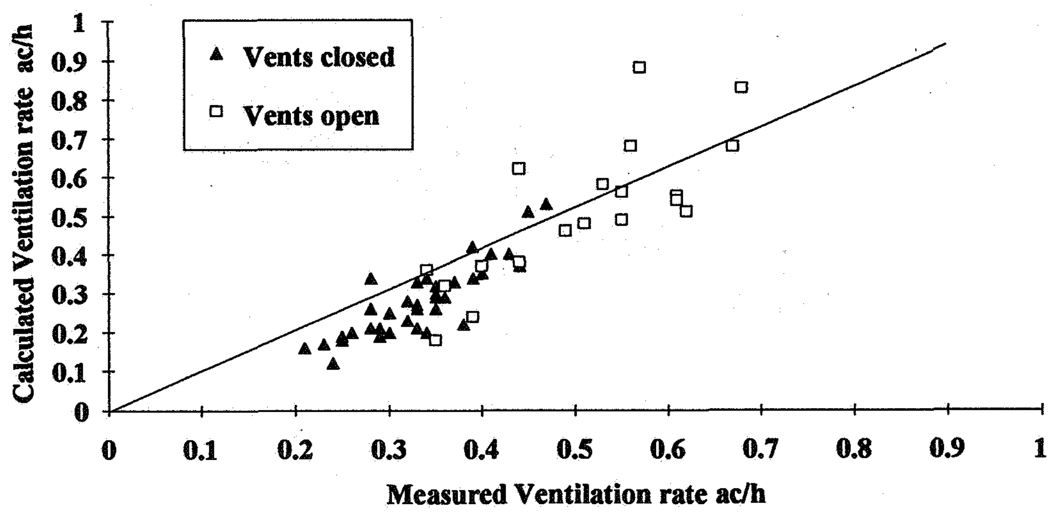

Figure 6: Comparison of measured and calculated infiltration in house C with passive vents open and closed

2.5 Discussion of ventilator performance

Measured ventilation rates were influenced by wind direction in each test house. While it was not practical to collect sufficient test data to define ventilator performance for all wind directions, there are sufficient data to define the ventilator performance for at least one wind direction. For houses B and C the measured ventilation delivered by the passive vents has agreed with the calculated ventilation rate with the normalised standard deviation of the residuals lying in the range 20-25%. For house A the agreement is less well resolved. Wind direction averaged ventilation rates corresponding to a wind speed at roof height of 3 m/s have been calculated for all cases. Figure 7 shows the relationship between ventilator size and average ventilation rate [1] overplotted here with the wind direction averaged ventilation rates for the experimental buildings. Houses B and C were sheltered by buildings of similar height and the ventilation rates are shown here to agree with those generated numerically using quite different building models. There is not the same level of experimental support for House A but the calculated wind direction averaged ventilation rate is included in Figure 7. This building was on largely undeveloped land at the time of the ventilation measurements and was modelled as exposed to wind. Figure 7 indicates the average ventilation rates to be expected from window-mounted ventilator systems for a range of wind exposure. Most houses in suburban areas have been found to be effectively sheltered, hence the darkened band indicating the most likely degree of exposure.

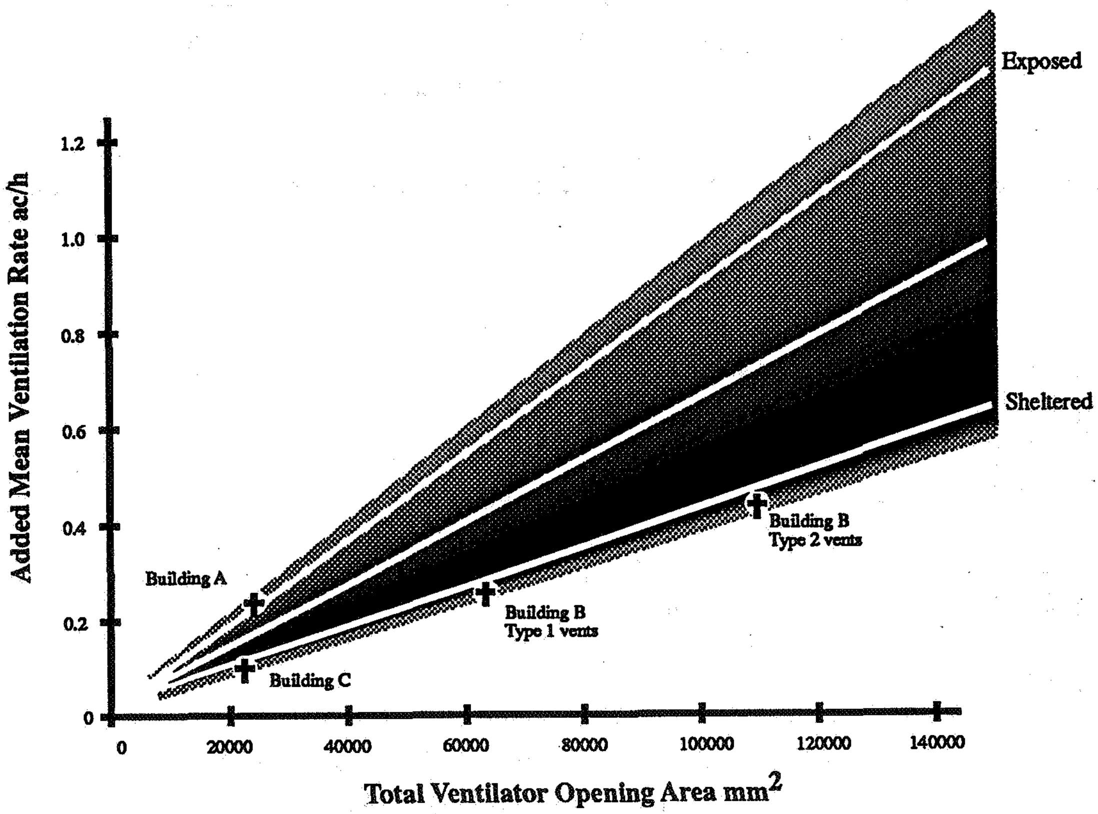

Figure 7: Average ventilation rates delivered by window ventilators as a function of open area - A comparison between measurement and theory.

It has been suggested [5] that 0.2-0.3 ac/h of secure and reliable ventilation should be added to current airtight house types in New Zealand to boost background ventilation levels closer to the 0.5 ac/h minimum recommended for indoor air quality. For a sheltered house (most houses located in suburban areas) this would require 40,000 - 60,000 mm² of passive ventilation opening. This is comparable with the British Building Regulations Approved Document F1 "Means of Ventilation" [6] which defines passive ventilator sizes to provide background ventilation in habitable rooms in the UK. In this regulation, trickle ventilators must have an open area of not less than 4,000 mm² in each room.

3. Conclusions - Passive ventilators for NZ homes

The air flows delivered by four passive ventilator systems mounted into the windows of three unoccupied houses have been measured using tracer gas methods. Comparison with numerical predictions has yielded the following results:

1 The experimental ventilators in the three test houses were sized between 21,000 and 110,000 mm^2 of opening area. In building airtightness tests this added between 18% and 130% to the individual house air leakage rate at 50 Pa.

2 The measured ventilation performance of the houses has compared favourably with ventilation rates calculated using site measured wind speeds and building airtightness characteristics. The accuracy of the total ventilation rate prediction is typical of that demonstrated by previous studies.

3 Air flow rates delivered by the passive vents in two houses have agreed with calculated rates to the same level of accuracy achieved for whole house infiltration rates thus offering direct support to the modelling approach taken in part one of this paper.

4 Air flows through the passive vents corresponding to a wind speed at roof height of 3 m/s have been compared with numerical data for a range of locations and building types in NZ. Support has been offered for a simple linear relationship between added ventilation and ventilator opening area.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Ventilation by the Windows in Classrooms: A Case Study

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SYNOPSIS

Four classrooms of two secondary schools located around Lyon in France have been monitored. The objectives are to analyse the quality of the indoor air and the thermal comfort and also the behaviour of the occupants towards opening of the windows.

This paper briefly describes the context and the nature of the monitoring campaign, and presents the results of the measurements with direct interpretation of the ventilation needs. Then, we try to make a statistical analysis of the influencing factors that lead to the opening of the windows, but our study is limited because of the small number of collected data.

Results from this study show that allowable CO2 levels are overpassed several times in a school day. The presence of a mechanical ventilation system leads to lower peaks but the fresh airflow is too small to prevent an indoor confining, that is also revealed by the aerobiological analysis. These measurements confirm a certain ill-being of the surveyed people, not in relation with thermal comfort. This feeling leads people to open windows provided that outdoor conditions are favourable (temperature, wind speed, noise, outside odours, ...).

1.0 CONTEXT OF THE STUDY

French regulation for schools does not impose that the buildings are equipped with a mechanical ventilation system except for some classrooms devoted to the physical sciences. Opening of the windows is supposed to be sufficient to insure the recommended hygienic airflows. To have a better knowledge of the practical use of the windows and the resulting indoor air quality inside classrooms, a first monitoring campaign has been undertaken by the Laboratory for the Building Sciences (ENTPE/LASH) in collaboration with a team from the technical network of the French Ministry of Public Works (CETE Lyon) and the laboratory of hygiene of the Lyon city.

The chosen sample of buildings is small for this first prospective study: two buildings, a first one with a mechanical ventilation system, a second one with natural ventilation by the windows. Two similar classrooms are monitored for each building on a weekly basis. This study could be extended to other buildings in the future.

1.1 Description of the monitoring

Measurements in the buildings include [1]:

- duration of the opening of each windows within a 2 minutes time step
- indoor temperature and relative humidity every 3 minutes (Vaisala)
- CO2 concentrations every 4 minutes in one classroom (Dräger Multiwarn IND equipment), or hourly averaged in second classroom (IR spectrometer Cosma Beryl100)

Ambient climate (wind, solar radiation, outdoor temperature) was recorded on the ENTPE site (between 5 and 10 km from the studied buildings).

These measurements were done during winter time : 10th-19th February 1993 for first school and 10th-24th March 1993 for second school. Unfortunately weather was particularly warm during the monitoring of the second school.

Aerobiological samples were taken several times during the monitoring. The biocollector is Joubert one, with 3 different media boxes: tryptone soja gelose, Baird-Parker, and Sabouraud [2].

1.2 Survey to the occupants

The measurements were coupled with a survey to the teachers of each school by questionnaires, and a few interviews to the occupants of the studied classrooms.

Questions were relative to the general well-being inside classrooms, with more directed questions about thermal comfort, air quality (or "feeling of suffocation"), building equipments knowledge and other factors that could influence the openings of the windows (noise, wind, sun...).

We try also to have an idea of the teachers' habits towards windows opening and closing.

The high return level of the questionnaires (50%) show the teachers' interest for their working environment.

2 RESULTS FROM THE MONITORING

Here are given some direct results about comfort level inside classrooms, that were got from the monitoring.

2.1 Air quality

The measured carbon monoxide concentration inside classrooms are very low (less than 4 ppm), without any risk for the occupants' health.

Conclusion is not the same for carbon dioxide; very high concentrations can be reached in both schools, with maximum of 7000 ppm in the school without any ventilation system (figure 1a). This level is much higher than the 800-1500 desirable values for indoor air quality [3]. In the building equipped with a mechanical ventilation system, the peaks are lower, with maximum of 3500 ppm after 3 hours of occupancy (figure 1b).

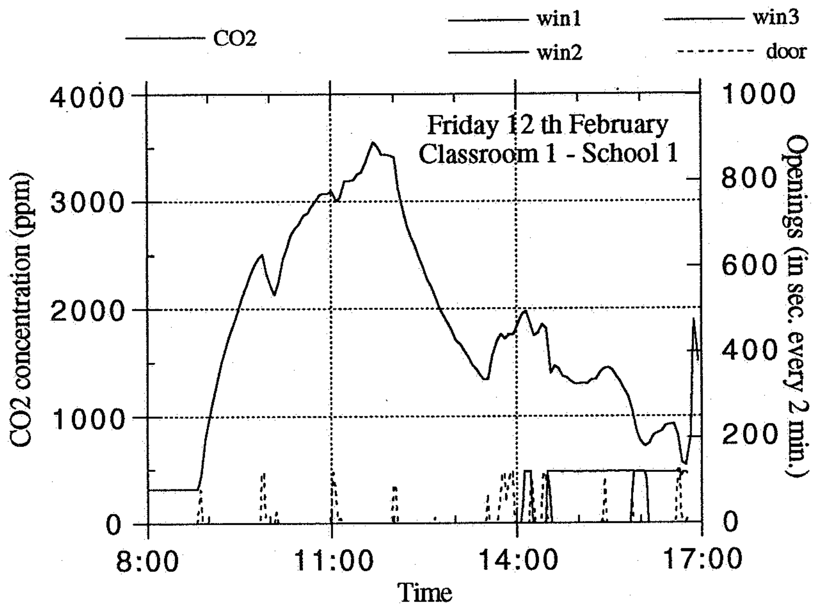
The CO2 level of 1500 ppm is overpassed during 66% of occupancy time for first school and 74% for second one.

The aerobiological analysis also reveals quite an important environmental bacteria and fungi flora load because of students' activities (table 1), but no pathogenic germ was found [2].

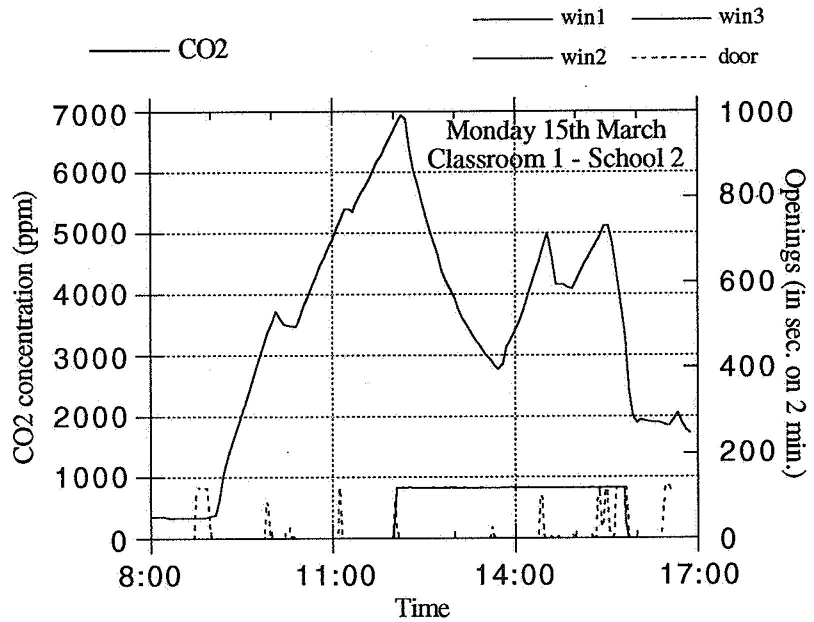
School	Bacteria		Staphylocoques		Fungi	
1	201	207	19	12	E(*)	27
2	192	118	23	29	32	34

(*) The counting was not possible because of Mucor colonies

Table 1 : Counting of the colony forming units (CFU) per 0.5 m3, after 6 days (2 samples each classroom during occupancy).



a)



b)

Figure 1 : CO2 measurement and openings duration during one class day

2.2 Thermal comfort

The couples (temperature, humidity) are in great majority inside the relative comfort zone ($-0.5 < PMV < 0.5$) defined for a clothes resistance of 1.5 clo and a metabolism of 1 Met [4]. Only a few periods get outside this zone during the occupancy periods (PPD>17% for temperature between 24 and 25°C). This result is valid for both schools during the monitoring.

2.3 Openings of the windows

Table 2 gives how many times one window has been opened during occupancy period. The daily duration is quite different between the two classrooms in the first school, according to different habits of the teachers .

School	Classroom	Average nb of openings /day	Average duration of opening	Average duration of 1 opening
1	1	2.5	65 minutes	26 minutes
	2	2.2	26 minutes	12 minutes
2	1	2.7	304 minutes	114 minutes
	2	2.3	243 minutes	108 minutes

Table 2 : Daily statistics about openings duration

Daily opening duration is important in the first school (up to 65 minutes) despite the ventilation system. It is much more large in the second school (up to 5 hours), as teachers used to opening the windows after one hour class and keep them open all morning long. Unfortunately, it is difficult to interpret these openings in terms of larger needs for fresh air because the weather was warmer during the second monitoring period, that could influence people.

2.4 Occupants' point of view

The number of returned questionnaires which is 18 for first school and 20 for second one is enough to allow a representative analysis. Table 3 gives the recorded answers (in percents) about thermal comfort and feeling of suffocation.

Table 3 : Comfort feeling from the teachers answers

% answers	School 1	School 2
Thermal comfort		
Neutral	89	85
Too hot	11	5
Too cold	0	10
No answer	0	0
Feeling of suffocation		
Yes	28	70
No	72	25

These figures confirm previous conclusions from the physical measurements during monitoring, that is a satisfactory temperature level during winter time for both schools. The feeling of suffocation in the second school (without mechanical ventilation system), that nearly does not exist in first school, can be linked to higher levels of relative humidity and CO2 concentration.

About windows openings, the answers do not allow to estimate properly the number and the duration of the openings. However, they show that some teachers never open the windows during winter in the first school when most of them at least open for a 20 minutes duration in second school.

Table 4 shows that the reasons for the openings are the same in both schools with 80% of the answers directed towards air quality. Reasons for closing the windows are also the same, to come back to a comfortable temperature and to dampen the outside noise.

Table 4 : Main reasons to open / close the windows

% answers	School 1	School 2
To get a comfortable temperature	17	17
To disperse bad smell	44	42
To have fresh air get in	36	41
To come back to a comfortable t°	37	25
To dampen outside noise	33	36
Because weather has changed	18	14

In the second school, 50% of the surveyed people know that the building has no mechanical ventilation system et 40% say that it is not acceptable. In the first school, 2 teachers say that the ventilation is not satisfatory and 9 teachers say it works well.

2.5 Summary

The measurements have shown that there are some problems of air quality in both schools (high CO2 concentrations, body odours...) which the occupants are sensitive to. The resulting "feeling of suffocation" in the building without ventilation system has no link with thermal comfort, that seems satisfactory.

3 VENTILATION EFFICIENCY

At the arrival of N students in a classroom, the CO2 concentration increases because of the metabolic production according to equation (1) :

$$V\frac{dc}{dt} = nV(c_o - c) + Np \qquad (1)$$

where
V is the volume of the room [m3]
$c(t)$ is the CO2 concentration [g/m3]
n is the air change per hour [vol/h]
co is the CO2 concentration of fresh air [g/m3]
p is the metabolic CO2 production [p = 32 g/h/pers.]

Carbon dioxide can be seen as a good tracer gas to measure air change per hour of the studied classrooms. If air change per hour is constant during times t_1 and t_2 n can be estimated with sampled values of $c(t)$ using a mathematical solver (we suppose that the volume is well mixed and the c measurement is representative):

$$c(t_2) = c(t_1)e^{-n(t_2-t_1)} + (1 - e^{-n(t_2-t_1)})(c_o + \frac{Np}{nV}) \qquad (2)$$

Outside concentration co is supposed to be equal to the concentration inside classroom at early morning, before arrival of the occupants; Values between 280 ppm and 340 ppm are found for first school and between 340 and 470 ppm for second school.

In the first school, the calculated air change per hour varies between 1.5 and 2.0 that is a volume of less than 10 m3/h per person. Following French regulation, it is required that 15 m3/h/pers of fresh air is supplied in classrooms. A simulation shows that the limit of 1500 ppm would not be exceeded with this rate.

In the second school, the calculated air change per hour when windows are closed is located between 0.3 and 0.6 that means that less than 3 m3/h of fresh air are supplied per person. Opening of the windows can induce air flows as large as 3 ACH during the monitored period. The hygienic air flow per person is then effectively provided.

4 INFLUENT FACTORS ON WINDOWS OPENINGS

One of the objectives of the study is to analyse efficiency of windows openings to maintain air quality inside classrooms. As the windows are not automatically moved, it is necessary to study the occupants' behaviour towards the windows. When, why, how do they open windows ?

4.1 Time for the openings

Teachers used to opening between two classes. 70% of the openings correspond with breaks, while closings can occur at any time This could mean that the teacher (which is most of the time the first person involved in the opening process) either is not bothered by discomfort feeling during his class, or his attention prevents him to be sensitive to it.

4.2 Effect of indoor climate

In order to better understand if there are specific reasons involving people to open the windows (or doors) of the classroom, the influence of various factors is analysed from the recorded measurements: CO2 concentration, indoor temperature, relative humidity

For each of the 68 recorded openings the corresponding values of these factors (when available) are gathered in a data base for statistical analysis. The following questions were looked at for each of this factor:

Is there a limit beyond that the opening is systematic ?
Is there a limit below that the opening is prohibited ?
Is the frequency of the openings dependent on the value of the factor ?

4.2.1 Influence of indoor temperature

Figure 2 plots the number of openings and closings in each temperature interval between 17 and 26°C, compared to the total number of measurements in the same interval when the room is occupied. There is a slight increase of the openings frequency when temperature is greater than 23°C, but there is no upper limit that leads people to systematically open one window. However, very few openings occur when temperature is below 21°C, while more closings happen.

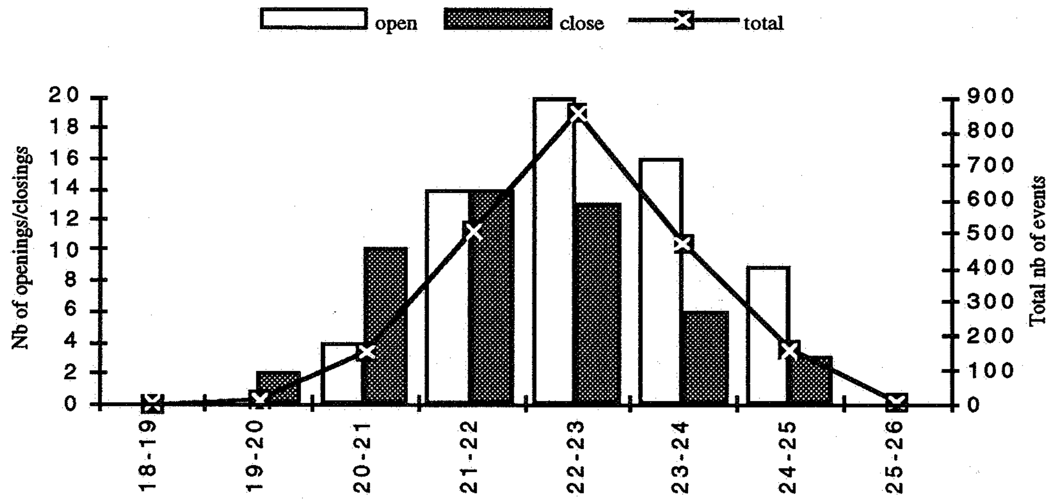


Figure 2 : Frequency of the openings/closings in a temperature range (°C).

4.2.2 Influence of CO2 concentration

Openings can occur at the early beginning of classes (figure 3) when CO2 level does not exceed 1000 ppm or may not occur for high levels (only one opening in the range 3500-7000 ppm which represents 15% of the recorded concentrations). 69 % of the openings occur in the range 1500-2500 ppm which is reached after 1 hour class.

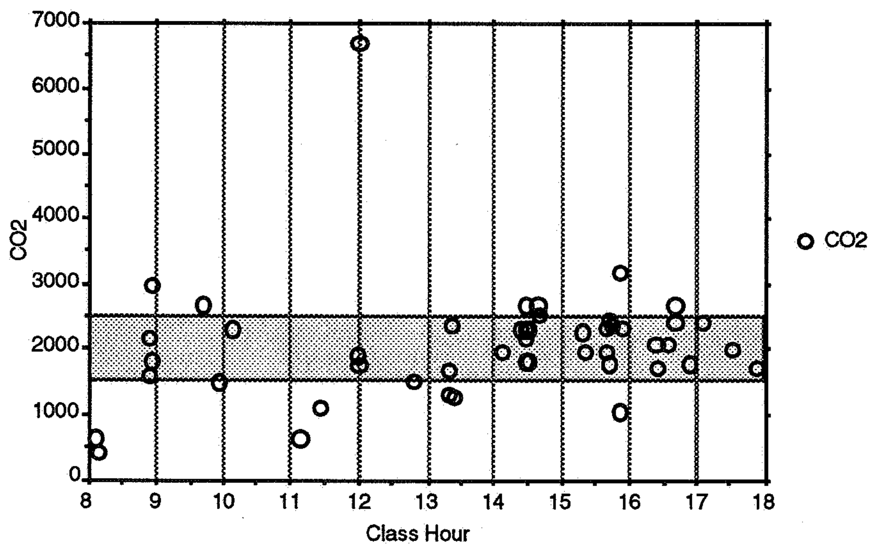


Figure 3 : CO2 concentration at the opening against time

4.2.3 Influence of relative humidity

Using the same approach, it is found that most of the openings occur when relative humidity is around 45-50% in first school or 50-55% in second school, but this factor is highly correlated to CO2 concentration because of human metabolism.

4.3 Conclusions

It is difficult to find specific factors to explain the openings. If temperature could play a significant role, showing that human body is quite sensitive to low values, it does not seem to be the case for air quality. There is no visible correlation between openings and CO2 concentrations, except that teachers open more often when CO2 level can reach 1500-2500 ppm. But this factor seems to be more sensitive at the class breaks, either when the teacher finishes his class or when he comes back to his classroom.

A lot of other factors are likely to influence windows openings and closings either from the indoor environment or the outdoor one (wind was found to be an influent factor on the closings in this study). Sociological and psychological factors [5,6] could also be analysed, but it was out of the scope of this study.

5 OVERVIEW

According to the objectives of the study, the pollution from the human presence inside classrooms was analysed thanks to some measurements and a survey to the occupants. It was demonstrated that problems exist in both studied schools. In the first building, insufficient fresh air flow was provided by the ventilation system, while in the second building, opening of the windows cannot be seen a reliable mean to prevent pollution by the human metabolism. Indeed, no direct correlation could be found between high CO2 concentration levels and frequency of the openings. The recorded openings seemed more linked to the school pace.

In the future, it seems interesting to develop this approach to a larger set of schools in order to assess effective ventilation air changes by the systems and to improve comfort inside classrooms.

ACKNOWLEDGEMENTS

The authors would like to thank ADEME Rhone Alpes for their support to this study.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
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Single-sided Ventilation: A Comparison of the Measured Air Change Rates with Tracer Gas and with the Heat Balance Approach

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Single Sided Ventilation: A Comparison of Measured Air Change Rates with Tracer Gas and with the Heat Balance Approach

Synopsis

In the frame of the European PASCOOL project, several experiments regarding single sided ventilation were carried out at BBRI in the outdoor PASSYS test cell. The test room of 30 m3 has a vertical window of about 1 m². During a first measurement period, an open cold box, which allows one to control the vertical wind speed, was placed in front of this window. During a second measurement period, the window was directly exposed to "real wind". The air change rates were evaluated by using two different methods: a tracer gas technique and the heat balance approach. The heat balance approach is very attractive in this test cell because the heat flow through the cell envelope can be accurately determined thanks to the Pseudo-Adiabatic-Shell. The tracer gas measurement is made difficult because a clear air flow pattern appears and accordingly, the concentration in the room is not homogenous. An error analysis has been applied on both methods. The agreement between both methods is very good and the heat balance approach proved to be more accurate than the tracer gas technique. A correlation model was derived from the first measurement period.

List of symbols

m_{SF6} = mass of SF6 contained in the test room (mg)
S_{SF6} = SF6 injection rate (mg/s)
Q_{in} = air flow rate leaving the test room (m3/s)
Q_{out} = air flow rate entering the test room (m3/s)
C^{in}_{SF6} = SF6 concentration of the air flow entering the test room (mg/m3)
C^{out}_{SF6} = SF6 concentration of the air flow leaving the test room (mg/m3)
T_{in} = temperature of the air entering the test room (K)
T_{out} = temperature of the air leaving the test room (K)
q_{sun} = global vertical solar radiation through the opening (W)
Q = heat contained in the air and in the materials present in the test room (J)
q_{PAS} = heat flow entering the test room through the PAS (W)
$q_{Heating}$ = heating power provided by electrical convectors in the test room (W)
q_{wall} = heat flow entering the test room through the reference wall (W)
$q_{ventilation}$ = heat flow leaving the test room due to ventilation (W)
ρ_{out} = density of the air leaving the cell (kg/m3)
c_p = specific heat of the air (J/kg)
ΔT = mean temperature in the test room minus outside temperature (K)
$V_{coldbox}$ = air velocity in the cold box
$Q_{thermal}$ = air flow rate through the large opening due to ΔT
Cd = coefficient of discharge
W = width of the opening (m)
H = height of the opening (m)
g = 9.81 (m/s2)
$\overline{T}$ = mean temperature of the air flows in the opening

1. INTRODUCTION

The estimation of air change rates in the case of single sided ventilation received the last years attention in several research projects. This paper presents experiments carried out in one of the outdoor PASSYS test cells on the BBRI site. Two separate approaches were used to evaluate the air change rates in a continuous way: on the one hand, tracer gas measurements, on the other hand, the heat balance of the test room. The objective of this experiment was to obtain information on the air flow through large openings. Both measurement approaches were compared and an estimation of the confidence interval integrated to the analysis. An attempt was made to correlate the measured air flow rates with the wind velocity and the temperature difference between inside and outside. This kind of model could be very useful for air flow simulation tools since, at the present time, they do not take into account the effect of the wind. In this paper we mainly focus on the comparison between the tracer gas measurements and the heat balance approach. After a description of the experimental set-up both methods are explained in detail, major sources of uncertainty, inherent in each method, are discussed. Results are then given and interpreted.

2. EXPERIMENTAL SET-UP

The outdoor PASSYS test cell is represented on figure 1. The south component is exchangeable. A "reference wall" was used during this experiment. It is made up of three layers: wood (12mm), PS30 (100mm) and wood (12mm) and has a vertical window of about 1 m^2.

The PASSYS test cell has its own instrumentation (for the measurement of air and surface temperature, solar irradiance, wind velocity and direction, heat fluxes, heating power,...). It is fully described in reference 1. In addition to those basic measurements, a specific instrumentation was employed for the needs of our experiments. It will be described hereafter in the sections devoted to the measurement methods.

During a first measurement period an open cold box, which allows one to control the vertical wind velocity, was placed in front of this window in order to obtain a kind of reference experiment for which the wind conditions are more or less controlled. The wind velocity in the cold box varied between 1 and 2 m/s. The opening was directly exposed to "real wind" during a second measurement period. During both periods, the test room was heated by electrical convectors.

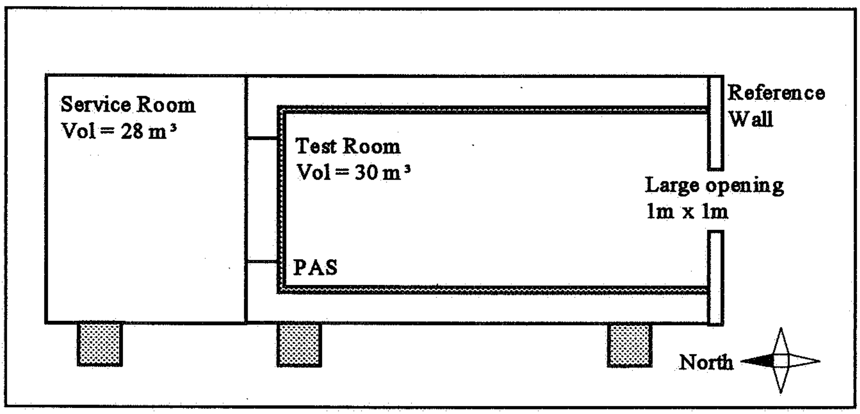

Figure 1: The PASSYS test cell - sectional view

3. Tracer gas method

3.1 Introduction

The single sided ventilation is driven by two "motors": the wind and the natural convection. In our experiments, the natural convection is mainly due to temperature difference between inside and outside.

The major difficulty encountered for measuring air flow rates is that a clear air flow pattern appears in the room. As the test room was continuously heated, the outside temperature is lower than the inside temperature. That implies that the cold air comes from the outside through the lower part of the window and runs down on the floor. Accordingly, the tracer gas concentration in the room is not at all homogenous. This makes it impossible to use classical tracer gas techniques for which a "perfect mixing" is required. When the effect of the wind is dominant such techniques could however be used because no clear air flow pattern appears and the concentration in the room can reasonably be considered as homogenous.

3.2 Specific set-up

We employed the Brüel&Kjaer tracer gas equipment (types 1302, 1303 and 7620). Sulphur hexafluoride was used as tracer gas. Eight injection points were placed in the test room. They were all connected to the same nozzle by tubes of the same length. Doing this, it was expected that the injection rate would be the same at each point. But it emerges from the analysis of the concentration measurements that sulphur hexafluoride was only injected at lower points. This is probably due to the static pressure resulting of the gas column in the climbing part of the tubes going to the higher points. Figure 2 shows the measurement locations as well as the injection locations. In order to eliminate inaccuracies due to very local fluctuations in gas concentrations, we used 4 sampling points connected together at each measurement location. They are distributed in a small zone of typically 20x20x20 cm3. A temperature sensor was placed at each sampling zone. The time step between two concentration measurements is about 15 minutes. Other measurements are scanned every minutes.

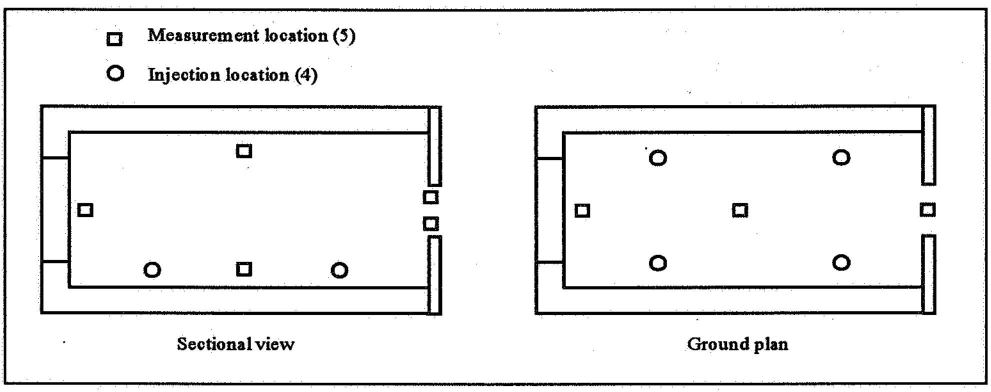

Figure 2: Measurement and injection locations

3.3 Equations

The mass balance equations of the room for the tracer gas and the air are:

$$\frac{dm_{SF6}}{dt} = S_{SF6} + Q_{in}.C_{SF6}^{in} - Q_{out}.C_{SF6}^{out} \qquad [mg/s] \qquad (1)$$

$$\frac{dm}{dt} = Q_{in}\cdot\rho_{in} - Q_{out}\cdot\rho_{out} \qquad [kg/s] \qquad (2)$$

Assuming that the mass of air in the cell remains constant, equations 1 and 2 yield:

$$\frac{dm_{SF6}}{dt} = S_{SF6} + Q_{out}.(C_{SF6}^{in}\cdot\frac{T_{in}}{T_{out}} - C_{SF6}^{out}) \qquad [mg/s] \qquad (3)$$

$$\frac{Q_{in}}{T_{in}} = \frac{Q_{out}}{T_{out}} \qquad [m3/s.K] \qquad (4)$$

We can derive the air flow rates (Q_{out}, Q_{in}) from equations 3, 4. The temperature difference between inside and outside is of about 5 Kelvin and accordingly, Q_{in} and Q_{out} do not differ from more than 2%.

3.4 EVALUATION OF THE DIFFERENT TERMS

The next paragraphs explain how the terms of equation 3 are derived from the measurements.

During all the experiment the temperature in the test room was kept higher than the outside temperature. This implies that the air flows in the large opening have always kept the same direction: the lower flow was entering the room and the upper flow was leaving the room. Therefore, the concentration of the air leaving/entering the room is measured at the sampling zone located in the upper/lower part of the opening. Doing that, we implicitly assume that the SF6 concentration of the air leaving/entering the test room is constant in the upper/lower part of the opening. As a matter of fact, the reality is different since the concentration in the room is not homogeneous. The value that we should use is an average (weighted by the air velocity) of the SF6 concentrations in the upper/lower part of the opening and we measure an average of SF6 concentrations at 4 points distributed in the sampling zone of the upper/lower part of the opening. Accordingly, a large uncertainty will be taken on those concentrations for the confidence band calculation.

The SF6 injection rate is measured by the B&K tracer gas system.

The mass of SF6 contained in the test room is obtained from the average of the concentrations measured at the 5 different sampling zone. It is a quite rough approximation but a sensitivity analysis showed that it does not affect strongly the calculated air flow rate.

3.5 ASSUMPTIONS REGARDING MEASUREMENT UNCERTAINTIES

The following table shows errors on the different measured values due to the measurement equipment (thermocouples, gas analyser,..). They are given by the manufacturers.

Amount of gas injected	2%
Temperature	0.2 K
Concentrations	2.5%

Table 1: Tracer gas method - measurement errors

Besides this first uncertainty, we have to take into account the fact that the values we measure are not the values we use in equations 3 and 4. The representativity of the different terms derived from the measurements must be estimated. The next table shows the assumptions chosen for errors of representativity.

m_{SF6}	Mass of SF6 in the room	40%
C^{in}_{SF6}, C^{out}_{SF6}	Concentrations in the opening	20%

Table 2: Tracer gas method - representativity errors

4. HEAT BALANCE APPROACH

4.1 INTRODUCTION

The energy balance equation of the test room allows one to calculate the losses or gains due to the ventilation. Knowing the temperatures of the air entering and leaving the room, the air flow rate can be derived.

4.2 SPECIFIC SET-UP

It was possible to use the heat balance approach because the PASSYS test cell in which the experiments were carried out is equipped with the Pseudo-Adiabatic-Shell (PAS). The PAS allows to evaluate the heat flux leaving the cell through all the walls except the south one. It consists of a second shell (130 mm) which can be heated at its outside surface by heating foils and which is added inside the cell envelope. The position of the PAS in the test cell can be seen on the figure 1. The heating foils are controlled (on/off regulation) in order to maintain the temperature difference between inner and outer surfaces of the PAS as small as possible. Doing this, the losses through the envelope are reduced to a minimum.

4.3 EQUATIONS

The heat balance equation of the test room is:

$$\frac{dQ}{dt} = q_{PAS} + q_{Heating} + q_{wall} - q_{ventilation} + q_{sun} \qquad [W] \qquad (5)$$

The heat flow is related to the air flow as:

$$q_{ventilation} = \rho_{out} . c_p . Q_{out} . (T_{out} - T_{in}) \qquad [W] \qquad (6)$$

Equation 6 yields Q_{out} and Q_{in} can be calculated from equation 4.

4.4 EVALUATION OF THE DIFFERENT TERMS

The heat flow through the cell envelope is derived from the external and internal average surface temperatures of the PAS. The RC-scheme modelling the PAS was identified with MRQT software on calibration experiment data.

The electrical heating power is directly measured (PASSYS instrumentation).

The heat flux through the reference wall was measured at two different places by fluxmeters. The total heat flow is estimated as the average of both measurements multiplied by the surface of the wall.

The variation of the energy contained in the air and in the materials present in the room is evaluated from the variation of the average air temperature of the room (7 measurements).

The materials are assumed to have the same temperature than the air. The sensitivity analysis shows that this approximation does not put a large uncertainty on the air flow rate.

The solar irradiance is directly measured.

Two thermocouples were placed in the large opening at the centre of the tracer gas sampling zones shown on the figure 2 in order to measure the temperatures of the air entering or leaving the room. The remark given for concentration measurements in the opening is still valid: the temperature of the air entering/leaving the cell is not constant. This is taken into account in the error calculation.

4.5 ASSUMPTIONS REGARDING MEASUREMENT UNCERTAINTIES

As for the tracer gas measurements, the difference is made, in the following tables, between measurement and representativity errors.

Electrical heating power	1%
Temperature	0.2 K
Heat flux	5%

Table 3: Heat balance approach - measurement errors

q_{PAS}	Heat flow through the PAS	20%
q_{WALL}	Heat flow through the reference wall	20%
T_{in}, T_{out}	Temperatures in the opening	10%
	Mean temperature of the test room	0.5 K

Table 4: Heat balance approach - representativity errors

The error chosen on the temperatures (heat balance approach) in the opening is smaller than the one chosen on the concentrations (tracer gas approach) in the opening. Indeed, the temperatures were measured every minutes and then averaged whereas the concentrations were measured every 15 minutes. Moreover, the temperature is more homogenous than the tracer gas concentration in the room.

5. EXPERIMENTAL RESULTS

5.1 EXPERIMENT WITH COLD BOX

5.1.1 AIR FLOW RATES AND CONFIDENCE BANDS

Figure 3 compares results obtained from both methods. The 95% confidence bands are represented.

As one can seen, the agreement between both method is very good, the confidence band overlaps during the whole measurement period. The heat balance approach appears to give a less fluctuating and more accurate air flow rate than the tracer gas measurements. A sensitivity analysis has shown that the main source of error for both methods is the uncertainty on the measurements taken in the large opening (temperature or concentration). Therefore, the confidence interval is mainly defined by the error of representativity: 10% on the temperatures in the opening and 20% on the concentrations in the openings.

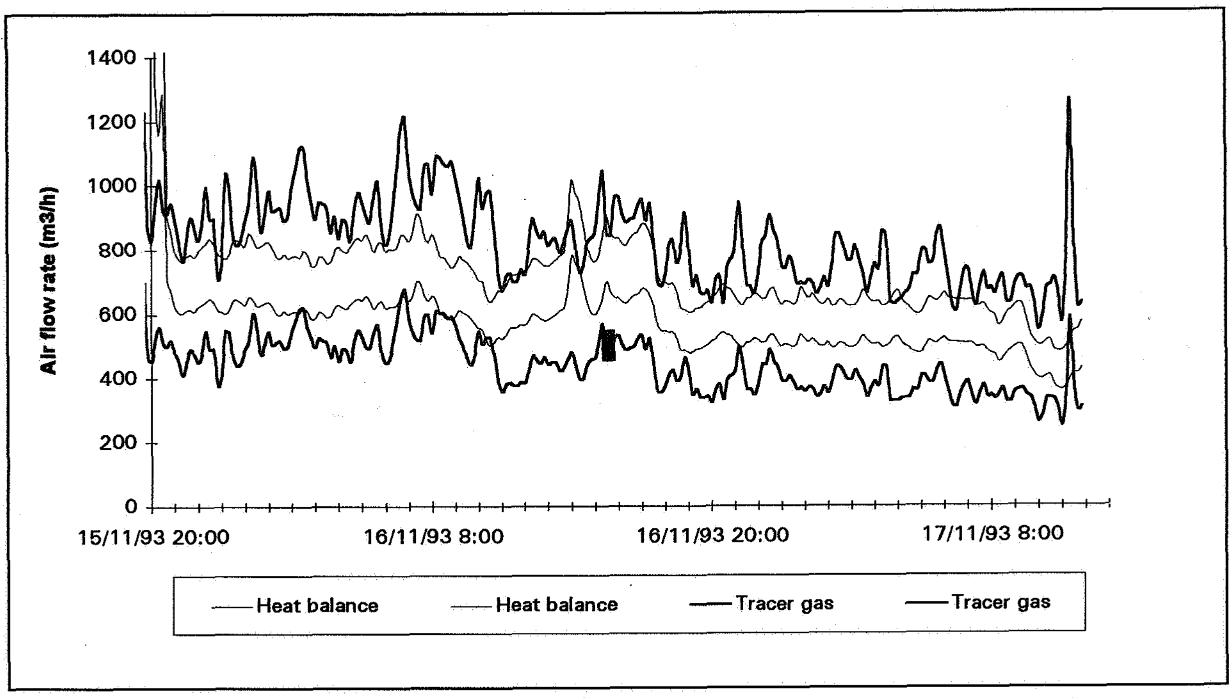

Figure 3: Period with cold box - Comparison of the air flow rates calculated from tracer gas and heat balance methods

5.1.2 CORRELATION MODEL

The results from the heat balance approach being more accurate, they were used to derive empirical model. The following correlation model was found:

$$Q_{out} = (153 \pm 8).\sqrt{\Delta T} + (195 \pm 6).V_{coldbox} \qquad \text{[m3/h]} \qquad (7)$$

The correlation coefficient is 0.78.

Using the classical simplified equation for the gravitational flow, we can give an evaluation of the coefficient of discharge.

Theory gives (see reference 2) :

$$Q_{thermal} = \frac{1}{3} \cdot Cd \cdot W \cdot H^{\frac{3}{2}} \cdot \sqrt{\frac{g}{T}\Delta T} \qquad \text{[m3/s]} \qquad (8)$$

Equation (8) gives in the studied case :

$$Q_{thermal} = Cd \cdot 257 \cdot \sqrt{\Delta T} \qquad \text{[m3/h]} \qquad (9)$$

Comparing the thermal part of the correlation model (7) and equation (9) yields:

$$Cd = 0.60 \pm 0.03$$

The confidence interval given comes from the statistical analysis performed. It gives an indication of how the model represents the measurements, but it does not take into account the measurement error. Since the error made on the air flow rate calculated by the heat balance approach is on average 12 %, in first approximation the error made on the discharge coefficient is of the same order.

5.2 EXPERIMENT WITHOUT COLD BOX

5.2.1 AIR FLOW RATES AND CONFIDENCE BANDS

Figure 3 compares results obtained from both methods. The 95% confidence bands are represented.

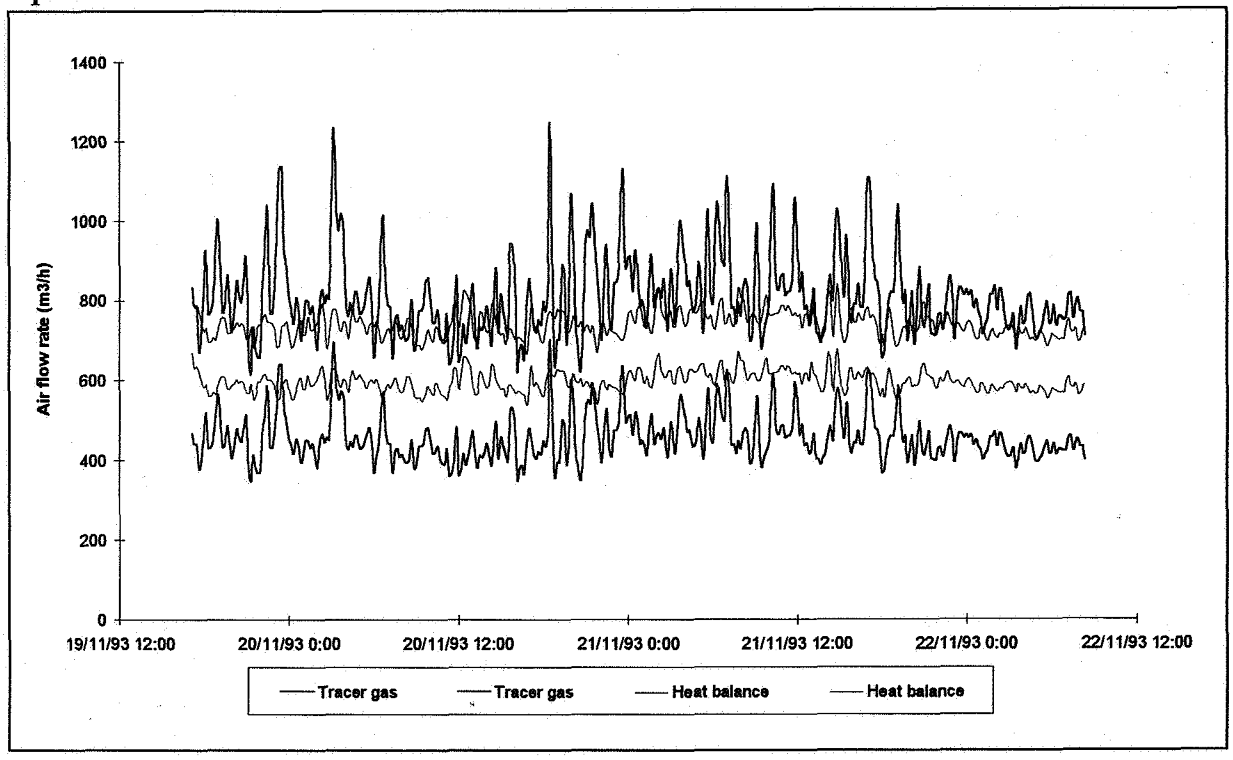

Figure 4: Period without cold box -Comparison of the air flow rates calculated from tracer gas and heat balance methods

The same remarks than for the first experiment can be made. The air flow rate obtained from the tracer gas measurements fluctuate even more. This was expected since the "artificial wind" provided by the cold box is obviously more stable (direction and speed) than the "real wind" to which the large opening was exposed during this second experiment.

5.2.2 CORRELATION MODEL

The temperature difference and the wind conditions, which are the parameters of the model, did not vary enough to derive a satisfactory empirical model.

5.3 IMPROVEMENT OF THE SET-UP

BBRI will set-up new experiment of the same type in a near future. The points of attention will be to put more measurement points in the opening and to make the temperature difference between the inside and the outside vary widely.

6. CONCLUSIONS

6.1 MEASUREMENT METHODS

Two different methods were used to analyse the single sided ventilation experiments: the heat balance approach and the tracer gas approach. Both methods have shown a good agreement.

The heat balance approach proved to be more accurate. For both methods the evaluation of the air flow rate through the large opening requires the measurement of physical characteristics of the air leaving the cell and of the air entering the cell. This is either the temperature (heat balance approach) or the tracer gas concentration (tracer gas approach). These measurements are achieved by placing devices in the opening. Therefore we have to know in which part of the opening the air will leave the cell and in which part of the opening the air will enter the cell. This is only possible for experiments for which the thermal part is clearly dominant. Indeed, in this case, there is only one neutral plane in the middle of the opening. If the wind effects are dominant, the direction of the air flow in the opening can change rapidly and several neutral planes can appear. In such a situation it is impossible to measure the physical characteristics of the air entering and the air leaving the cell. The heat balance approach allows however to calculate the heat flow through the large opening in every cases. A tracer gas measurement is still possible if the assumption of perfect mixing is fulfilled. That will be the case if the wind are largely dominant. If the thermal effect is not negligible, an air flow pattern will appear in the room and the gas concentration will not be homogenous. For the cold box experiment, the thermal effect is not dominant but the wind is stable and does not affect the positions of the air flows in the opening. In the second experiment, the thermal effect is largely dominant. The accuracy of both method can probably be improved by a better set-up.

6.2 Results

The accuracy of the analysis by tracer gas is about 30 %. The accuracy of the analysis by heat balance is about 12 %. The analysis of the cold box experiment allows us to propose a model for a wind parallel to the plane of the opening. The analysis of the second experiment does not give a satisfactory model. This is mainly due to the fact that the temperature difference and the wind velocity do not vary enough. The results are however valuable. They should be used with other results from other team to derive a correlation model.

Acknowledgements

This work is done as part of the EC PASCOOL project (Passive cooling for buildings). The authors would like to thank the sponsoring organisation.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Natural Ventilation Through a Single Opening - The Effects of Headwind

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SYNOPSIS

The airflow between a warm room and cool exterior can be significantly affected by an external headwind. Pollutant concentrations within the space depend on the relative sizes of the wind and the undisturbed stack driven flow. Two scenarios are described.

Firstly, a space is filled initially with buoyant polluted air. The space is then naturally ventilated through a single opening. In the "no wind" case, a gravity current of external air flows into the space. All the polluted air is expelled from the room. At high wind speeds the turbulence associated with the headwind produces mixing just inside the doorway. Under some conditions, ventilation levels are reduced. The second scenario considered is the natural ventilation of a space containing a continuous source of buoyant pollutant. For weak headwinds, fresh external air flows into the room and the pollutant concentration in that lower layer remains close to zero. High headwind speeds again generate doorway mixing. Air flowing into the space becomes contaminated with pollutant

These flows were studied experimentally using small-scale saline modelling techniques. Simple mathematical models are presented which agree closely with the experimental results. In both the transient and continuous cases, an increase in the headwind could lead to reduction in ventilation and an increase in internal pollutant levels. Natural ventilation through a single opening is not necessarily enhanced by wind.

LIST OF SYMBOLS

B	Buoyancy flux per unit width in room (m^3s^{-3})
c_1	Pollutant concentration in upper layer
c_2	Pollutant concentration in lower layer
D	Height of doorway (and room)
e	Entrainment constant for continuous flows
E	Entrainment constant for transient flows at high Fr
Fr	Froude number
g	Gravitational acceleration
g'	Reduced gravitational acceleration
k	Constant of proportionality in equation for velocity of gravity current
K	Scaled volume flux per unit width through doorway
$Pé$	Péclet number
Q	Volume flux per unit width down the room at high Fr
Q_1	Volume flux per unit width carried by gravity current
Q_e	Volume flux per unit width entrained into buoyant plume
Q_s	Source volume flux per unit width
Q_0	Volume flux per unit width across the doorway at high Fr
Q_{gc}	Volume flux carried by theoretical half-height gravity current
Re	Reynolds number
u_1	Gravity current velocity

u_{gc}	Gravity current velocity at $Fr = 0$
U	Headwind speed
$\delta\rho$	Density difference between interior and mixed region in high Fr model
$\Delta\rho$	Density difference between interior and exterior
ϕ	Scaled density difference
κ	Diffusivity
λ	Fractional height of room occupied by gravity current
ν	Kinematic viscosity
ρ	Ambient density
σ	Scaled volume flux per unit width down the room

1. Introduction

Natural ventilation is increasingly been seen as a viable option to air-conditioning for the removal of internally generated pollutants and heat. It is generally assumed that satisfactory ventilation air flows will be achieved by means of cross ventilation driven by the combination wind and buoyancy forces. One central feature of such designs is that the client is encouraged to avoid closed perimeter offices [1]. There are however many circumstances when cellular offices are desirable, privacy and status for example. In general such office spaces will be relatively shallow and experience suggests that there is little need to worry about internal environmental conditions with single sided ventilation.

There is however always the desire to stretch things to the limit so the question of how deep can a space be before it is essential to use cross ventilation arises. One experimental study [2] suggests that acceptable ventilation can be achieved within a 10m space. Internal gains in that case were not typical of modern commercial office spaces and further it is difficult to generalise the results of a single experiment. In particular it is difficult to isolate the relative effects of the two driving forces, wind and buoyancy. This paper presents an experimental and theoretical investigation with particular emphasis on the interaction of these two driving forces.

Airflows caused by temperature differences either side of an opening have been studied extensively in recent years. Early experimental work on stack driven flows is described in [3 - 6]. The incoming air flows through the doorway takes the form of a gravity current flowing into the room. The dynamics of gravity currents are well known and are described in [7].

A number of factors may influence such stack driven flows. One of the most significant of these is external wind. Air speeds through a typical doorway due to a temperature difference of 5°C are around $0.4 ms^{-1}$. Wind speeds are often of this magnitude or greater. Therefore, it might be expected that the effects of wind on buoyancy driven flow are considerable.

The different flows caused by different headwinds can be categorized in terms of an external Froude number. This Froude number is the ratio of external headwind velocity to a typical velocity produced by buoyancy alone. In the transient and continuous cases the Froude number is defined by

$$Fr = \frac{U}{\left(g\frac{\Delta\rho}{\rho}D\right)^{\frac{1}{2}}}, \quad \text{and} \quad Fr = \frac{U}{2\left(\frac{B}{2}\right)^{\frac{1}{3}}} \tag{1}$$

respectively, where U is the headwind speed, B is the buoyancy flux per unit width, g is gravitational acceleration, $\Delta\rho/\rho$ is the fractional density difference and D is the height of the room. Note that zero headwind has $Fr = 0$. Buoyancy effects become negligible in the limit $Fr \to \infty$.

All experimental work was done using the saline modelling technique, the paper first presents the basis of that method and then details of the experiments followed by the development of a simple mathematical model.

2. Experimental Study

2.1 Modelling Techniques

2.1.1 Dynamical Similarity

Water was used as the working fluid for the experiments. Density differences (corresponding to temperature differences) were produced by adding of brine. The validity of representing full-scale airflows by small-scale water flows requires that the ratios of the important forcing terms in the equations of motion are the same at small-scale as at full-scale. This dynamical similarity is equivalent to matching the important dimensionless parameters of the flows. For buoyancy driven airflows the important dimensionless numbers are the Reynolds number, *Re*, and the Péclet number, *Pé*. These are defined by

$$Re = \frac{UD}{\upsilon} \quad \text{and} \quad Pé = \frac{UD}{\kappa}. \tag{2}$$

For ventilation airflows, the Reynolds and Péclet numbers are both high and the flows are turbulent. At such high values, it would be expected that the dependence of the flows on the values of *Re* and *Pé* is not great.

2.1.2 Experimental Techniques

The experiments were carried out using a clear perspex room with rectangular cross-section. This modelled a section of a deep office with depth to height aspect ratio equal to five. A removable cover was placed on the upwind end of the room. For the transient experiments, the room was filled initially with dyed buoyant fluid. For all of the experiments, brine was the source fluid and was therefore more dense than the ambient environment. By viewing the experiment upside-down, the effect of buoyant fluid rising was achieved. This inversion was valid since density differences were small. The experiment was started by uncovering the end of the room. The flow modelled was that of single-sided ventilation of a deep office. This geometry was chosen for simplicity, with the flows in the interior being close to two-dimensional.

For the continuous experiments, a constant supply of dyed buoyant fluid was pumped into the space next to the closed end. The buoyant fluid was supplied through a line source occupying the full width of the room. The source was placed at the top of the room pointing downwards so that a downward flowing two-dimensional plume of dense fluid was produced. At full-scale, the source of buoyancy may be due to a heat source and no volume flux would be supplied to the room. For these experiments the volume flux was kept as low as possible (typically ten times less than the flux carried towards and away from the source).

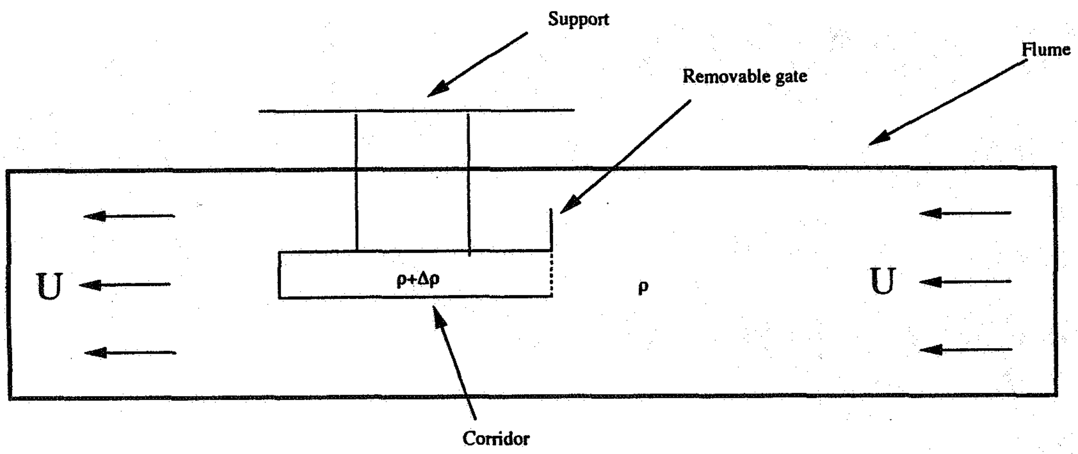

Figure 1. Experimental apparatus

To simulate an external headwind, the room was suspended in a much larger flume tank. The flume produced a time-independent uniform flow along its length and modelled an oncoming headwind. The experimental apparatus was as shown in figure 1. Video recordings were made of the experiments to enable measurements of velocities and other flow properties. Thermal exchanges with the walls of the room were neglected. This is a reasonable assumption provided that heat flux from the boundaries was much less than the heat flux associated with any buoyancy driven flow.

2.1.3. Experimental Parameter Ranges

The transient emptying of buoyant fluid from the room was studied for Froude numbers ranging from 0 to 11.4. The case $Fr = \infty$ was also investigated. For these experiments, the values of Reynolds number and Péclet number were approximately 5000 and 10^6 respectively. For the experiments with a steady source of buoyancy, Froude numbers lay between 0 and

7.7. Values of Reynolds number and Péclet number were similar to those for the transient experiments.

2.2 Transient Flows

2.2.1 Qualitative results

For flows driven by buoyancy only (*i.e.* $Fr = 0$) the observed flow was a gravity current occupying close to half the height of the room as shown in figure 2. The interface between inflow and outflow was situated approximately halfway up the room. The gravity current travelled at constant speed along the room and was reflected at the end. When the reflected gravity current had returned to the doorway, the interface ceased to be at half-height and the flow through the doorway was no longer quasi-steady.

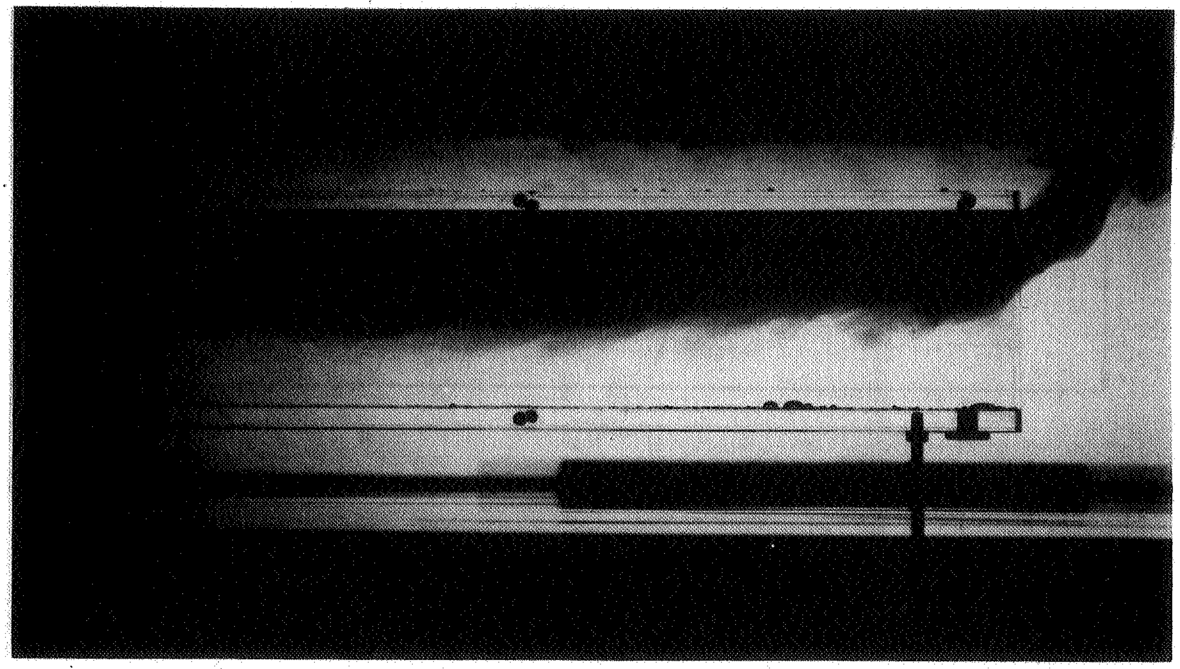

Figure 2. Gravity current at $Fr = 0$

For larger values of Froude number the interface between incoming and outgoing fluid was raised at the doorway as shown in figure 3. The different pressure distribution at the doorway due to the headwind affected the interface height. Within the room, the interface adjusted by sloping downwards and a gravity current flowed into the interior. At low Froude number, little mixing occurred between the two layers. Interfacial mixing near the doorway increased with Froude number. This increase appeared to be due to an increase in shear between the counterflowing layers.

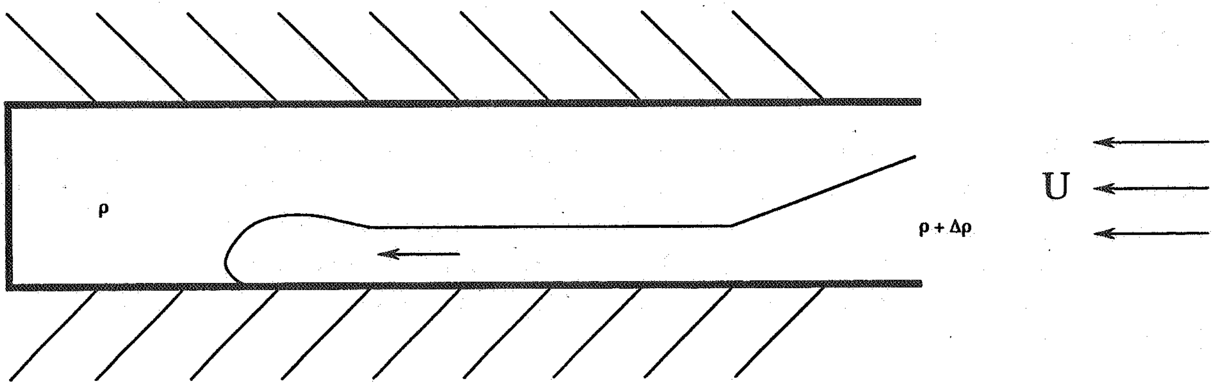

Figure 3. Raised doorway interface at intermediate *Fr*

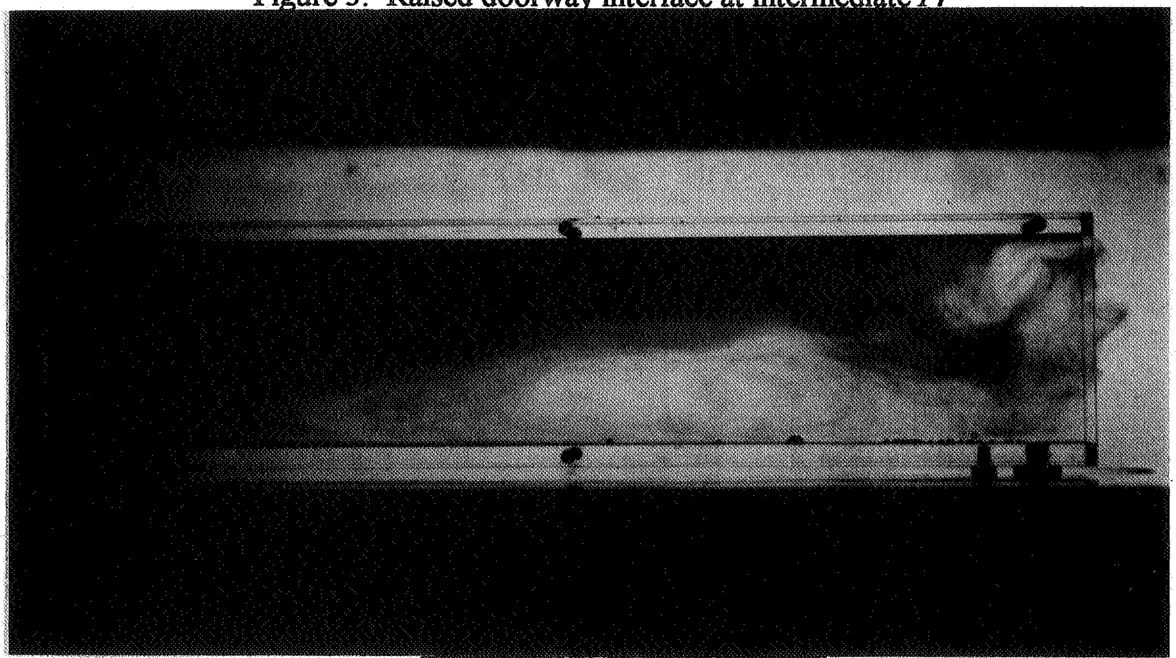

Figure 4. Flow at high *Fr*

For Froude numbers greater than $Fr \approx 8$, interfacial mixing had increased to the point where no clear interface existed between inflow and outflow at the doorway. For Froude numbers of this magnitude and higher, fluid within the room was observed to be well mixed near the doorway. Wind effects dominated the flow here. In spite of this, a gravity current was seen to flow from the mixed doorway region towards the interior as shown in figure 4. As a result of mixing, the gravity current contained both buoyant and ambient external fluid. The density difference driving the gravity current was less than that in the low Froude number cases. Therefore it was expected that the gravity current velocity would be lower than at $Fr = 0$.

At high Froude numbers, the flow between the doorway and the exterior was seen to be unsteady and irregular. Puffs of fluid of various sizes were seen leaving at random heights through the doorway.

In the case $Fr = \infty$, exterior fluid entered the room by means of a wind driven turbulent mixing process. Fluid was mixed across the doorway at a rate proportional to the wind speed. The amount of wind driven mixing reduced further down the room because turbulence was damped by viscosity. It is worth noting that the flow would be very different if the leeward end of the room were uncovered. In that case, a "plug" flow would rapidly purge the buoyant fluid from the room.

2.2.2 Quantitative results

The scaled velocities and scaled volume fluxes of the observed gravity currents are plotted against Froude number in figures 5 and 6. Note that the effect of interfacial mixing at high Froude number is to reduce both the velocity and volume flux associated with the gravity currents. Fractional heights occupied by gravity currents ranged from close to 0.5 at $Fr = 0$ to approximately 0.35 at $Fr = 11.4$. The velocity of the gravity current at high Froude number was found to be approximately 40% of the velocity at $Fr = 0$.

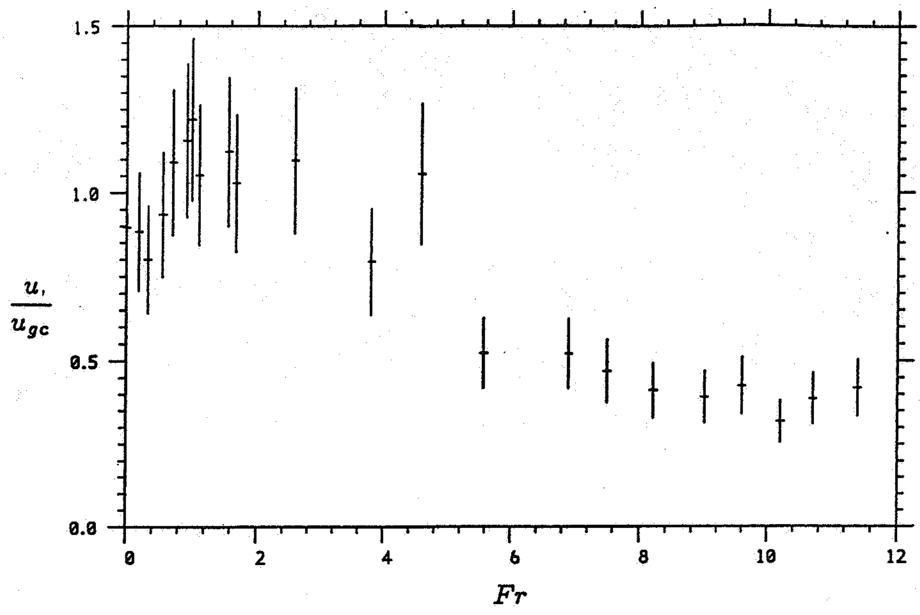

Figure 5. Scaled velocity plotted against *Fr*

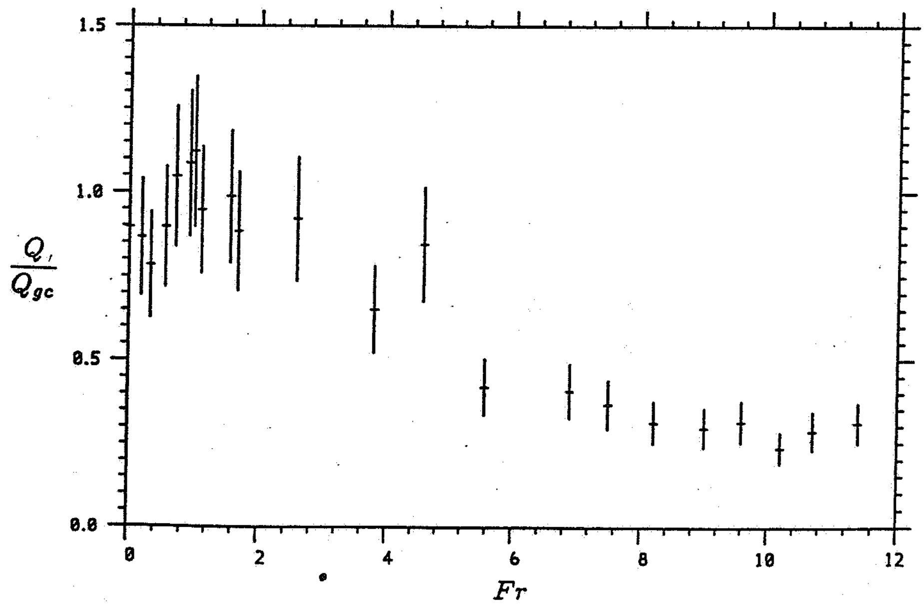

Figure 6. Scaled volume flux plotted against *Fr*

2.3 Continuous Flows

2.3.1 Qualitative results

The nature of the flow at *Fr* = *0* depended significantly on the source volume flux, Q_s. Varying the reduced gravity, $g'=g\Delta\rho/\rho$, did not noticeably affect the qualitative nature of the flow. For small values of Q_s, the buoyant plume entrained less fluid from the room than plumes produced by higher source volume fluxes. For this reason, low values of Q_s led to shallow gravity currents. After the gravity current head had left the room, the flow reached a steady state and a stable two-layer stratified flow was established within the room.

For intermediate Froude numbers ($1 \leq Fr \leq 4$), the only qualitative difference to the flow occurred near the doorway. Once again, the different pressure distribution caused by the headwind led to a shallowing of the outflow through the doorway. The influence of the wind, however, was localised to the region near the doorway. Further down the room, the flow was qualitatively similar to that at *Fr* = *0*.

Further increases in Froude number again led to increases in interfacial mixing. For Froude numbers greater than $Fr \approx 7$ no interface between inflow and outflow at the doorway was apparent. The flow just within the doorway was dominated by the wind turbulence. Nonetheless, buoyancy dominated the flow within the room. The buoyant source plume appeared relatively unaffected by the external wind.

2.3.2 Quantitative results

Experimental measurements in the case $Fr = 0$ give the least squares fit relationship between u, the velocity in the buoyant layer, and B, the buoyancy flux per unit width of the source, as

$$u = (0.83 \pm 0.07)\, B^{(0.35 \pm 0.04)}, \qquad (3)$$

where $B = g'Q_s$. This value is close to the theoretical prediction for layers each occupying half the height of the room $u = (B/2)^{1/3}$.

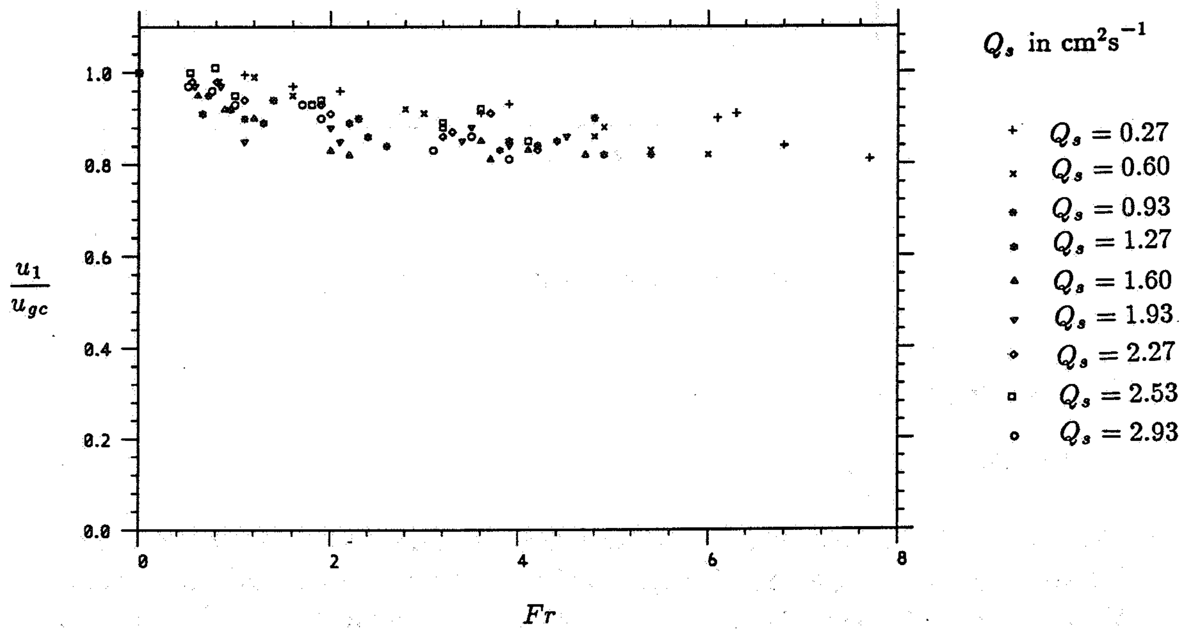

Figure 7. Scaled velocity of buoyant layer plotted against Fr

Figure 7 shows a plot of scaled velocity within the buoyant layer against Froude number. The velocities are scaled with experimentally measured values at $Fr = 0$.

Unlike the transient case of §2.2, velocities within the buoyant layer decrease only slightly with increasing Froude number. This result demonstrates that the entraining plume at the closed end of the room strongly influences the two-layer exchange flow. The observation also suggests that the amount of entrainment into the plume is not drastically affected by the density of fluid it entrains.

3. Mathematical Models

3.1 Transient Flows

3.1.1. Gravity Current Model at $Fr = 0$.

Inviscid dissipationless theory is used in [7] to predict that a gravity current flowing into an infinitely deep office would occupy half the height of the room and have a steady velocity

$$u = \frac{1}{2}(g'D)^{\frac{1}{2}}, \tag{4}$$

where $g' = g\Delta\rho/\rho$ is the reduced gravity. In reality, energy loss occurs. Experimental measurements described in [8] found considerable variations in velocity with changing height. They expressed the gravity current velocity as

$$u = k(\lambda)(g'D)^{\frac{1}{2}}, \tag{5}$$

where λ is the fractional height of the gravity current. The results of [8] found values including $k(0.3)=0.41$ and $k(0.05)=0.3$. The experiments at high Fr revealed a gravity current with $\lambda=0.35$ corresponding to $k=0.43$.

3.1.2 Mixed Doorway Region Model At High Froude Number

Consider a flow régime illustrated in figure 8. Assuming the flow to be steady, it can be deduced that

$$(\rho + \Delta\rho)Q_0 - (\rho + \delta\rho)Q_0 + \rho Q - (\rho + \delta\rho)Q = 0. \tag{6}$$

It is then possible to solve this equation to produce an equation relating the volume flux carried from the mixed region to the interior and the volume flux entrained through the doorway.

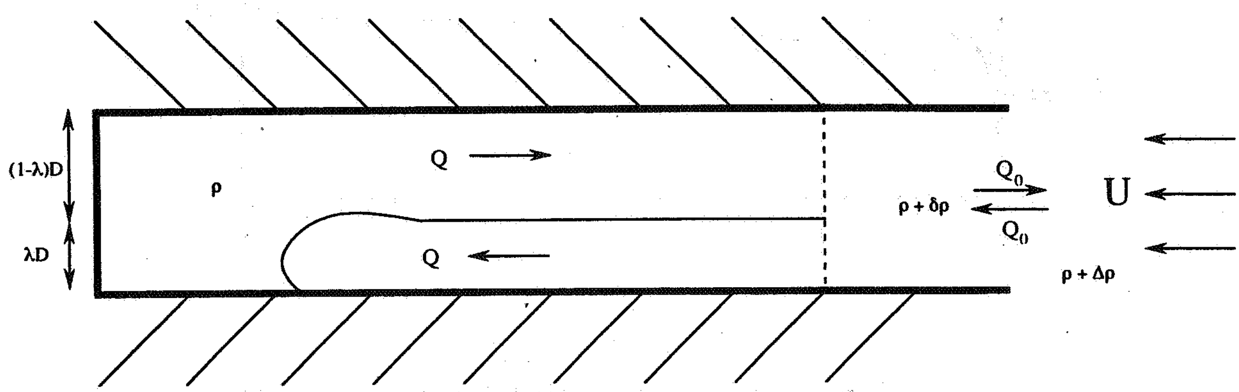

Figure 8. Model of flow at high Fr

The volume flux per unit width carried into the interior, Q, is given by

$$Q = 4k\lambda\,\phi^{\frac{1}{2}}Q_{gc} \tag{7}$$

where

$$Q_{gc} = \frac{D}{4}\left(g\frac{\Delta\rho}{\rho}D\right)^{\frac{1}{2}} . \tag{8}$$

Equations (7) and (8) together imply

$$(\sigma + K)\sigma^2 - 16k^2\lambda^2 K = 0 \tag{9}$$

where $\sigma = Q/Q_{gc}$ and $K = Q_0/Q_{gc}$. Using an entrainment assumption in the spirit of that proposed in [9], it is possible to hypothesize a value for the volume flux across the doorway. Assuming that the mixing effects of the wind turbulence drive the flow across the doorway then the volume flux across the doorway can be estimated as

$$Q_0 = \frac{DEU}{2}, \tag{10}$$

where E is an entrainment constant.

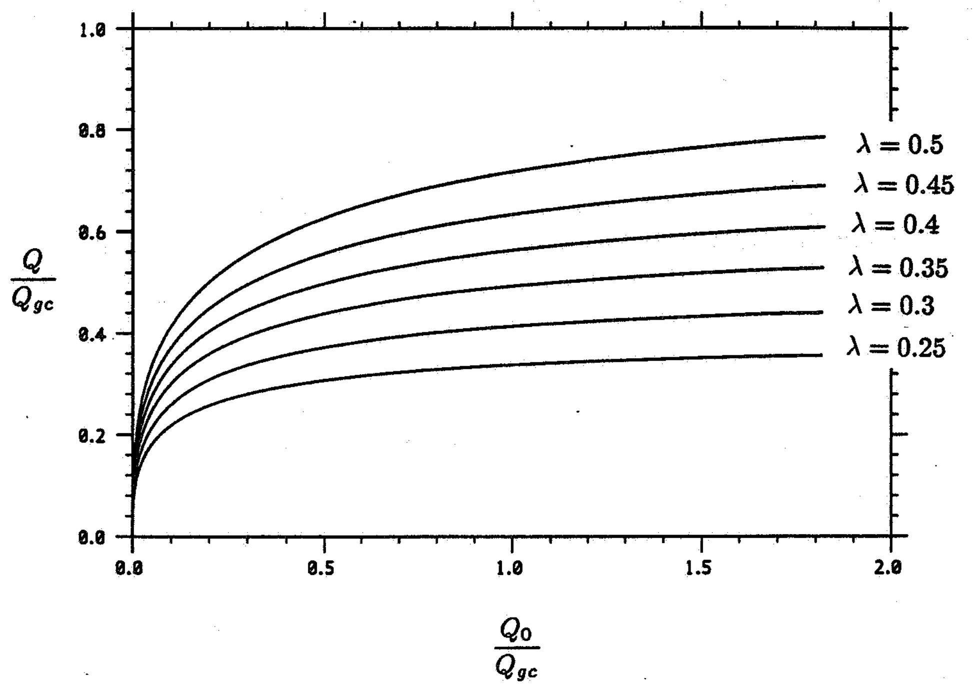

Figure 9. Flux along the room plotted against flux through the doorway

Figure 9 shows a plot of flux along the room as a function of flux through the doorway. The results are shown for a range of values of λ. Corresponding values of k are taken from [8]. The experimental results suggest a value $E = 0.003$ as the appropriate entrainment constant. This value of entrainment constant is considerably lower than that for a plume or jet.

3.2 Continuous Flows

3.2.1 The Entrainment Counterflow at $Fr = 0$

One way to analyse the flow at $Fr = 0$ is to consider the amount of entrainment into the buoyant source plume at the closed end of the room. Using an entrainment assumption it is possible to predict the volume flux of fluid entrained into the plume. Then, the total volume flux carried in the buoyant layer is simply

$$Q = Q_e + Q_s \tag{11}$$

For a two-dimensional buoyant plume (see [10]), plume theory predicts that the mean velocity within the plume is given by

$$u = \left(\frac{B}{e}\right)^{\frac{1}{3}} \tag{12}$$

where e is an entrainment constant. If λ is the fractional height occupied by the buoyant layer then the volume flux of fluid entrained can be predicted to be

$$Q_e = \left(1-\lambda\right) D e^{\frac{2}{3}} B^{\frac{1}{3}} \tag{13}$$

Conservation of mass then implies that

$$\left(Q_e + Q_s\right)^{\frac{3}{2}} = k\lambda\, D^{\frac{3}{2}}\left(g\frac{\Delta\rho}{\rho}\right)^{\frac{1}{2}} Q_s^{\frac{1}{2}} . \tag{14}$$

The experimental results lead to a value $e \approx 0.09$. This value is typical for buoyant plumes.

3.2.2 Mixed Doorway Region Model At High Froude Number

For this continuous case, a simple model can be constructed for the flow at high Froude number. The model is similar to that in the transient case. This model predicts that the velocity of the buoyant layer at high Froude number will be less than that at $Fr = 0$ by a factor

$$\left(\frac{Q_0}{Q_0 + Q_s}\right)^{\frac{1}{3}} \tag{15}$$

Close agreement is found between experimental data and the model. Using an entrainment hypothesis similar to that described for the transient case described above, it is possible to estimate the entrainment into the doorway region. Comparison with the experiments suggests a value $E = 0.008$. This value is higher than that predicted for the transient case. This result could suggest that there are fundamental differences between the mixed doorway regions in the transient and continuous cases.

3.2.3 Pollutant Dispersion

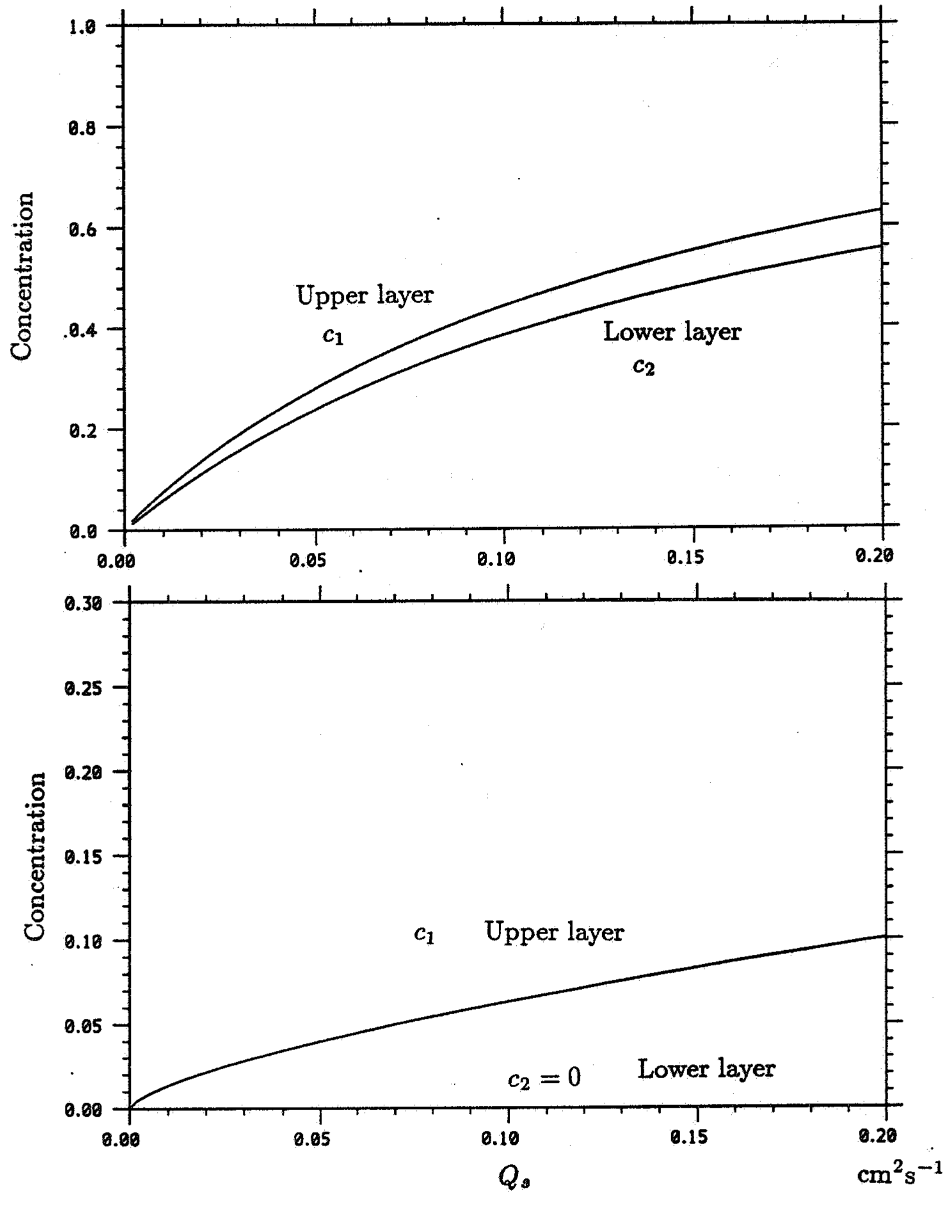

Figure 10. Concentration predictions at high and low *Fr*

The continuous source of buoyancy described above could well be a leaking gas pipe. In this case, the concentrations of gas in both the upper and lower layers within the room are of some significance. Mixtures of natural gas and air with 5 to 15% gas are potentially explosive. When wind effects are small, the concentration of gas within the lower layer is close to zero. This would not be the case in very windy external conditions where mixing at the doorway leads to some natural gas being carried from the doorway region back down the room. Using the model described in the previous section it is possible to predict gas concentrations within both the upper and lower layers. These concentration predictions are shown in figure 10 as functions of source volume flux both at high and low values of Froude number. The predictions assume $E = 0.008$ as suggested by the continuous experiments.

4. Conclusions

Several conclusions can be drawn from these studies. Firstly, an increase in headwind can be detrimental to single-sided natural ventilation. Air change rates can be reduced. Polluted air from the buoyant layer can be mixed into the lower layer, increasing levels of pollutant there. The second main deduction is that, for single-sided ventilation, wind effects are largely confined to the region near the doorway. Buoyancy forces dominate the flows away from the doorway region.

The results in this study are consistent with full scale tests carried out on a test room [2] and on a test house [11]. Buoyancy driven flows through a single opening under the additional influence of turbulent mixing on one side of the opening were studied in [12]. He also observed a decrease in volume flux through the doorway

Acknowledgements

This work was carried out while G.M.J.D. was studying for a Ph.D. degree at the Department of Applied Mathematics and Theoretical Physics, University of Cambridge under the supervision of Dr. P.F. Linden. The Ph.D. study was funded by a U.K. Science and Engineering Council CASE Award co-sponsored by the Ove Arup Partnership.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Investigation of Ventilation Conditions in Naturally Ventilated Single Family Houses

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Synopsis
The reason for the present project is the need for more reliable information about the actual ventilation conditions in naturally ventilated, detached houses. The aim has been to quantify the ventilation and humidity conditions and to establish a better basis for elaborating directions and guidelines on proper ventilation of detached houses.

A national questionnaire survey covering more than 2100 households has been carried out, together with detailed investigations in about 150 houses. The investigations comprised measurements of the average outdoor air supply and the average relative humidity. The main bedroom was investigated separately. The measurements were performed during the heating period. Passive measurement techniques were used.

Results show that the air change rate on average is about 0.35 h^{-1}. In more than 80 per cent of the houses the air change rate is lower than the recommended rate of 0.5 h^{-1}. The relative humidity is on average 0.45 in the living-room and 0.53 in the bedroom.

1. Introduction

In detached houses, the ventilation is often based on natural ventilation. The principle is that air is removed from the house through vertical air ducts in the kitchen and bathroom/toilet, while replacement air - outdoor air - is supplied to the other rooms in the house through open windows and/or outdoor air inlets and random leaks in the building envelope.

Natural ventilation systems are often evaluated on the basis of theoretical considerations. The function of the system is based primarily on the difference between the temperature of the room air and that of the outdoor air, but is also affected by wind. Consequently, the effectiveness of the system is more dependent than other ventilation systems on the outdoor climate. Also the behaviour of the occupants affects the performance of the system. Therefore, natural ventilation systems should be evaluated, instead, on practical experience and results obtained by measurements - preferably long-term measurements - in occupied houses.

Results of earlier field investigations [1] [2] have indicated that the air change rate is low in naturally ventilated, detached houses compared to traditional views on appropriate ventilation. However, the material was too limited to allow any reliable conclusions to be drawn on causal relationships. Aiming at procuring more reliable information about the actual ventilation conditions in naturally ventilated houses a national questionnaire survey covering more than 2100 households has been carried out and detailed measurements have been performed in about 150 houses. In connection with the measurements the occupants completed a supplementary interview form. This paper deals with the results of the measurements.

2. Procedures and measurement techniques

From the Ministry of Housing's Register of Buildings and Houses, BBR, about 2100 addresses of detached houses built since 1982 were selected. The addresses were selected at random, however, the geographical distribution of the houses was representative with respect to the total number of detached houses built in Denmark between 1982 and 1989. In addition to the addresses BBR reported various design data for the houses.

A questionnaire was sent to the householders. The questionnaire dealt primarily with questions concerning the household and its use of the house and questions concerning ventilation arrangements installed in the house. About 1400 householders, corresponding to 67 per cent, returned usable replies. Based on these replies 150 houses were selected for closer examination.

Ventilation and humidity measurements were carried out. The ventilation measurements were performed using a passive multiple tracer gas method, the PFT-method [3] [4], and the relative humidity was measured using moisture-calibrated beechwood blocks [5]. The measuring period in each house lasted about two weeks. In connection with the placing of the passive measuring equipment in the houses, the room air temperature was measured both in the living-room and in the main bedroom, and the occupants completed a supplement interview form.

3. Results

In about 150 naturally ventilated, detached houses measurements have been performed of the average total outdoor air supply and of the average outdoor air supply to the main bedroom. Also the average relative humidity in the living-room and in the bedroom have been measured together with the room air temperature in the rooms mentioned. Through simultaneous use of two different tracer gases, the air exchange between the bedroom and the rest of the house has been determined. The air change rate of the house has been calculated as the ratio between the average total outdoor air supply and the net volume of the house. Table 1 shows some characteristics of the houses investigated and table 2 shows the main results of the measurements.

Table 1. Some characteristics of the houses.

	Average
Net floor area of house	115.6 m^2
Area of main bedroom	15.1 m^2
Occupants per house	3.3 persons
Adults (≥ 16 yrs) per house	2.2 persons
Children (< 16 yrs) per house	1.1 persons
Living area per person	38.5 m^2/pers.

Table 2. Main results of the measurements.

	Average ± Standard Error		
Whole house			
Outdoor air supply	25.5	± 0.9	l/s
Outdoor air supply per m^2	0.22	± 0.01	l/s per m^2
Outdoor air supply per pers.	8.36	± 0.30	l/s per pers
Air change rate	0.35	± 0.01	h^{-1}
Living-room			
Relative humidity	0.45	± 0.01	
Room air temperature	21.2	± 0.1	°C
Main bedroom			
Outdoor air supply	4.5	± 0.2	l/s
Outdoor air supply per m^2	0.31	± 0.02	l/s per m^2
Relative humidity	0.53	± 0.01	
Room air temperature	20.9	± 0.2	°C

Figure 1 shows the results of the measurements of the total outdoor air supply, l/s, and the calculated air change rate, h^{-1}. The curves show the percentage of the houses in which the total air supply and the air change rate, respectively, is lower than the values shown on the abscissa.

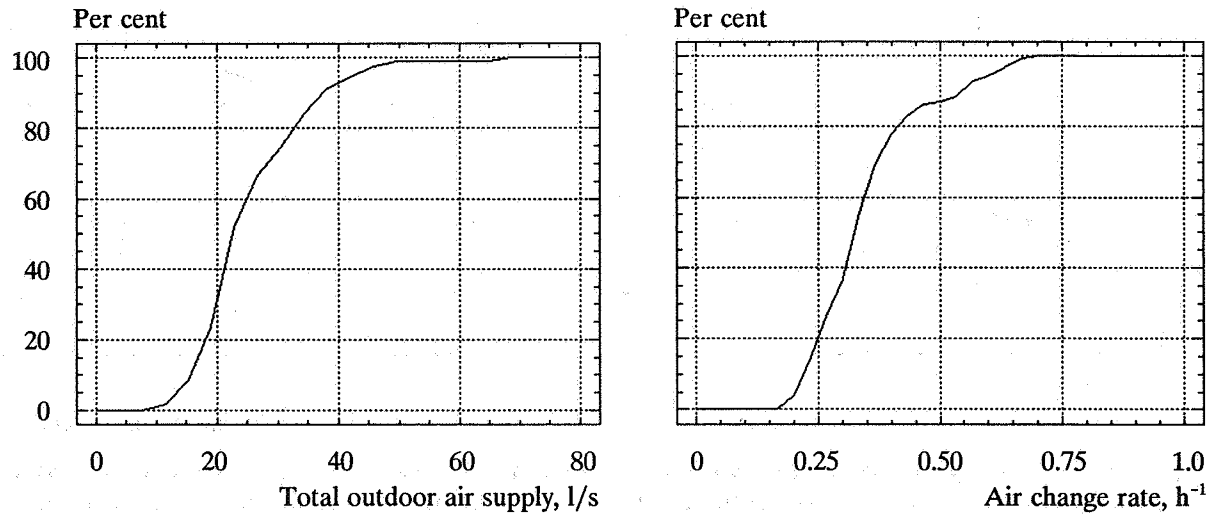

Figure 1. Cumulative, relative frequency of the total outdoor air supply, l/s and of the air change rate, h^{-1}.

Figure 2 shows the measured airflows *from* the bedroom *to* the rest of the house and *from* the rest of the house *to* the bedroom. The difference between the two airflows is denoted the net transference of air. Acceptable results exist from measurements in 102 houses. The measurement results have been numbered from number 1 to number 102, sorted according to increasing net transference of air and displayed on the abscissa on figure 2.

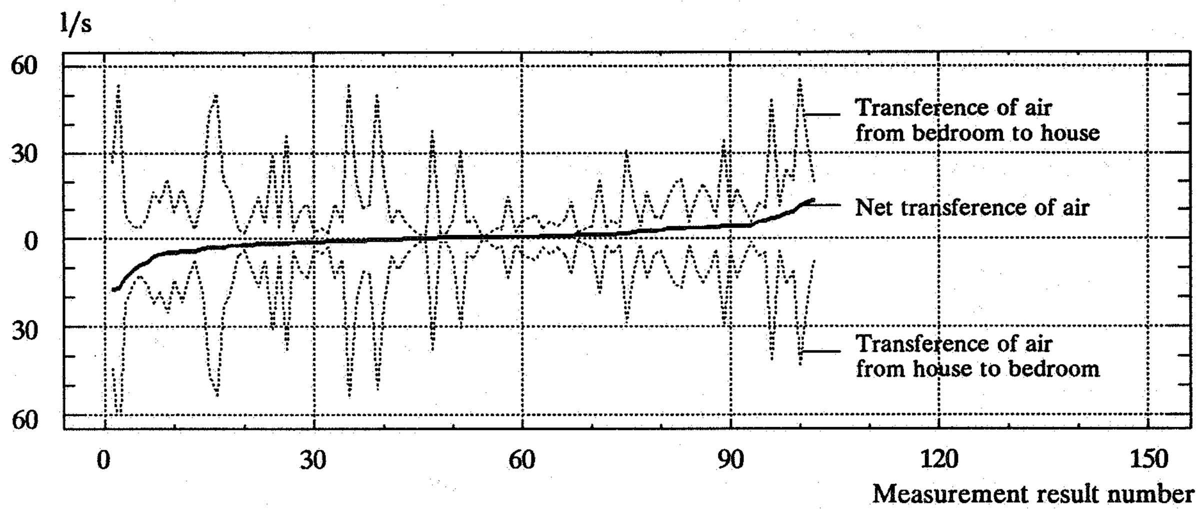

Figure 2. Internal air exchange between bedroom and the rest of the house. Airflow displayed above the horizontal 0-line indicates transference of air from bedroom to the rest of the house. Airflow displayed below the 0-line indicates transference of air in the opposite direction, i.e. from the house to the bedroom. The net transference of air is the difference between the two airflows. The abscissa is a numbering of 102 measurement results sorted according to increasing net transference of air.

Figure 3 shows the regression for the relation between the relative humidity of the room air and the outdoor temperature. Regressions for the living-room and the bedroom are shown.

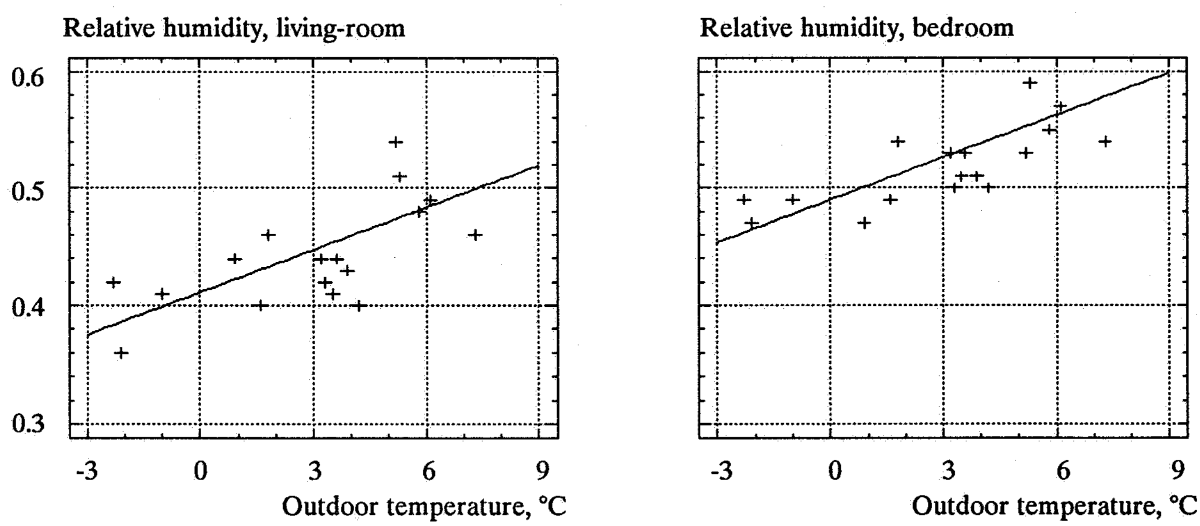

Figure 3. Regression of the measured relative humidity in the living-room and in the bedroom, respectively and the outdoor temperature. Each point represents the average of a number of measurement results measured simultaneously and at the outdoor temperature in question. The regressions have been calculated on the basis of the individual results, i.e. not only on the average values.

Figure 4 shows the cumulative, relative distributions of the measured relative humidity of the room air.

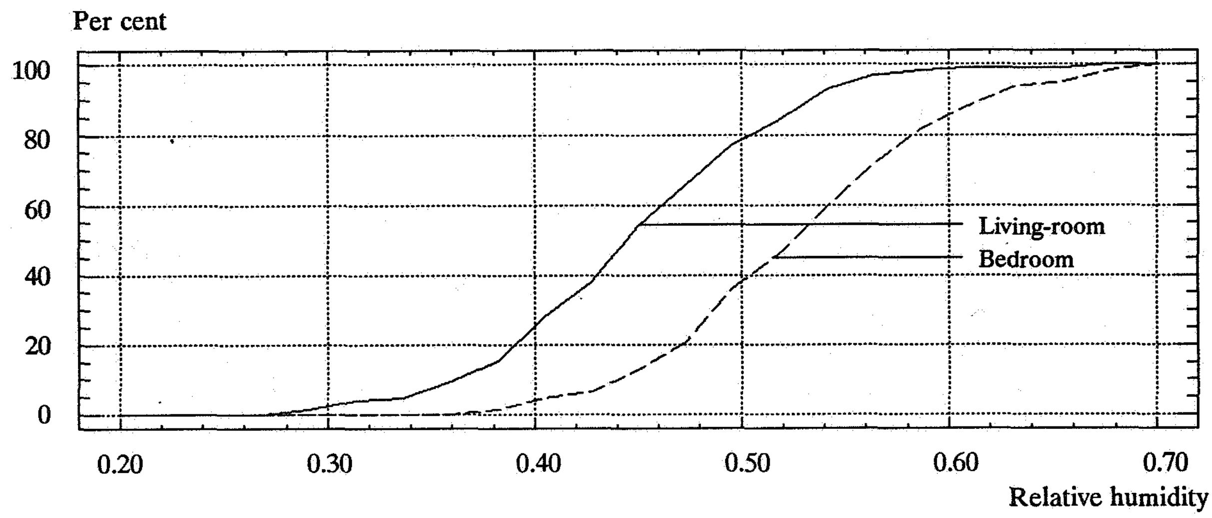

Figure 4. Cumulative, relative frequency of the relative humidity of the room air. The curves show the percentage of the houses in which the relative humidity is measured to be lower than the values shown on the abscissa.

On the basis of the measured room air temperatures and relative humidity the vapour pressure and the vapour content of the room air can be calculated. The vapour content of the room air is governed by the vapour content of the outdoor air supplied, together with the supply from moisture producing sources such as persons, cooking, washing and house cleaning. Providing stationary conditions the relative supply from moisture producing sources to the vapour content of the room air mainly depends on the ventilation. Table 3 shows the difference between the vapour content of the room air and that of the outdoor air, together with the calculated average moisture supply for the whole house as well as for the bedroom alone. The calculations for the whole house are based on the measurements of the relative humidity in the living-room and on the measurements of the total outdoor air supply to the house including the bedroom.

Table 3. Difference in vapour content indoor/outdoor, g H_2O/kg air, and the calculated average moisture supply, kg H_2O/day and kg H_2O/day per person.

	Average ± Standard Error
Difference in vapour content indoor/outdoor	
Living-room	3.2 ± 0.1 g H_2O/kg air
Bedroom	4.3 ± 0.1 g H_2O/kg air
Moisture supply	
Whole house	8.2 ± 0.3 kg H_2O/day
Whole house per person	2.7 ± 0.1 kg H_2O/day per person
Bedroom	2.0 ± 0.1 kg H_2O/day

4. Discussion

From table 2 it can be seen that the average air change rate is measured to be 0.35 h^{-1}, and figure 1 shows that the air change rate is 0.5 h^{-1} or lower in about 85 per cent of the houses investigated. The Danish Building Code states as a general rule concerning the basic ventilation of houses, that the ventilation of a house should enable an air change rate of at least 0.5 h^{-1}.

A frequently stated value for the necessary basic ventilation in houses is 0.35 l/s per m^2 net floor area. In houses with normal room height, that corresponds to an hourly supply of outdoor air equal to about half the net volume of the house, i.e. an air change rate of 0.5 h^{-1}. It will be seen from table 2 that the supply of outdoor air per m^2 net floor area in the houses investigated averaged 0.22 l/s per m^2.

The purpose of ventilating a dwelling is, in addition to satisfying people's needs for acceptable indoor air quality from the point of view of hygiene and comfort, to control the humidity conditions in the rooms. The direct moisture emission from one person is in the order of 50 g vapour per hour corresponding to about 1 kg per day. However, the total moisture supply from people and processes in a household is considerably larger. As a key figure it is often assumed that a family of four supply about 10 kg water per day to the room air. From table 3 it can be seen that in this study it has been found that the average moisture supply is 8.2 kg water per day. On average the size of a household in this study is 3.3 persons. Calculations of the average moisture supply per person show that one person supplies 2.7 kg water per day. The results of the measurements thus substantiate the key figure. Table 3 also shows that on average the bedrooms are supplied 2.0 kg water per day. It must be noted that normally the bedrooms are only used part of the day and the bedrooms are used by 1-2 persons.

Two different viewpoints can be put forth to form basis for recommendations to the maximum acceptable level of the humidity of the room air in residential buildings. One viewpoint is that the vapour content of the room air must be so low that the number of house dust mites are reduced to none or just a few per gramme house dust. Another viewpoint is that the humidity must be kept at a level where condensation on the windows will not occur.

Regarding the first viewpoint, the vapour content of the room air ought to be lower than 7 g water per kg air, corresponding to a relative humidity of about 0.45 at 20-22 °C, a couple of months in the winter period. From figure 4 it can be seen that at the time the measurements were performed the average relative humidity in the living-room was more than 0.45 in about half of the houses investigated. In the bedrooms the relative humidity was on average lower than 0.45 in about 10 per cent of the houses. In practice, the risk of an elevated humidity level leading to moisture problems can be most distinct in bedrooms, as the moisture production often will take place while the ventilation is low.

Regarding the second viewpoint, where condensation on the windows must be prevented, a difference in vapour content between indoor and outdoor air of 2.5-3.0 g water per kg air will usually not result in condensation problems. A difference of 4.0-5.0 g water per kg air may cause problems in connection with double glazing, when the temperature is lowered and curtains are drawn. Table 3 shows that the average difference in vapour content between indoor and outdoor is 3.2 g water per kg air in the living-room and 4.3 g water per kg air in the main bedroom.

Approximately two thirds of the measurements were performed in January and February and one third of the measurements were performed in October and November. Examination of the results of the measurements of the relative humidity with respect to the measurement period show that the relative humidity on average is significantly higher in the autumn/winter period than in the winter/spring period. The results reflect known variations in the humidity of the outdoor air. Variations in the humidity of the room air are to some extent subdued because of moisture accumulation in building materials and furniture. However, the results support the theory that the moisture accumulated will be released when the temperature falls, i.e. at the beginning of the heating season.

The average outdoor air supply to the bedrooms have been measured to be 4.5 l/s, cf table 2. The generally accepted minimum outdoor air supply to a bedroom from the point of view of hygiene and comfort is 4 l/s per person. If the bedrooms on average are used by 1.5-2 persons, the ventilation in the bedrooms thus appears to be lower than desirable. However, the results of the measurements of internal air movements show that there is an air transfer of at least 5-10 l/s from the rest of the house to the bedroom. Assuming that this air transfer acts to some extent as a supplement to the ventilation, it can be concluded that the bedrooms are satisfactory ventilated.

Natural ventilation systems are based on thermal buoyancy in the ducts. The driving forces originate in the temperature difference between indoor and outdoor, and the effect is therefore dependent on the outdoor climate. In addition, the efficiency of the system is dependent on the vertical height of the duct. In low houses the conditions for obtaining sufficient height and with that reliable performance are unfavourable, especially if the inclination of the roof is small. In this study about 90 per cent of the houses were one-storeyed. Furthermore, the system is susceptible to influence from wind just as the internal distribution of the outdoor air supplied is exposed to the action of the wind. The performance of the natural ventilation system is also influenced by the tightness of the building envelope. Theoretically, the performance will be reduced concurrently with increasing tightness. However, a tight building envelope will, together with correct placing and proper construction and use of outdoor air inlets, be able to improve the possibilities for providing appropriate internal air distribution in the house.

5. Conclusion

In defiance of the fact that natural ventilation systems now and then are considered obsolete there is no proof that the system is so unsuitable that the use should be dissuaded. The ventilation installations can only provide the possibility for the users to ventilate properly. The attitude and the behaviour of the occupants are determining for the actual ventilation conditions in dwellings is. From the national questionnaire survey it was found that in spite of the low ventilation rates about 97 per cent of the households judged the air quality in their house as "fresh" or "ordinary" and only a few per cent judged the air quality as "poor".

It has not been possible in this study to identify structural or design factors with such a crucial effect on the ventilation and humidity conditions that changes would definitely be regarded as likely to improve matters. The ventilation problems encountered in lowering room temperatures and tightening buildings in order to reduce energy consumption are presumably not only technical but also a question of proper information.

6. References

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Methods for Investigating Indoor Air Conditions of Ventilated Rooms

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Synopsis

The acquisition of temperatures and velocities is a permanent recurrent task for the investigation of air flow in ventilated rooms. On the one hand it is important to measure the temperature and velocity field with a high spatial resolution. On the other hand, in general, varying outdoor conditions prevent from reaching a steady state and an additonal demand consists in short measuring times.
Sometimes, the obtained measuring results are used both to supply appropriate boundary conditions for numerical computations and to verify the CFD-codes used. Therefore, the processing of received data has further importance.
In this paper the advantages and limitations of thermography, the usage of thermocouples and hot wire anemometry for temperature and velocity measurement is discussed. It is shown how the application of modern system components and data post processing in connection with these methods can satisfy better the already mentioned requirements.

List of symbols

T [°C]	-	temperature
U [V]	-	voltage
ε	-	emissivity

1 Introduction

Nowadays the investigation of temperature and flow in ventilated rooms is carried out by a variety of different methods. Each method shows advantages and disadvantages. Requirements related to a high resolution in space and time are commonly connected with extensive technical effort and high investments.
Therefore, the improvement of conventional methods for field measurements is focused on.

2 Temperature measurement

2.1 Thermography

Thermograhpy utilizes the emitted radiation in the infrared spectrum of a body to determine its temperature without contact. In contrast to pyrometry which records an average radiant density the thermography provides a radiant density distribution. Commonly, the emissivity of the measured object effects substantially (compared to other influences for instance reflection of background radiation, absorption in the air etc.) the conversion of radiant density into temperatures [1]. Using a one-point-calibration, the emissivity is easily calculated based on a local temperature measurement at the surface of the object under investigation (e.g. with a thermocouple). This emissivity is applied to the hole infrared picture. Errors occure, if this local emissivity varies along the measuring object. In most cases there is also no co-ordinate transformation which maps picture points onto object points.

These two disadvantages are removed by applying a suitable image post-processing* . Since the thermography system is used for temperature measurement in rooms with plane rectangular walls, the co-ordinate transformation can be reduced to a simple picture rectification. Objects recorded by the camera are distorted according to a central projection. Figure 1 shows a geometric method how a picture point P can be related to a point in the wall plane by repeated bisection.

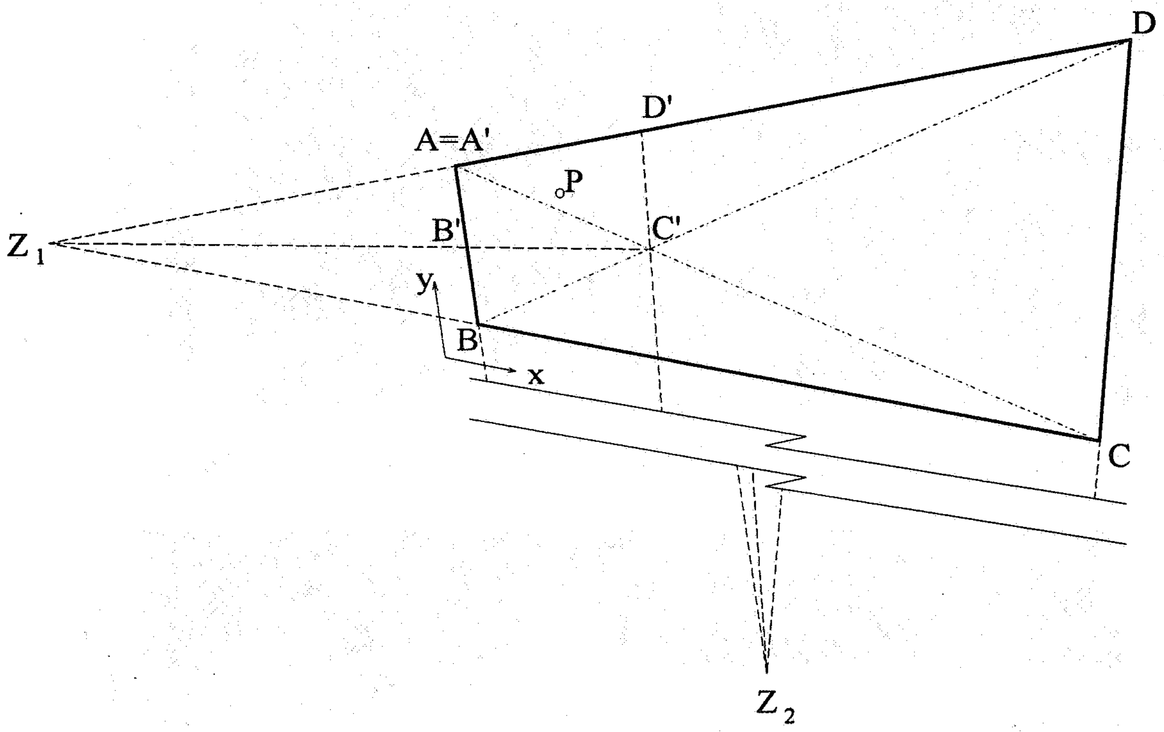

Figure 1: Rectification of a plane rectangular wall using a simple geometric bisection method

For the rectangle ABCD in the wall plane as well the wall as the picture co-ordinates are known. The intersection of the two diagonals defines an additional point. The two connecting lines between this point and the projection centres Z_1 and Z_2 split the region into four subregions. It follows the determination of the subregion (A'B'C'D') in which the considered point P is located. Afterwards, the process of bisection is restarted and continued as long as the subregion reaches the picture resolution. Because a lot of picture points have to be mapped, sophisticated programming allows to reduce the computing costs per point.
The particular advantage of this method is that neither the camera position nor the camera orientation need to be known. Only four picture points forming a rectangle in the wall plane have to be indentified by their wall co-ordinates.
If required, local emissivities for different regions (e.g. windows) can be read in from an external geometry file. These emissivities can be either obtained by the one-point-calibration mentioned above or they are already known. Finally, the corrected temperatures are available in dependence on the wall co-ordinates,
Figure 2 shows the original infrared picture of an inside wall with 3 windows of a gymnasium [2]. Picture rectification and emissivity correction as described above lead to figure 3.

* The presented algorithm is implemented in a selfmade image post-processing software

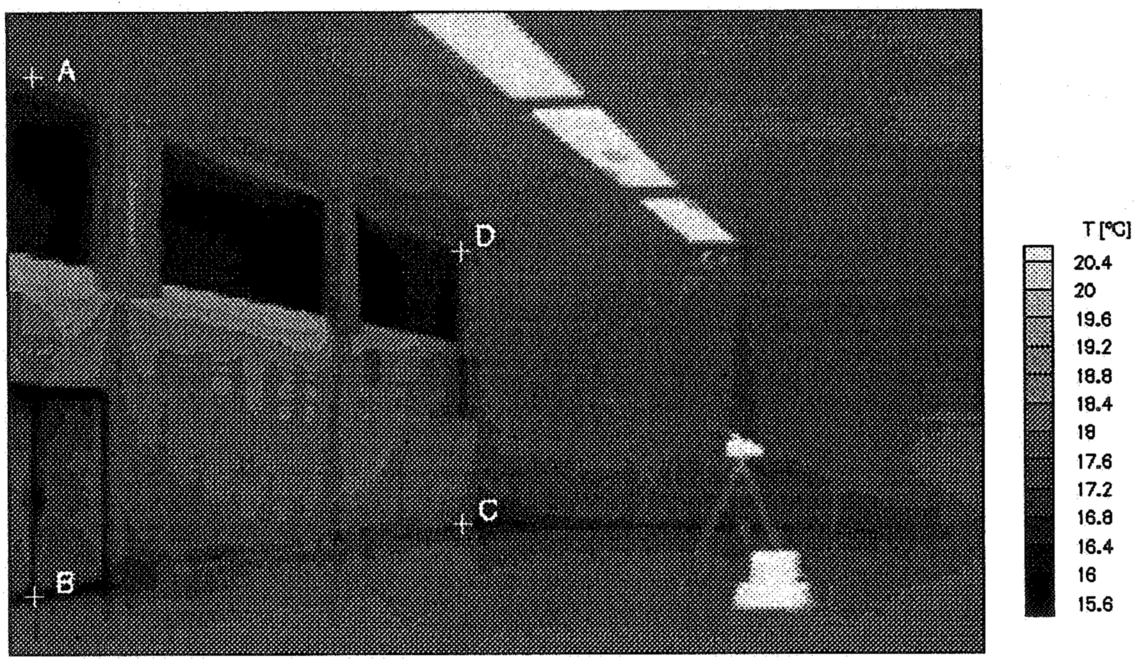

Figure 2: Original infrared picture with corner points A, B, C and D for rectification

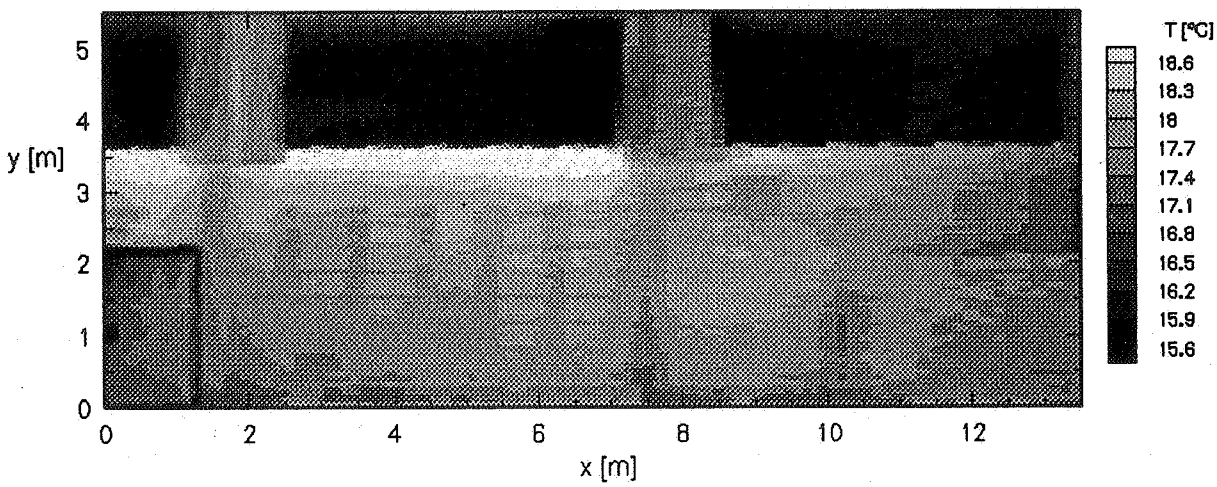

Figure 3: Obtained picture after rectification and ε-correction from figure 2

The applied camera (Jenoptik LW1011) is equipped with only one infrared sensitive element and generates its pictures using a two-dimensional reflector scanner with a resolution of 300x200. This results in costs which are much lower than for alternative systems [3]. The lower picture repetition frequency of approximately 2 Hz seems to be sufficient for this application and should not be viewed as a disadvantage.
The temperature data can also be used as boundary conditions for numerical computations. An interpolation onto the underlying grid seems to be sufficient (see figure 4).

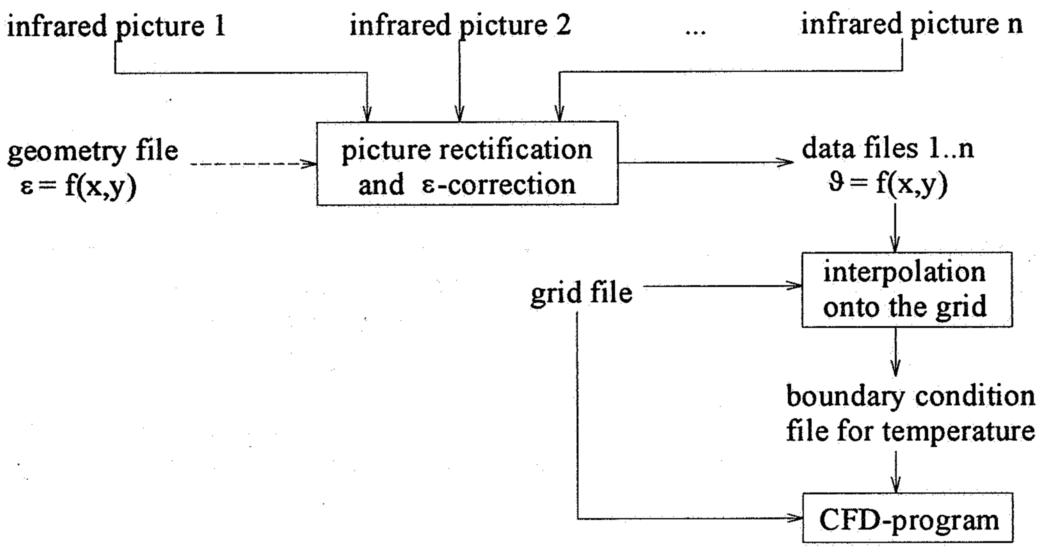

Figure 4: Structure of the infrared picture processing

2.2 Thermocouples

Methods for direct registration of a two-dimensional air temperature distribution (e.g. holographic interferometry [4]) are connected with high technical effort and high costs. They are therefore unsuitable for the practical investigation of large enclosures. Because of this the temperature field is constructed of a number of single measuring points.

Themocouples utilize the Seebeck effect after which in a closed circuit made of two materials a thermoelectric voltage is generated in dependence on the used materials and the temperature difference at the junctions [5]. The application of multiplexers (MUX) with cold-junction compensation (CJC) allows to connect a great number of thermocouples. The thermoelectric voltage is amplified and via an analogue-digital converter (ADC) read into a computer (Figure 5).

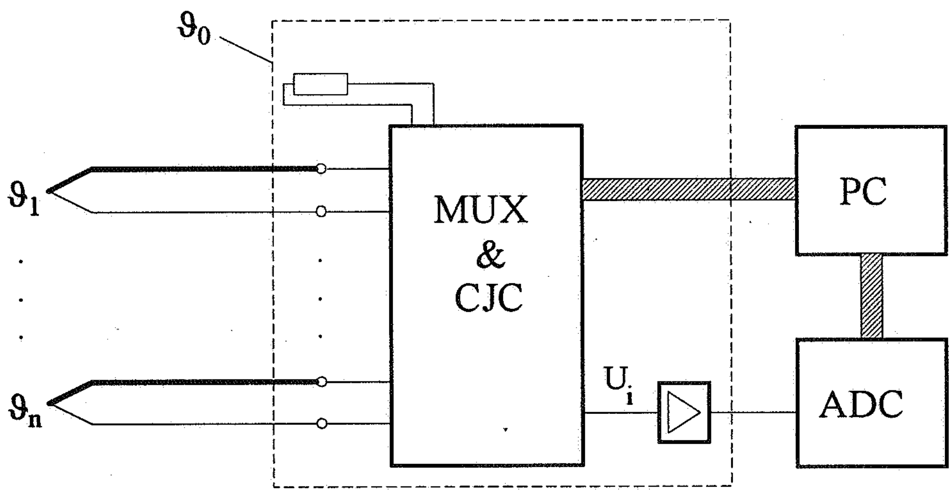

Figure 5: Measuring chain for temperature acquisition

A resistance thermometer measures the cold-junction temperature ϑ_0. For high accuracy, the multiplexer and amplifier must have the same temperature ϑ_0, because they are made also of different materials. Therefore, all these devices are placed in a thermically isolated box. The applied ADC is equiped with difference inputs causing noise reduction. Its resolution of 16 bit in connection with a polynomial function for the conversion of thermoelectric voltages into temperatures causes an accuracy of ± 0.1°C using iron-constantan thermcouples.
The thermocouples are placed in polished brass tubes protected from radiation and mounted in desired distances on a portable bar. This permits to measure the temperature along the bar nearly simultaneously. Moving the bar in a two-dimensional mesh leads to a three-dimensional temperature field. The pausing time at a point before starting the data acquisition is determined by the time constant of the thermocouples and influences the measuring time.
In any case the development of a reference temperature (e.g. intake air) should be recorded during the whole measuring process to check the constant boundary conditions or to apply a suitable correction.

3 Hot wire anemometry

Hot wire anemometry holds a special position under the numerous methods for velocity measurement [6]. Particular in turbulence research this method is often applied and although laser-optical techniques become more and more important it will be used furthermore. These are the main reasons:

- equipment for hot wire anemometry is still much lower priced than for laseroptical methods
- hot wire probes are easy to handle and can provide a high resolution in space and time
- hot wire anemometry enables to analyze momentary and average velocity vector, turbulence intensity, energy spectrum and correlations in space and time

Apart from these advantages there are also following limitations:

- disturbance of flow field → probes have to be very small
- cut-off frequency of about 10 kHz
- high mechanical sensitivity
- limitation of the spatial resolution due to wire dimension

Nowadays, the effective application of this conventional method is achieved using computer controlled system components. For example, the abilities of hot wire anemometry can be enlarged using fast ADC as following:

- functions of time with a sampling rate of 80kHz and more can be recorded
- average values, fluctuating values etc. can be investigated in nearly aribitrary time intervals
- if the ADC is equiped with simultaneous sample & hold inputs (SS&H) several probes can be measured at the same time allowing to compute multi-point correlations or velocity vectors (3- and 4-wire probes)

Figure 6 shows our applied configuration for velocity measurement, whereby up to 7 probes can be sampled simultaneously.

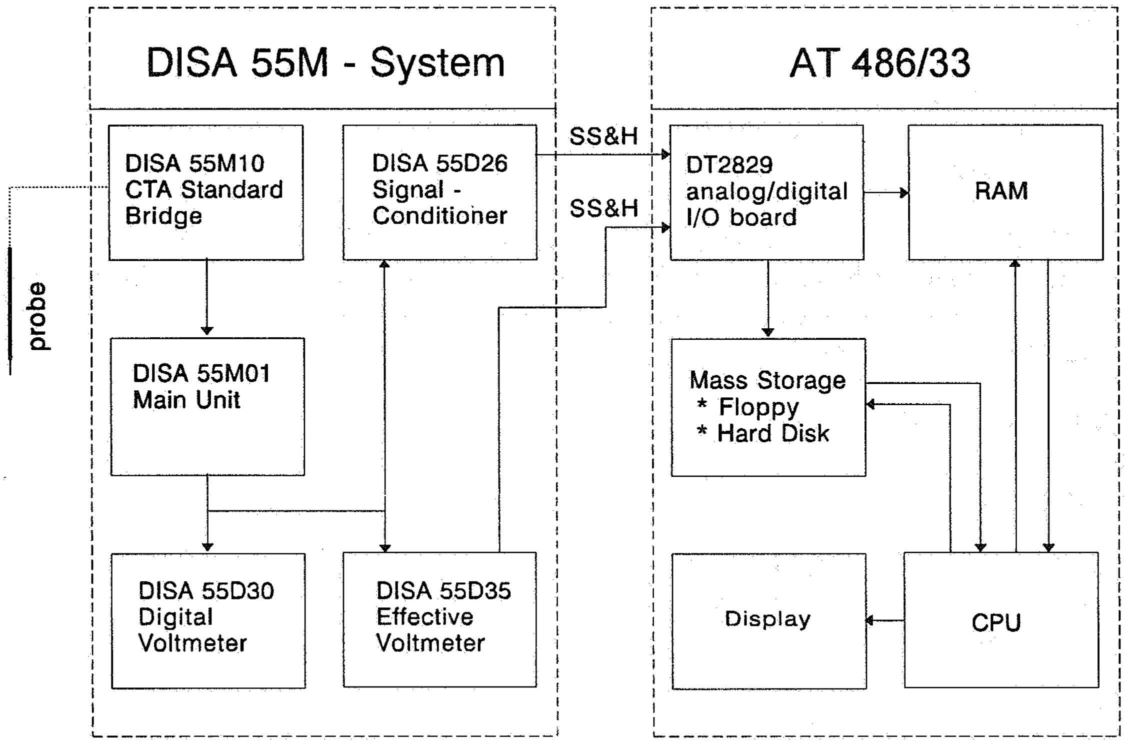

Figure 6: Measuring chain for velocity acquisition (only 1 probe shown)

A further automation is reached using positioning systems. This allows computer controlled measurement along axes, in planes and in chambers with high spatial resolution.
After processing of the numerous row data the results are easily visualized applying standard software (e.g. Tecplot™).

4 Conclusions

Thermography has a special advantage, because it supplies a complete temperature distribution. A method is presented allowing information from infrared pictures to be easily imported as temperature boundary condition into a CFD-code.
Computer controlled devices like ADC and positioning systems allow to improve and automate conventional measuring methods. Difficulties appear if thermocouples and hot wire anemometry are applied for field measurements with varying boundary conditions in time.

5 Acknowledgement

This work is part of a research project supported by the Bundesministerium für Forschung und Technologie under the contract 0329016D.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

High Quality Ventilation Systems - A Tool to Reduce SBS Symptoms

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High quality ventilation systems - a tool to reduce SBS-symptoms

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SYNOPSIS

The present case study refers to a larger office building in Sweden. The employees in this building, which was built in 1982, began to complain about the indoor environment around 1985-86. A preliminary examination of the building started in 1989.

The preliminary investigation showed that the concrete framed floors were levelled off with self levelling compound containing casein and that there were relatively high concentrations of ammonia under the PVC-flooring. Chemical measurements showed that the total VOC-concentration in the building was relatively low but that higher concentrations could be found in certain places. Average total VOC concentration 190 µg/m^3, maximum value 1230 µg/m^3, minimum value 80 µg/m^3. The concentration 2-ethyl-1-hexanol, plasticiser in PVC-flooring, was also measured. Average 2-ethyl-1-hexanol concentration 11 µg/m^3, maximum value 32 µg/m^3, minimum value 1 µg/m^3. Moisture measurements showed that the concrete frames were dry, the relative moisture in the concrete less than 60 %. Based on these findings measures relating to the floor were considered.

Next the chemical pollution in the actual ventilation system was investigated. Results showed that very high concentrations of total hydrocarbon were given off in the building at certain times and that these hydrocarbons reentered via the supply air as the ventilation system was of the recirculating type. The source of the periodically high concentrations of total hydrocarbon was a printing works located in the building. Based on these results it was decided that the device for recirculating the air in the ventilation system should be removed.

The reconstruction of the ventilation system was followed up partly with technical measurements, partly with a questionnaire among the employees both before and after the reconstruction of the ventilation system. The result of the questionnaires shows that complaints about the indoor air quality decreased to a level close to a "healthy building".

METHODS

The concentrations of carbon-dioxide, total hydrocarbon and the vapour concentration were measured with gas monitor Brüel & Kjær 1302. A sampling unit of type Brüel & Kjær 1303 was attached which made possible continuous measuring in up to 6 measuring points. Found concentrations for total hydrocarbon (THC) are given i ppm as methane. In certain reports the THC is also given as VOC_{PAS} when using this method of measuring.

The total VOC concentrations were measured using charcoal absorbents and analyses with GC-MS. The findings are given as toluene equivalents.

RESULTS

Ventilation system

Originally the ventilation system was equipped with a temperature controlled recirculation regulator with a regulation curve shown i figure 1.

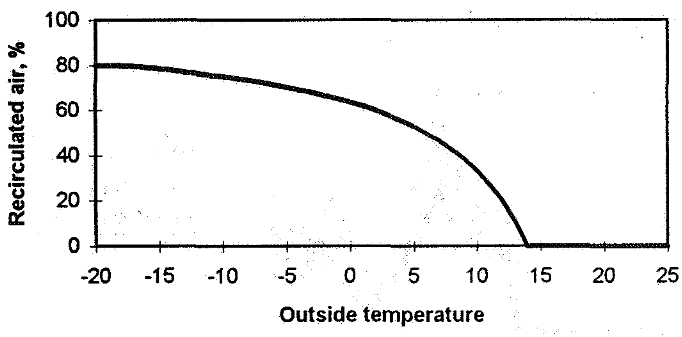

Figure 1, regulation curve

During a normal winter, November - March, the recirculation level will be on average 60 %.

The flow was originally ca 35 l/s and person, including recirculated air.

Before reconstruction the ventilation system was in use 24 hours a day.

Results of measurements taken before reconstruction of the ventilation system

The continuous measuring took place over 24 hours. VOC-sampling was carried out for 4 hours.

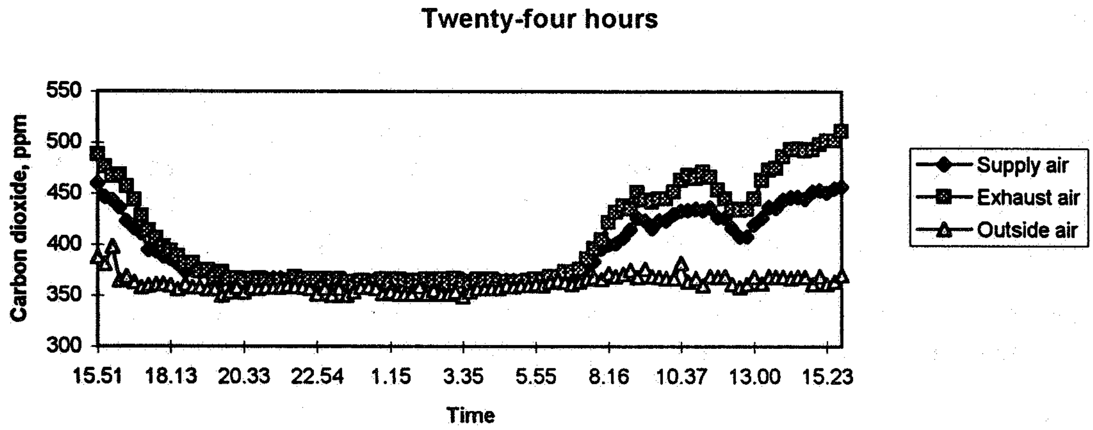

Figure 2, continuous measuring of CO_2 before reconstruction of the ventilation system

Figure 2 shows that the carbondioxide concentration in the building at the time of the measuring was fairly low, max ca 500 ppm. The variation in time agrees well with the activity in the building.

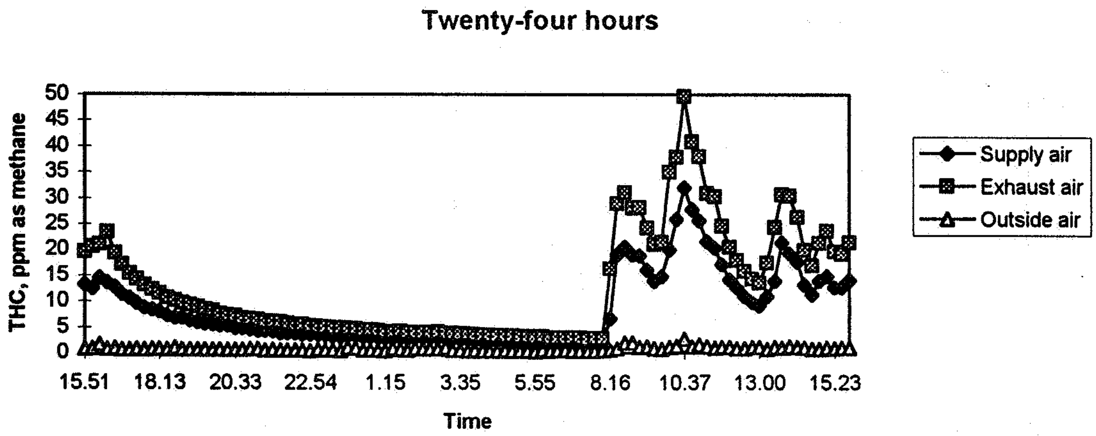

Figure 3, continuous measuring of THC before reconstruction of the ventilation system

Figure 3 shows that the THC varies considerably according to time and that the variation coincides with the variation in carbondioxide. The conclusion is, therefore, that it is not the building itself that is polluting the air but the activity in the building. This is supported by the fact that the THC concentrations diminish during the night when there is no activity in the building. In order to further illustrate this connection the carbondioxide and THC concentrations are shown together in figure 4.

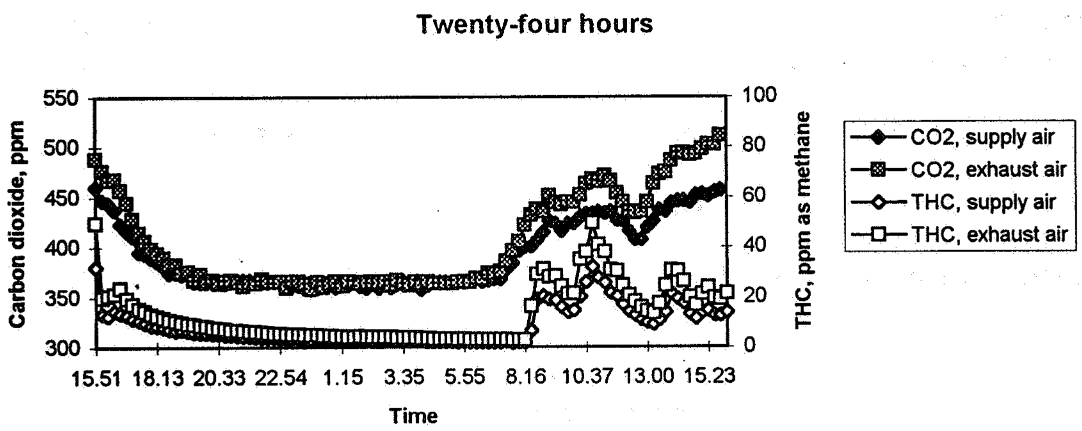

Figure 4, continuous measuring of CO_2 and THC before reconstruction of the ventilation system

Parallel with the continuous measuring VOC-samples were taken. The following results were obtained:

Supply air	3900	µg/m³
Exhaust air	5700	µg/m³
Outside air	73	µg/m³

The concentrations found in the supply air clearly exceed the values considered acceptable for the total VOC in the indoor air, ca 300 µg/m³.

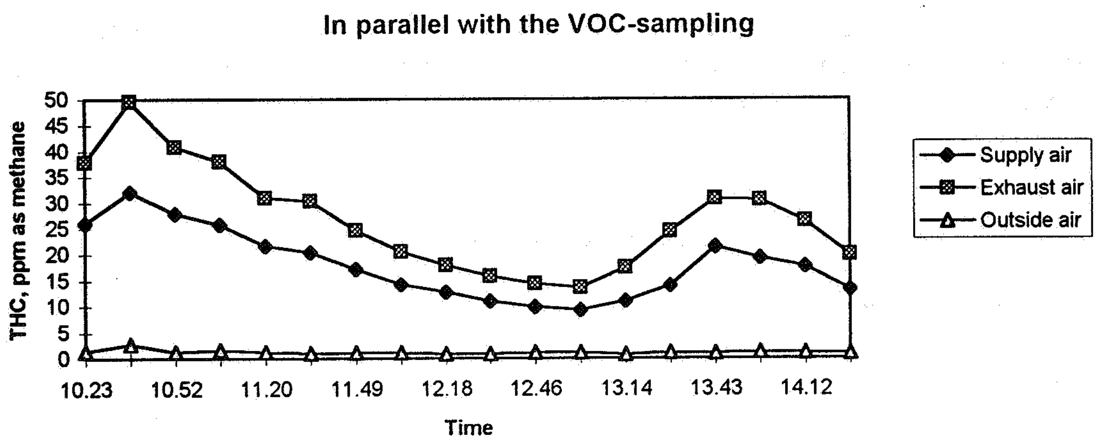

Figure 5, continuous measuring of THC before reconstruction of the ventilation system

The continuous measuring of the THC parallel with the VOC-sampling is shown in figure 5.

Reconstruction of the ventilation system

Based on results obtained it was decided that the device for recirculating the air in the ventilation system should be removed. To make heat recovery possible a heat recovery system based on fluids was installed.

In connection with the reconstruction of the ventilation system the air flow in the building was reduced from 35 to 20 l/s and person.

After the reconstruction the ventilation system continued to be in operation 24 hours a day.

Technical measurements after reconstruction

Measurements were taken in the same way after the reconstruction as before.

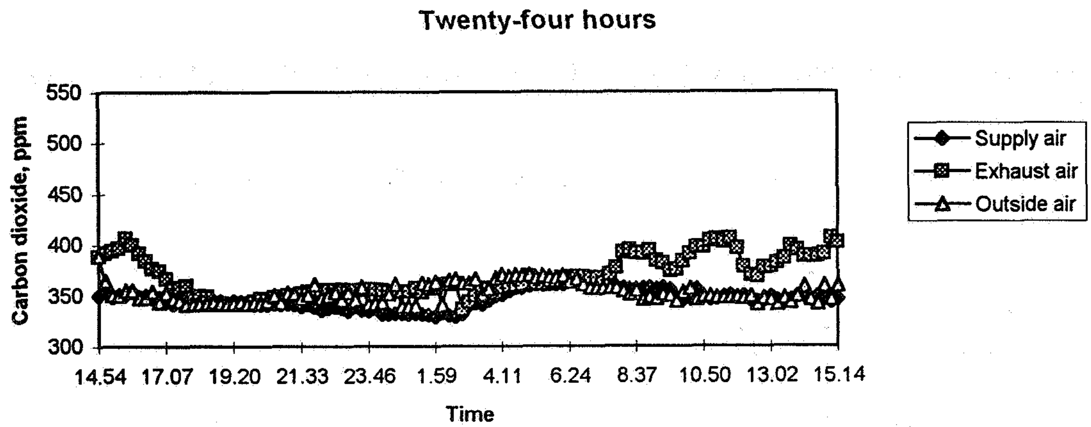

Figure 6, continuous measuring of CO_2 after reconstruction of the ventilation system

Figure 6 shows the carbondioxide concentration after the reconstruction of the ventilation system. The carbondioxide concentration in this measurement is somewhat lower compared with measurements taken before the reconstruction. It also shows that the supply air has the same concentration of carbondioxide as the outside air. In other words there is no recirculation.

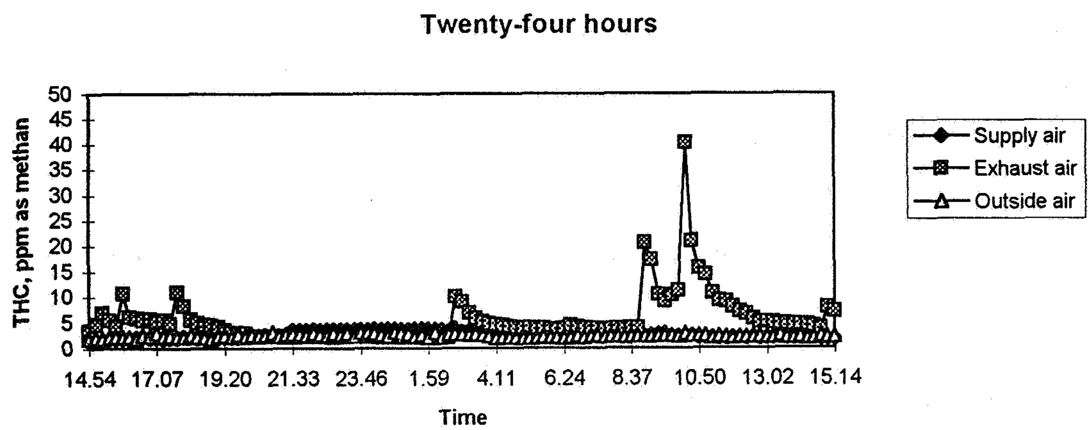

Figure 7, continuous measuring of THC after reconstruction of the ventilation system

Figure 7 shows the THC concentration after reconstruction. Also here the concentration is lower compared with measurements taken before the reconstruction. Finally it shows that the supply air has the same concentration of THC as the outside air.

Parallel with the continuous measuring VOC-samples were taken. The following results were obtained:

Supply air	40	µg/m^3
Exhaust air	1710	µg/m^3
Outside air	10	µg/m^3

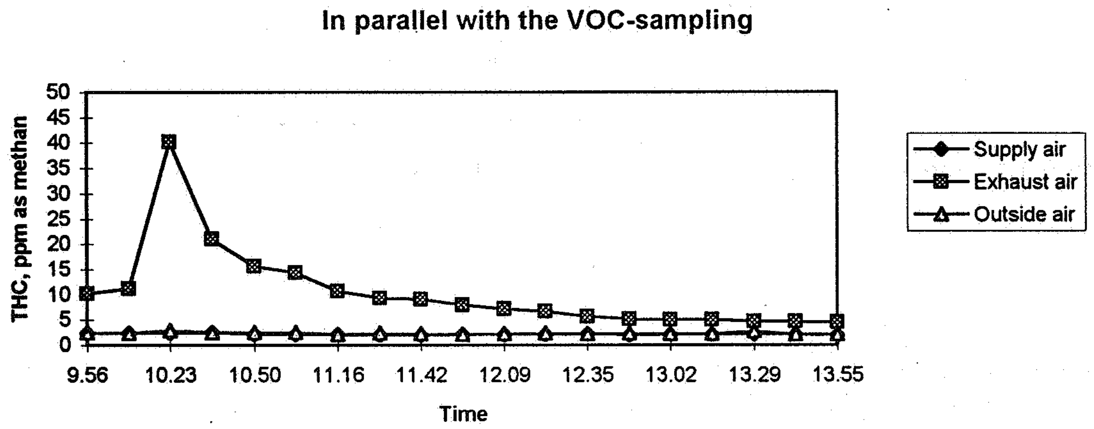

Figure 8, continuous measuring of THC after reconstruction of the ventilation system

The continuous measuring of THC parallel with the VOC-sampling is shown i figure 8.

The total VOC concentration and levels of 2-ethyl-1-hexanol in the indoor air were also tested after reconstruction. The average concentration of total VOC was found to be 155 µg/m^3, maximum value 450 µg/m^3, minimum value 50 µg/m^3. No detectable levels of 2-ethyl-1-hexanol were registered in the indoor air.

Questionnaire

A questionnaire was carried out before and after the reconstruction. The questionnaire used were taken from YMK Örebro.

The following results were obtained:

	Before reconstruction	After reconstruction
Background data		
Number of persons	229	216
Years in present employment	7	8
Proportion of women, %	58	60
Proportion of men, %	42	40
Proportion of smokers, %	19	18

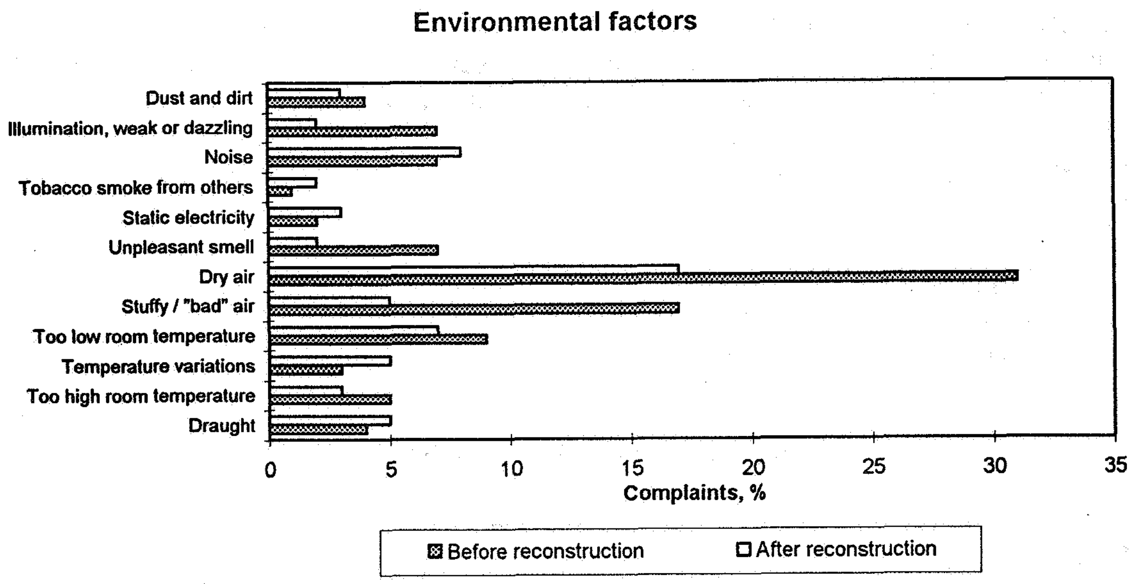

Figure 9, occurance of complaints on environmental factors

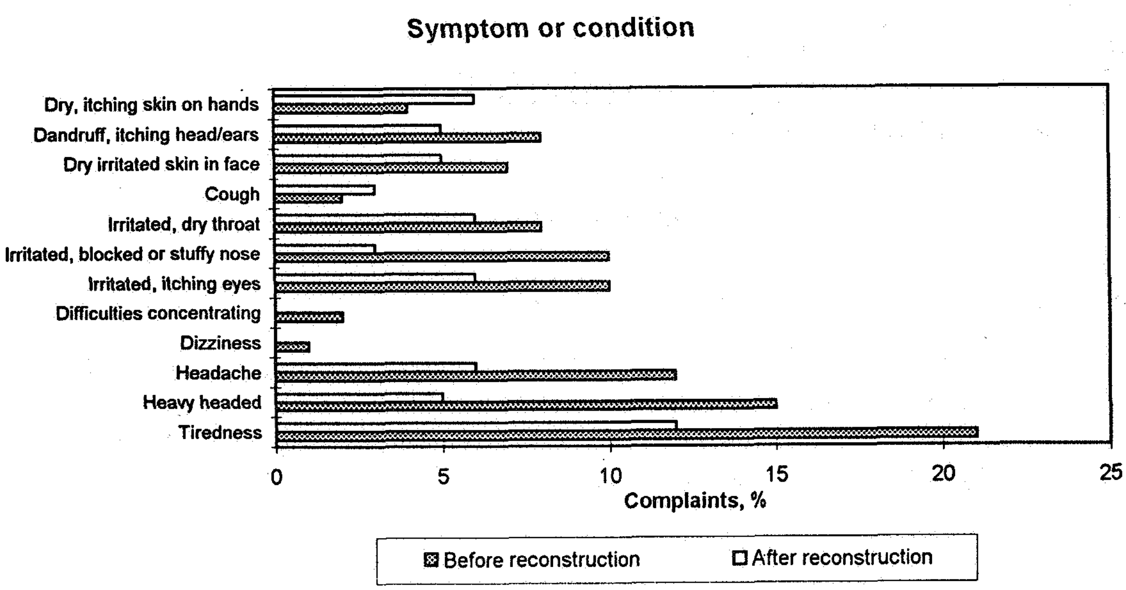

Figure 10, occurrence of self reported health symptoms

As the results show, the number of symptoms decreased after the reconstruction of the ventilation system.

CONCLUSIONS

The investigation shows that the indoor environment was considerably improved by the reconstruction of the ventilation system, in spite of the fact that no measures were taken relating to floor.

The results of the technical measurements were in full agreement with the result of the questionnaire.

The reconstruction of the ventilation system also led to a reduction in energy consumption by the fan motors of ca 50 %. This means that 130000 kWh are saved annually.

The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Numerical assessment of thermal comfort and air quality in an office with displacement ventilation

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SYNOPSIS

Computational fluid dynamics has been used for assessing the thermal comfort and air quality in an office ventilated with a displacement system for a range of supply air conditions. Thermal comfort is predicted by incorporating Fanger's comfort equations in the airflow model. Indoor air quality is assessed according to the predicted contaminant concentration and local mean age of air. The performance of the displacement ventilation system is then evaluated based on the predicted thermal comfort and indoor air quality. It is shown that discomfort in offices with displacement ventilation results more often from unsatisfactory thermal sensation than from draught. It is also shown that optimal supply air conditions of a displacement system depend on the relative position between the air diffuser and occupant. It has been found that increasing the air flow rate improves indoor air quality but may result in local thermal discomfort.

1. INTRODUCTION

The technique of positive displacement has been widely used in Scandinavia in particular for ventilation of industrial buildings. The system has now gradually been employed for office ventilation despite the concern over draught near feet. Numerical predictions [1, 2] show that displacement ventilation generally creates a better thermal environment and average air quality in the occupied zone than traditional mixing systems.

Recently computational fluid dynamics (CFD) has been applied in predicting the thermal environment of displacement-ventilated offices based on the draught risk [3, 4, 5]. In this paper the performance of the displacement ventilation system is assessed according to the predicted thermal comfort and air quality in an office using the CFD technique.

2. METHODOLOGY

Numerical evaluation of the indoor environment is based on the airflow model in conjunction with comfort models.

2.1 Air Flow Model

The airflow model consists of the continuity equation, momentum equation, enthalpy equation, concentration equation and the equation for the age of air together with the k-ε turbulence model equations. For an incompressible steady-state flow the model is represented by the following time-averaged equation:

$$\frac{\partial}{\partial x_i}(\rho U_i \phi) = \frac{\partial}{\partial x_i}\left(\Gamma_\phi \frac{\partial \phi}{\partial x_i}\right) + S_\phi \qquad (1)$$

where ϕ stands for mean velocity component U_i in x_i direction, mean enthalpy H, mean concentration C, local mean age $\bar{\tau}$, turbulent kinetic energy k or dissipation rate of turbulent kinetic energy ε; ρ is the air density; Γ_ϕ is the diffusion coefficient for ϕ; S_ϕ is the source term for ϕ.

A complete description of the model equations and the solution method is given elsewhere [6, 7].

2.2 Thermal Comfort

Assessment of thermal comfort includes overall thermal sensation and draught risk as well as local thermal discomfort from either or both of them.

2.2.1 Thermal sensation

Thermal sensation is evaluated in terms of the predicted mean vote (PMV) and the predicted percentage of dissatisfied (PPD) [8]. These indices are functions of air velocity, air temperature, mean radiant temperature, water vapour pressure, clothing thermal resistance and occupant's metabolic rate.

The air velocity, temperature and water vapour pressure distributions in a room are calculated from the air flow equations. The distribution of mean radiant temperature is attained with the help of a radiation heat exchange model. The procedure to calculate the mean radiant temperature at each grid point is as follows [6]:

i) calculate room surface temperature from the heat balance equations for conduction, convection and radiation;

ii) calculate room surface radiosity based on the room surface temperature and radiation shape factors;

iii) calculate six plane radiant temperatures for each rectangular parallelepiped grid cell;

iv) calculate the mean radiant temperature for the grid cell; it is taken as a weighted mean of the plane radiant temperatures.

2.2.2 Draught risk

Draught risk is assessed according to the draught model [9]. In this model, the percentage of dissatisfied due to draught (PD in %) is associated with air temperature (T in °C), velocity (V in m/s) and turbulence intensity (Tu in %) in the following form:

$$PD = (3.143 + 0.3696\, V\, Tu)\,(34 - T)\,(V - 0.05)^{0.6223} \qquad (2)$$

2.2.3 Local thermal discomfort

In displacement-ventilated offices, local thermal discomfort particularly around occupants' legs and feet presents a potential problem for ventilation system designers and building users. In this study, the local thermal discomfort is assessed according to the levels of discomfort from cold thermal sensation or draught which are expressed in terms of the numbers of grid points around the legs and feet where the predicted comfort levels exceed 10% for PPD [10] and 15% for PD [11] and are denoted as N_{PPD10} and N_{PD15} respectively.

2.3 Air Quality

Indoor air quality is dependent on the quality and quantity of supply air and the distribution of fresh air in the space. It is assessed according to the concentration of CO_2 and the local mean age of air predicted from the airflow equations. The local mean age of air is defined as the average time for air to travel from a supply outlet to any point in a room.

3. RESULTS

Numerical predictions of thermal comfort and air quality are carried out for an office room in summer conditions. The office has dimensions of 4.7 m long, 3.65 m wide and 2.5 m ceiling height. It consists of one external wall and five internal walls including the floor and ceiling. The internal walls are assumed adiabatic. The external wall is insulated to a U-value of 0.22 W/m^2K. It has a double-glazed window of width 2.95 m and height 1.3 m with a U-value of 2.9 W/m^2K and with external shading. It is assumed that the window is closed and the room is ventilated by a displacement system. This is achieved by introducing cool air horizontally from a diffuser installed in the rear or curtain wall. The outdoor air temperature is assumed 30°C and the wind speed 3 m/s normal to the external wall.

The office is occupied by one person, seated by a desk. The simulated occupant generates metabolic heat of 70 W/m^2 of which 30% is considered to be latent heat and produces CO_2 of 4.72×10^{-3} l/s. The moisture production rate by the occupant is estimated from the amount of latent heat. The occupant wears clothes equivalent to a clothing level of 0.6 clo (1.0 clo = 0.155 m^2K/W).

Simulations were performed for 12 sets of supply air conditions. The conditions for the simulations are presented in Table 1. Table 2 shows the predicted thermal comfort, local thermal discomfort and air quality indices.

Figures 1 and 2 show the distributions of the predicted room environmental parameters, thermal comfort and air quality indices in the office for the first and third cases studied. These two cases represent (i) the supply air diffuser in the real wall and the occupant 1.2 m away from the window and (ii) the diffuser installed under the window for the same location of the occupant. As indicated by the velocity vectors, the supply air spreads over the

Table 1 Conditions of supply air and locations of diffuser and occupant for simulations

Case	Supply air					Location	
	Velocity (m/s)	Flow rate (l/s)	Temp. (°C)	RH (%)	CO_2 (ppm)	Diffuser	Occupant
1	0.2	24	20	71	350	D1	O1
2	0.2	24	20	71	350	D2	O1
3	0.1	12	20	71	350	D2	O1
4	0.1	24	20	71	350	D2	O1
5	0.1	24	22	62	350	D2	O1
6	0.1	12	20	71	350	D2	O1
7	0.2	24	20	71	350	D2	O2
8	0.1	12	18	80	350	D2	O1
9	0.15	18	18	80	350	D2	O1
10	0.2	24	18	80	350	D2	O1
11	0.25	30	18	80	350	D2	O1
12	0.3	36	18	80	350	D2	O1

Notes:

Diffuser location:
D1 — rear wall
D2 — curtain wall

Occupant location:
O1 — 1.2 m from window
O2 — mid room length

Table 2 Predicted thermal comfort and air quality in the occupied zone and local thermal discomfort around occupant's legs and feet

Case	V (m/s)	T (°C)	PMV	PPD (%)	PD (%)	N_{PPD10}*	N_{PD15}*	CO_2 (ppm)	$\overline{\tau}/\overline{\tau}_n$#
1	0.028	24.4	0.05	5.29	3.9	0	0	518.1	0.91
2	0.021	24.5	0.07	5.37	3.6	34	10	522.2	0.90
3	0.019	25.9	0.45	9.44	2.3	0	0	694.0	0.95
4	0.021	24.5	0.07	5.35	3.9	21	5	522.0	0.89
5	0.019	25.2	0.25	6.51	2.9	1	0	520.3	0.90
6	0.032	24.9	0.10	5.34	3.5	2	0	694.0	0.96
7	0.024	24.4	0.05	5.27	3.8	0	0	521.4	0.92
8	0.021	25.4	0.30	7.11	2.8	4	0	692.4	0.94
9	0.021	24.5	0.04	5.36	3.2	36	7	581.4	0.92
10	0.021	24.0	-0.10	5.62	3.8	59	17	523.9	0.87
11	0.022	23.5	-0.22	6.46	4.6	83	27	489.4	0.86
12	0.023	23.2	-0.30	7.48	5.7	104	42	466.0	0.85

* Total number of grid points around the legs and feet = 144.
\# $\overline{\tau}$ is the local mean age of air and $\overline{\tau}_n$ is the nominal time constant = net room volume divided by air flow rate.

floor and after reaching the occupant air then rises due to thermal buoyancy. The vertical temperature stratification can be observed in Figures 1b and 2b but the stratification in the occupied zone is not excessive (<2°C) because of small internal heat production. The variation of water vapour pressure with space is also small except for the area around the moisture generation source where the vapour pressure is high. The mean radiant temperature is higher near the 'hot' window than that in the area about one meter away from the window. For both cases, the area near the window is warm whereas the area along the supply air stream is cool (seen from the PMV contours in Figures 1e and 2e); the potential draught risk also exists along the air jet (Figures 1f and 2f). The CO_2 concentration is generally between 400 ppm and 800 ppm and relatively high near the source of generation. The local mean age of air is low near the floor and increases with the height as air flows upwards.

4. DISCUSSION

As seen from Table 2, in terms of predicted comfort indices, both the overall thermal sensation and draught risk are acceptable (PPD < 10%; PD < 15%) under all the supply air conditions investigated. However, local thermal discomfort is also predicted for most of the cases. Besides, the predicted local discomfort often arises simultaneously as a result of cold thermal sensation and draught and the majority of this is caused by dissatisfaction with thermal sensation rather than draught as commonly reported at low temperatures. It is also seen that optimum supply air conditions vary with the relative position between the air diffuser and occupant; Cases 1, 3 and 7 give the least thermal discomfort for three different combinations of air diffuser and occupant locations.

When air is supplied from the rear wall near the floor at a velocity of 0.2 m/s and a temperature of 20°C and the occupant is over three meters away from the air diffuser (Case 1 and Figure 1), there is no risk of local thermal discomfort for the occupant either due to cold thermal sensation or due to draught. This is the result of decreasing momentum and increasing temperature as the supply air diffuses along the floor such that the fresh air reaches a thermally acceptable level near the occupant. As the distance between the diffuser and occupant is reduced, the warmth and movement of fresh air experienced by the occupant deviate from the optimum conditions. Hence, when the diffuser is moved to the curtain wall and the air is supplied at 0.2 m/s and 20°C (Case 2), there exists thermal discomfort around the legs and feet due to draught as well as cold thermal sensation.

For this distance between the diffuser and occupant (i.e. 1.2 m), when the supply air velocity is reduced to 0.1 m/s (Case 3 and Figure 2), the thermal discomfort around the legs and feet disappears but there is some discomfort due to warm thermal sensation at head level because of the reduced air flow rate and the thermal stratification. When the air flow rate is doubled by increasing the diffuser opening area (Case 4), the warm thermal discomfort at the head level is avoided but the cold thermal discomfort around the legs and feet reappears. When the supply air temperature is increased to 22°C (Case 5), the cold thermal discomfort at foot level is almost eliminated but again the head will experience slightly warm feeling. It is clear that at this distance between the air diffuser and occupant a very delicate adjustment of supply air conditions is required to achieve a satisfactory indoor thermal environment and that even so this may not be a good solution to maintain the comfort level because of unavoidable disturbance to the surroundings in practice. One way to alleviate the local thermal discomfort problem is to make use of a chilled ceiling system. The chilled ceiling can be achieved, for example, by installing cooling panels onto the ceiling [12].

Figure 3 shows the predicted indoor thermal environment for the same supply air conditions as Case 3 but with a chilled ceiling system covering 50% ceiling area (Case 6). In this simulation, it is assumed that the area with the chilled ceiling is at a temperature of 20°C while the rest of the ceiling is adiabatic. It is seen that warm discomfort at head level is avoided although slight cool discomfort at foot level occurs. The PPD level around feet slightly exceeds 10% at two out of 144 grid points, 10.8% at one point and 13.9% at another point. This improvement of thermal comfort, compared with Case 3, results from reduced air temperature and mean radiant temperature at head level. The air temperature and mean radiant temperature in the occupied zone are reduced from 25.9°C and 25.9°C without the chilled ceiling to 24.9°C and 24.4°C with the chilled ceiling, respectively. The mean radiant temperature decreases with the distance from the floor whereas air temperature increases; one stratification offsets the other. As a result, the distribution of the comfort indices in the space is fairly uniform apart from the areas near the air diffuser and hot window. Hence the occupant will not have the feeling of warm head.

For the same location of the diffuser, the discomfort can also be relieved or eliminated if the occupant moves away from the window. For example, when the occupant is supposed to sit at the mid room length, the local thermal discomfort is avoided (Case 7 and Figure 4) and the occupant will feel as comfortable as in Case 1, even though the air is supplied at 0.2 m/s and 20°C.

The predicted CO_2 concentration and local mean age of air in the occupied zone decrease with the increase of air flow rate as seen from Cases 8 through 12. For example, when the air flow rate increases from 12 l/s (Case 8) to 24 l/s (Case 10), the CO_2 concentration is reduced from 692.4 ppm to 523.9 ppm and the normalised age of air from 0.94 to 0.87. Hence, indoor air quality can be improved by increasing the air flow rate for displacement ventilation. This is not always true for conventional ventilation systems where short circuiting of supply air could sometimes occur so that contaminants could be trapped in the breathing zone. However, increasing the amount of supply air requires more energy use and for displacement ventilation increases the possibility of local thermal discomfort. Therefore, a proper balance is needed between the requirement for good indoor air quality and that for acceptable thermal comfort when setting supply air conditions.

5. CONCLUSIONS

This numerical investigation shows that in displacement ventilation optimal supply air conditions vary with the distance between the air diffuser and occupant other than the cooling load and load distribution. The distance between the diffuser and occupant should preferably be greater than two meters for maintaining a comfortable indoor thermal environment.

A better indoor environment is achievable through fine tuning of the conditions of displacement ventilation in combination with a chilled ceiling system than is provided by conventional ventilation systems.

When designing a displacement ventilation system and determining supply air conditions the need for good air quality and the need for acceptable thermal comfort should be carefully considered in order to minimise the energy requirement for ventilation.

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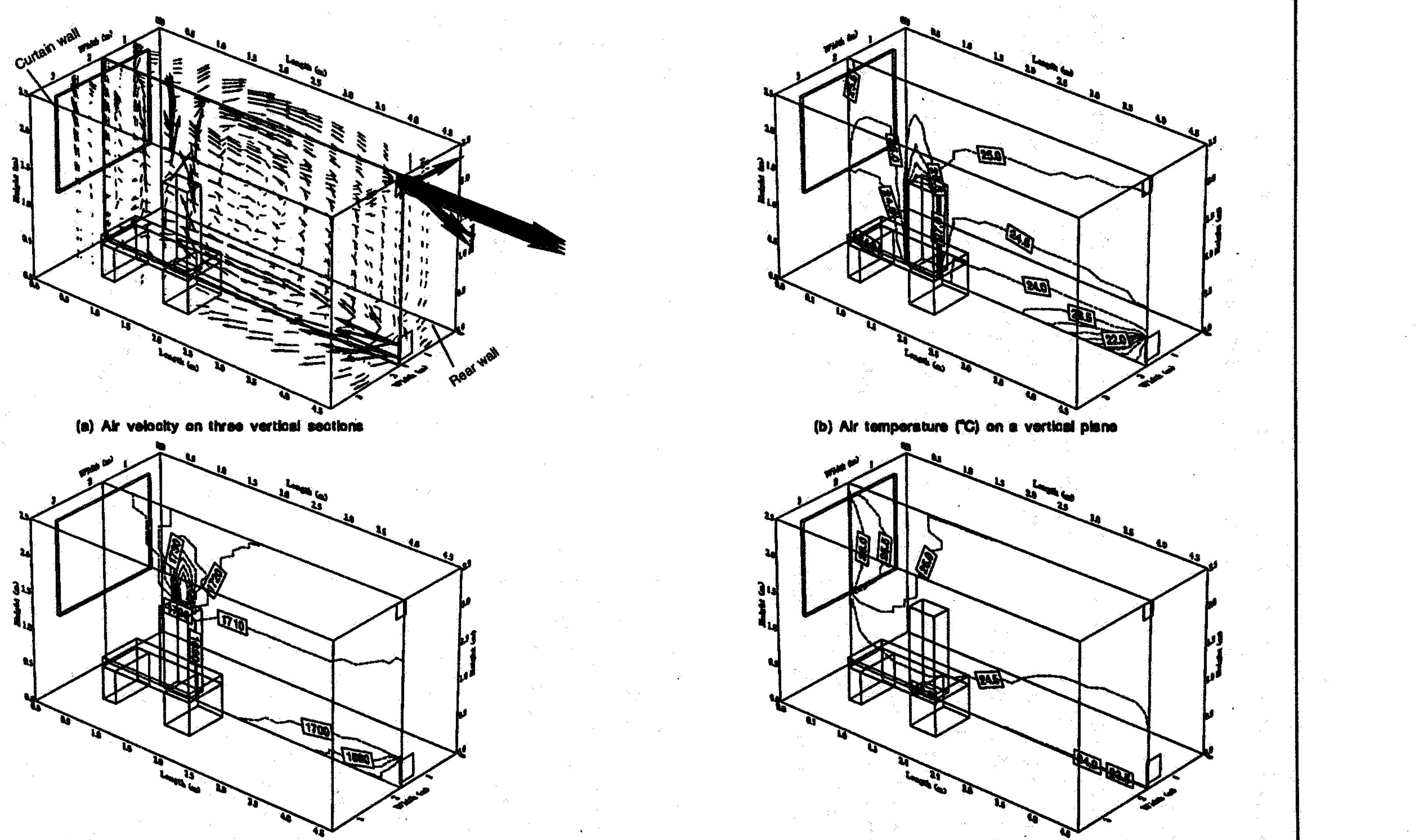


(a) Air velocity on three vertical sections

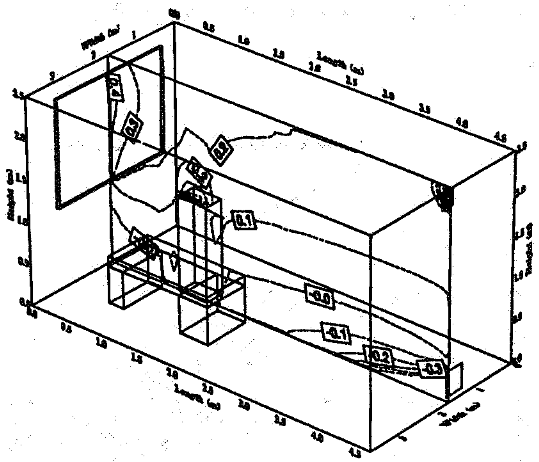
(b) Air temperature (°C) on a vertical plane

(c) Vapour pressure (Pa) on a vertical plane

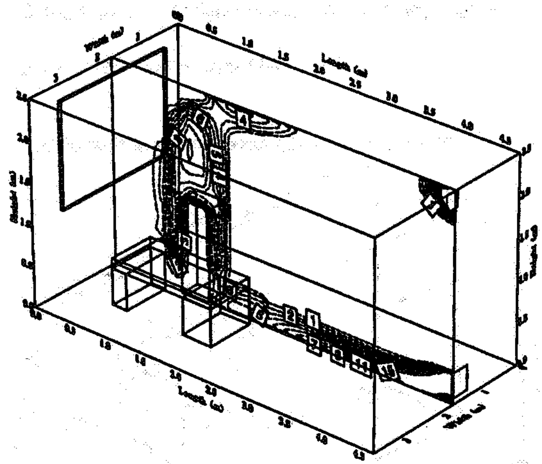
(d) Mean radiant temperature (°C) on a vertical plane

Supply air: V = 0.2 m/s; T = 20.0°C; RH = 71%; CO2 = 350 ppm

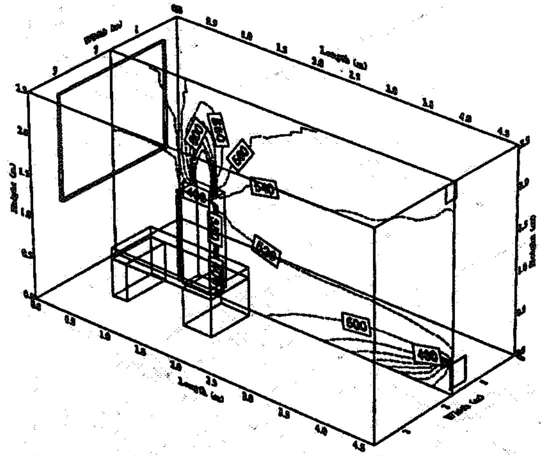
Fig. 1 Predicted environmental parameters, thermal comfort and air quality in the displacement-ventilated office



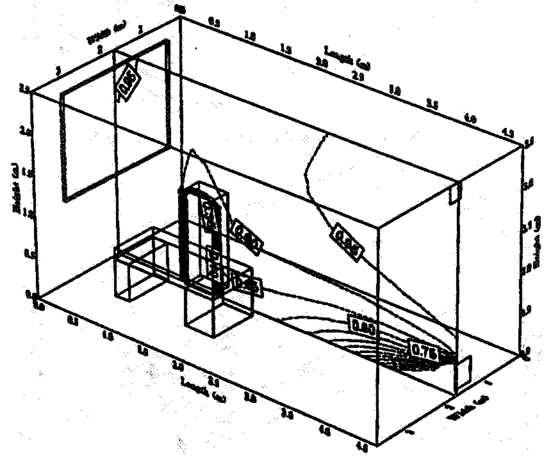
(e) Predicted mean vote



(f) Percentage of dissatisfied due to draught (%)



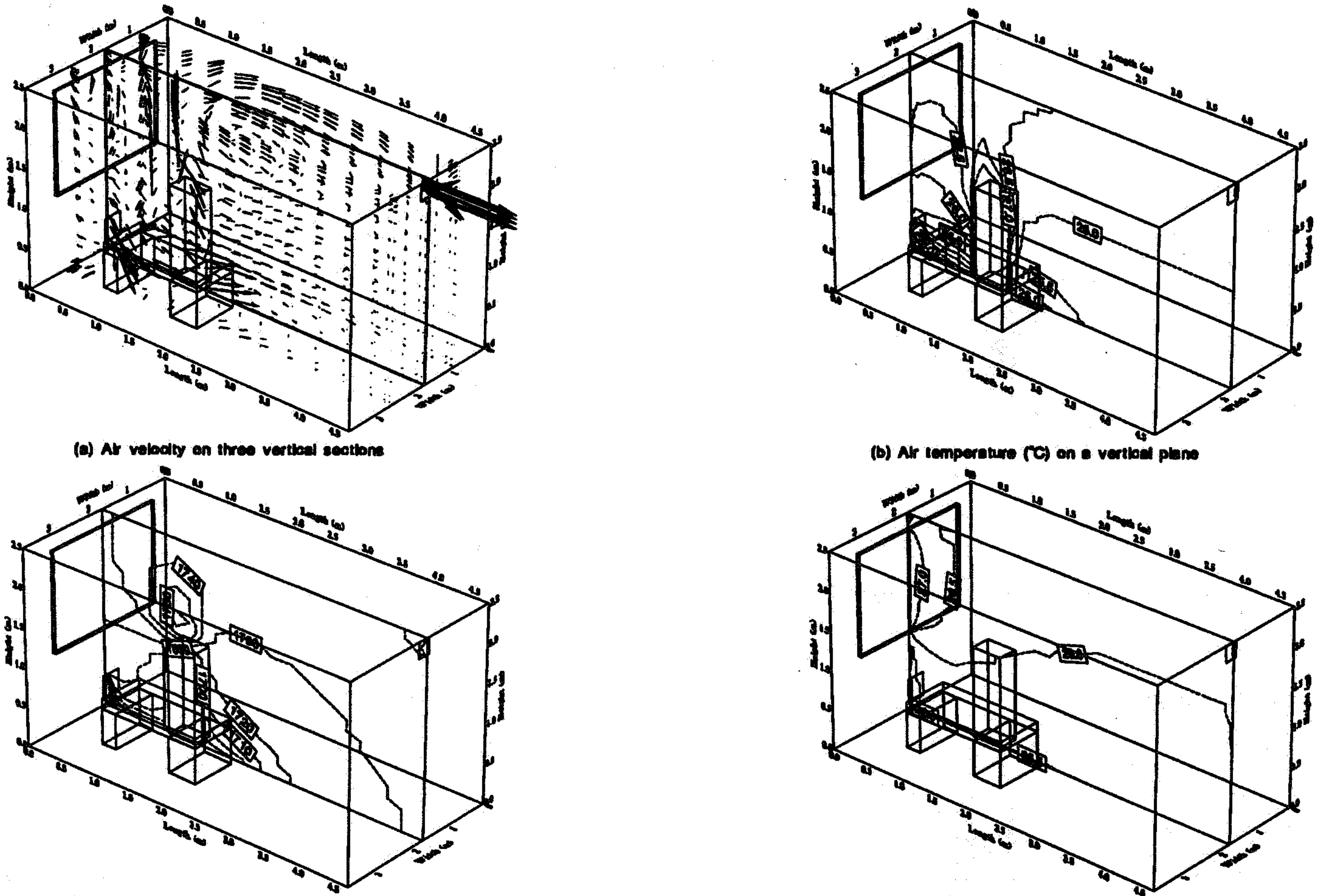
(g) CO2 concentration (ppm)



(h) Normalised local mean age of air

Supply air: V = 0.2 m/s; T = 20.0°C; RH = 71%; CO2 = 350 ppm

Fig. 1 Continued



(a) Air velocity on three vertical sections

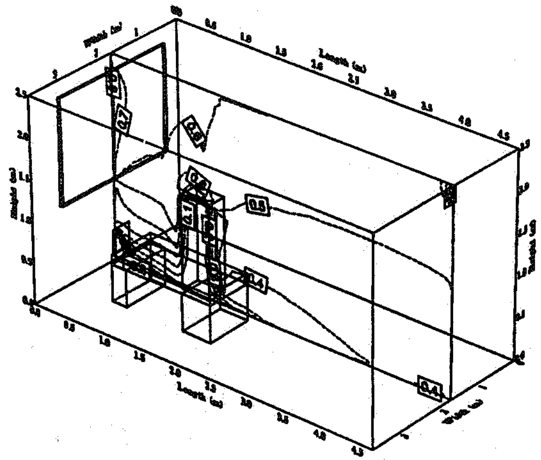
(b) Air temperature (°C) on a vertical plane

(c) Vapour pressure (Pa) on a vertical plane

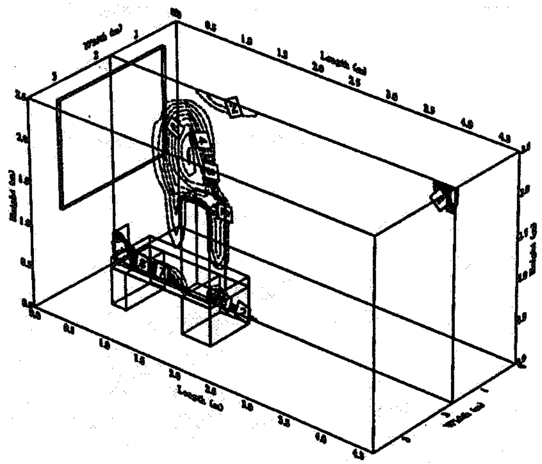
(d) Mean radiant temperature (°C) on a vertical plane

Supply air: V = 0.1 m/s; T = 20.0°C; RH = 71%; CO2 = 350 ppm

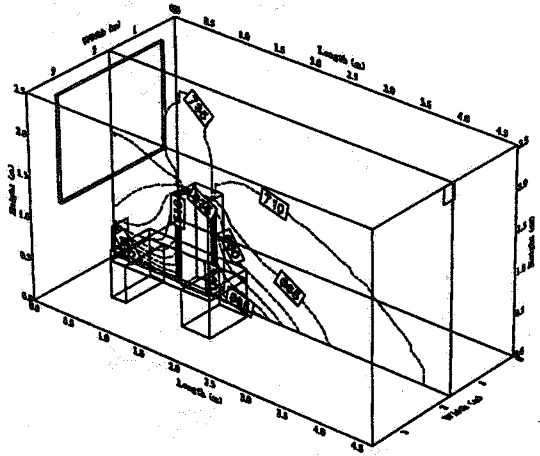
Fig. 2 Predicted environmental parameters, thermal comfort and air quality in the displacement-ventilated office



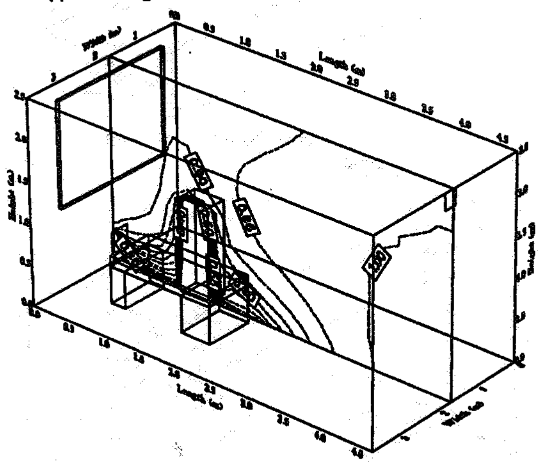
(e) Predicted mean vote



(f) Percentage of dissatisfied due to draught (%)



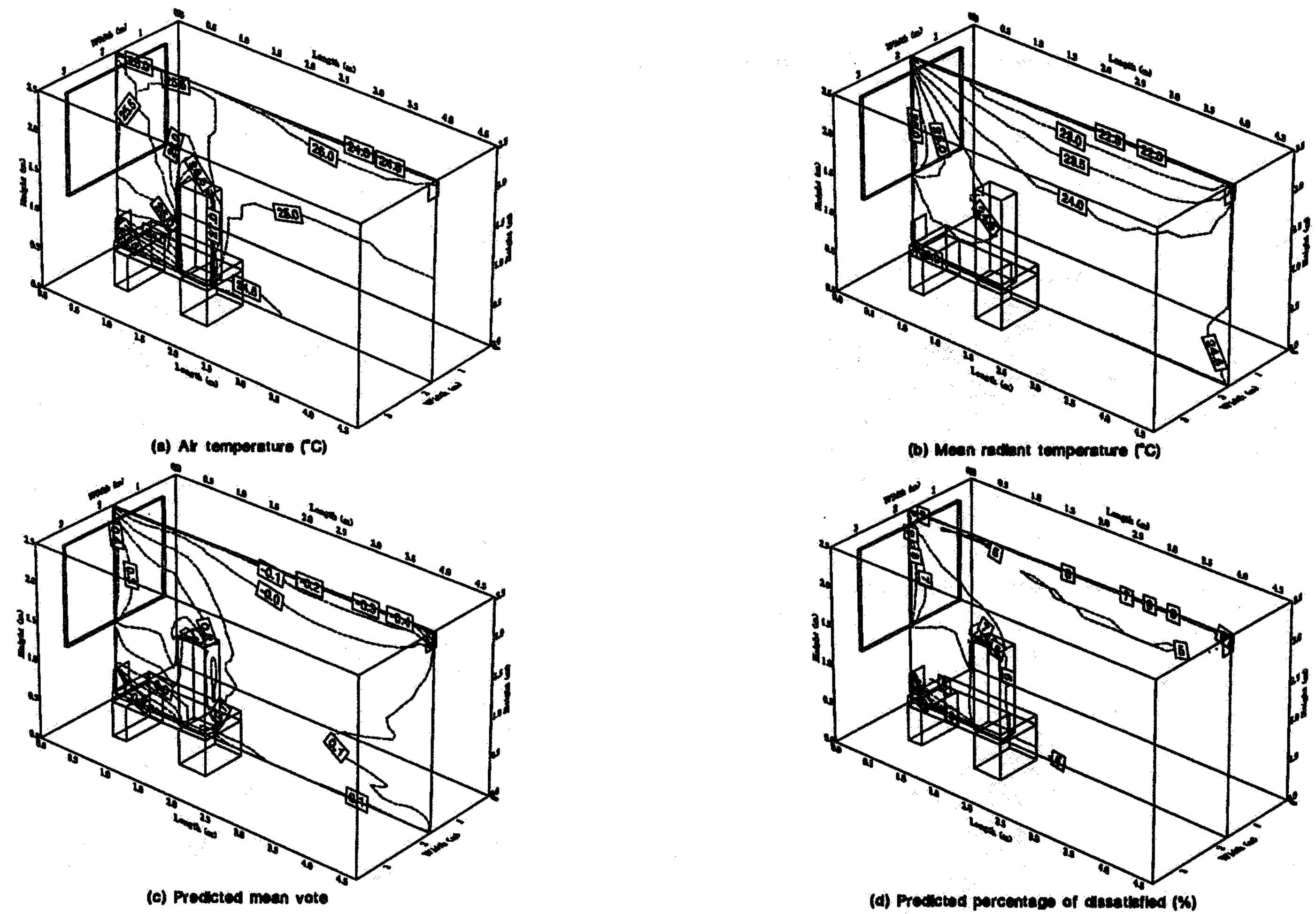
(g) CO2 concentration (ppm)



(h) Normalised local mean age of air

Supply air: V = 0.1 m/s; T = 20.0°C; RH = 71%; CO2 = 350 ppm

Fig. 2 Continued



(a) Air temperature (°C)

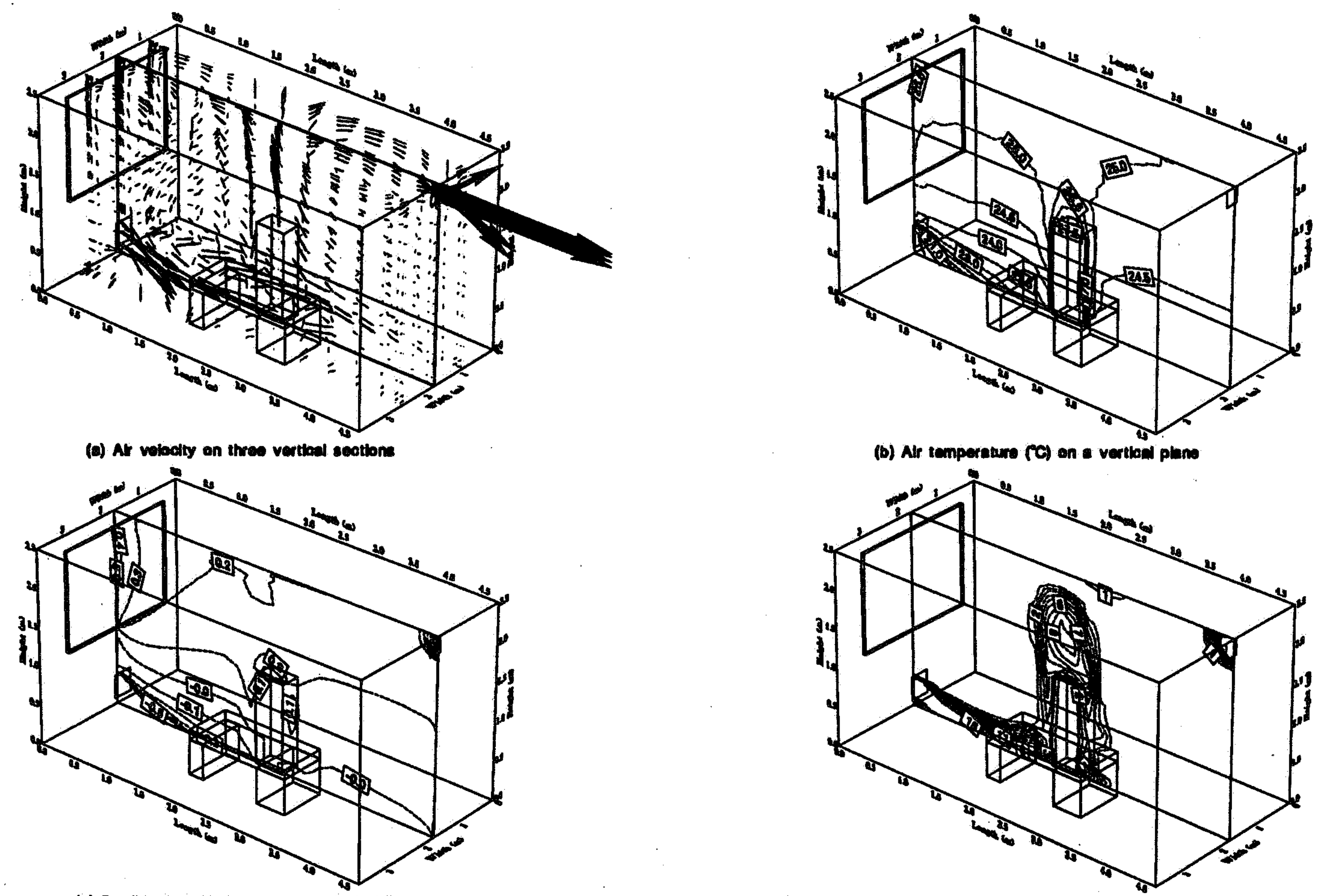
(b) Mean radiant temperature (°C)

(c) Predicted mean vote

(d) Predicted percentage of dissatisfied (%)

Supply air: V = 0.1 m/s; T = 20.0°C; RH = 71%

Fig. 3 Predicted thermal comfort on a vertical plane in the displacement-ventilated office with 50% chilled ceiling



(a) Air velocity on three vertical sections

(b) Air temperature (°C) on a vertical plane

(c) Predicted mean vote on a vertical plane

(d) Percentage of dissatisfied due to draught (%) on a vertical plane

Supply air: V = 0.2 m/s; T = 20.0°C; RH = 71%

Fig. 4 Predicted air flow and thermal comfort in the displacement-ventilated office

The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

The Role of Infiltration for Indoor Air Quality - A Case Study in Multifamily Dwelling Houses in Poland

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THE ROLE OF INFILTRATION FOR INDOOR AIR QUALITY - A CASE STUDY IN MULTI-FAMILY DWELLING HOUSES IN POLAND

SYNOPSIS

Multifamily buildings with natural ventilation are still being built and exploited. Such buildings are often equipped with individual gas-fired water heaters located in windowless bathrooms. It implicates the possibilities of improper gas exhaust as a result of the decrease of infiltration, what could be sometimes even harmful for the occupants' health.

Based on the numerical simulations, analysis of ventilating air flows in typical multifamily dwelling house will be carried out. It will be shown that effectiveness of natural ventilation in particular flats depends not only on the active factors determining infiltration phenomenon (like wind and temperature difference) but is also strongly connected with flat location inside the complex structure of a building.

1. INTRODUCTION

Air infiltration is one of the elements that determine the process of ventilation of dwelling houses. In particular, it refers to the buildings with natural (gravity) ventilation.

The knowledge of air flows within the complex structure of a building is of interest from a few points of view. First - it is important to ensure a determined minimum rate of air change between the inside and the outside , necessary from the hygienic point of view. Secondly - the air inflow from the outside of a building (infiltration) is a prerequisite for proper course of natural ventilation process. At the same time the identification of infiltrating air flows enables to predict the infiltration caused heat losses what makes it possible to calculate heat requirements for a whole building. Thirdly - of a significance is also a question of thermal comfort. One must remember not only discomfort resulting the underheating of infiltrating air but also the directions of air flows, the latter question being of less consideration though of an importance. For example, it often occurs that the standard air change rate is satisfied, however some part of the air flows into the flat from the staircase. In this case, it is hard to admit this air change rate in the flat to be satisfactory.

Very often demands are made to minimize infiltration of the air by tightening its natural places of inflow, namely windows. The effect of such steps is, of course, positive as for heat losses, but on the other side it is conflict with postulate of correct ventilation of flats, especially in the case of natural ventilation.

2. POSSIBILITIES OF IDENTIFYING VENTILATION AIR FLOWS

To achieve information on the course of the process of ventilating flats it is required to identify air flows in the space under examination as completely as possible. The most natural and definitive way to do it seems to be measurement. Beside the methodological difficulties one must be conscious of fairly considerable randomness of the results so obtained: they are possible to be achieved in general in a limited range; it is hard to guarantee the simultaneity of measurements in different spaces; the picture obtained represents the situation of one measuring moment, i.e. for given, instantaneous conditions that force the flows.

In the case of natural ventilation we may distinguish two main driving forces of the process: wind pressure on a building and heat buoyancy. While heat buoyancy, which is determined by air temperature difference between the inside and the outside of a building, is a slow varying (e.g. in an hour cycle) quantity, then wind has definitely random character respecting both direction and velocity. It particularly refers to compactly developed lands where natural turbulent wind structure is additionally disturbed in flows around the all sorts of obstacles surrounding an examined building.

The practical way achieve full information about the ventilation air flows in multi-zone object is computer simulation. The method of mathematical modelling and numerical simulation is reliable as much as the accuracy is kept with which the phenomenon is described and the parameters are settled. The advantage of this way compared with empirical method consists, among others, in the possibility of declaring variable climate parameters, building location and location of calculation zones - according to current needs.

3. NUMERICAL PROCEDURE AND PARAMETERS OF SIMULATION

The calculation results presented below come from the simulation made by my own numerical programme TRANSVEN. This procedure allows to simulate ventilation process in multi-zone object in the quasi-dynamic course and with the time step of 1 hour. The climate parameters, which are the input of the programme, were appointed on the base of many years' meteorological observations for the south-west region of Poland [1].

Basing the analyses of ventilation efficiency on the time runs seems to be more useful than calculation at steady climate data. Then, it becomes possible to analyse the variation of ventilation conditions in a given time period with the variation of climate parameters. Especially, it concerns the variation of velocity and direction of wind. Such type simulations

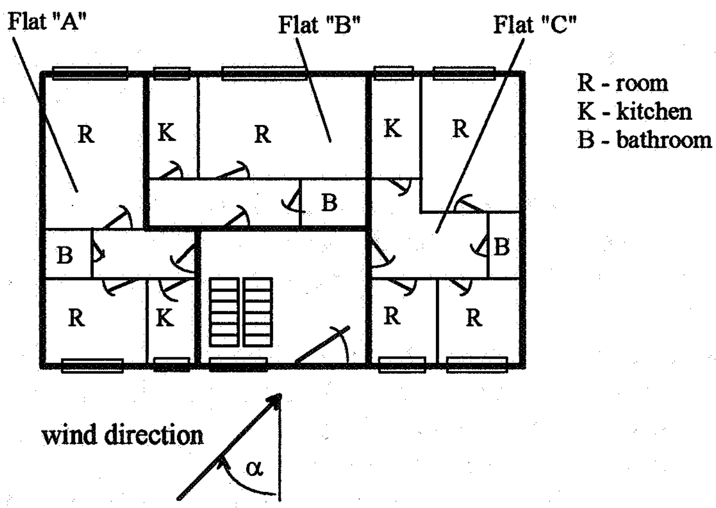


Fig. 1. Plan of a storey of the examined building.

open up good possibilities for analysing a great amount of individual cases, delivering material for generalization of statistical nature.

The calculations were made for a 5-storyed dwelling house with 3 flats on each floor. All flats were equipped with gas-fired water heaters placed in bathrooms without windows. Air change in the flats could occur thanks to individual gravity ventilation ducts and also exhaust ducts. The simulations were carried out for climate parameters representing the transition period of winter and spring (3 weeks of March) for the average temperature of this period approximately -1.3°C and wind velocity 3.3 m/s.

4. SIMULATION RESULTS AND DISCUSSION

While making a design of ventilation system of a building one should meet the need of essential from hygienic point of view air change rate. It is commonly assumed to be approximately 1 air change rate per hour for dwellings. The realization of this postulate mostly means to design the ventilation ducts that make such air flows possible.

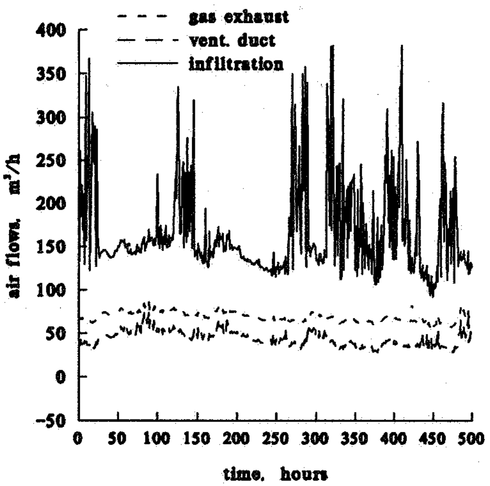


Fig. 2. Ventilation air flows in the flat "A" on the 1st floor.

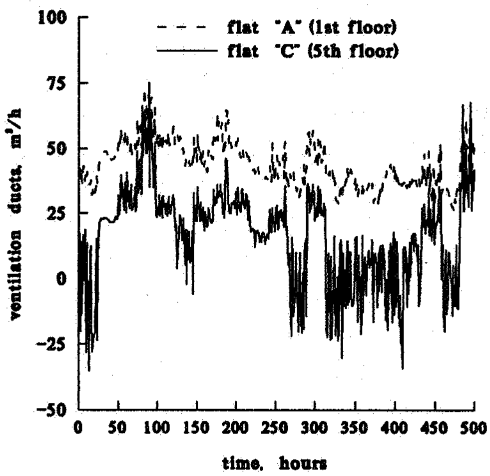


Fig. 3. Comparison of ventilating air flows in the flat "A" and "C".

For the analysed building cubage (approx. 4400 m^3), in the time period the calculations dealt with the air change rate in the whole building amounts, on average, to the level 0.5 air changes per hour. This value is thought to be a minimum air change rate (in hygienic aspect) [2].

However, owing to complicated path of air flows it turns out that air infiltration is markedly varied in a complex structure of individual storeys. This fact can be illustrated by comparison of ventilation air flows in the flat "A" on the first floor (Fig. 2) and in the flat "C" on the fifth floor. Outdoor air infiltration, which decides the air change, is in the flat "A" about 2.5 times greater than in the flat "C" at the same climate conditions. At the comparable cubages of both those types of flats, it implicates the difference of twice as large air change rate being absolute too little. The effect of such low level of infiltration is respectively worsened operation conditions of ventilation and exhaust ducts. It can be noticed that in some time periods there occur backflows in ventilation ducts in the flat "C" (Fig. 3). They are particularly dangerous for occupiers' health and even life. A situation like this takes places mainly in the flat "C" perpendicular, appearing more rarely on its lower floors.

Observations of the utilization of such type buildings confirm the above presented simulation results. In Polish foothills, where foehn type winds occur in some seasons, cases of poisoning of the occupiers took places many a time in consequence of inadmissible carbon monoxide concentration in bathrooms with gas-fired water heaters. Death cases were also stated.

For finding those climate conditions in which backflows in ducts can occur the flow rates in ventilation ducts of the flat "C" were settled in function of variable wind velocity and ambient temperature. Fig. 4 shows space picture of this variation in a wide range of climate parameters. It can be noticed that the most disadvantageous operation conditions take place at the wind velocity 4÷7 m/s and ambient temperature 0÷5 °C. These values of climate parameters are typical for the transition period (winter - spring) in heat season in Poland.

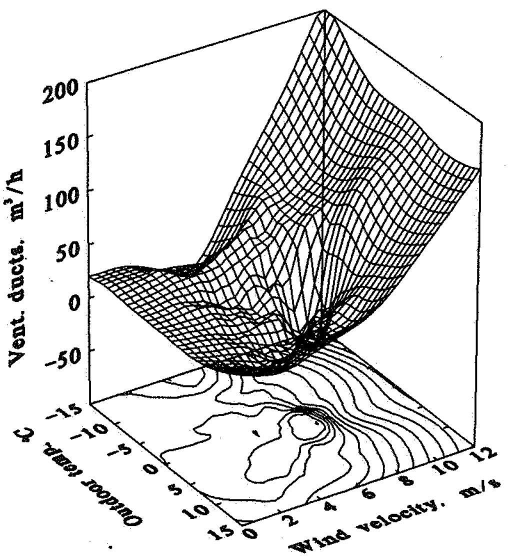


Fig. 4. Spatial picture of changeability of air flows through the ventilation duct in the flat "C" on the 5th floor.

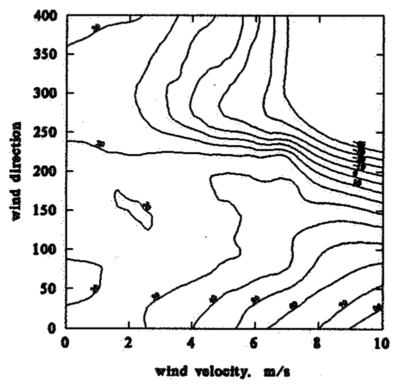
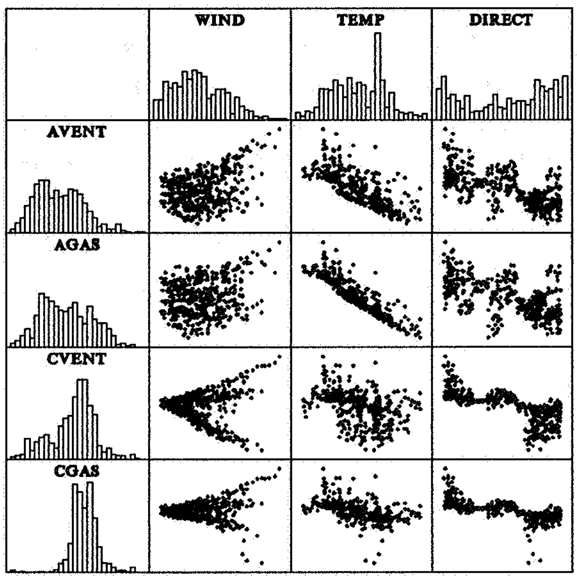


Fig. 5. Isolines of the air flow rate in the flat "C" as a function of wind direction and velocity.

What is also important here is the location of a given flat in building structure and resulting from it the orientation of windows towards the predominant directions of wind. Unadvantageous influence of the wind velocity above 4 m/s is confirmed, but backflows in ducts occur at the wind directions from the range of 180 ÷ 360 ° by starting counting from the building facade. The phenomenon of backflows in ducts practically does not exist in the "A" type flats, although the total length of their window gaps is comparable to the case of the flats "C", thus the conditions of infiltration from this point of view are similar. Such an effect is explained by inner structure of both types of flats; a major role is played here by the location of kitchen in which an inlet to the ventilation duct is placed.

The influence of the layout of individual flats particularly becomes apparent when making statistical analyses. While having a great amount of simulations at one's disposal (in the case under consideration - 450) it is possible to search a correlation between ventilation air flow rates and variable climate parameters. The results achieved are illustrated by scattering matrix (Fig. 6). In the "A" type flats, of considerable importance is the influence of the



Legend:
A[C]VENT - air flows in vent. duct in the flat "A"[C];
A[C]GAS - air flows through the gas exhaust in the flat "A"[C];
WIND - wind velocity;
TEMP - outdoor temperature;
DIRECT - wind direction.

Fig. 6. Comparison of wind and outdoor temperature effect on ventilating air flows.

temperature difference between the outside and the inside of a building; the correlation coefficient is here from -0.65 to -0.75. Whereas the influence of wind direction and velocity is of less significance (correlation coefficient 0.4 ÷ 0.6). In the flat "C", what is primary for duct operation is wind direction (correlation coefficient -0.65 ÷ -0.8).

The simulation results point to good agreement of the proposed model of infiltration with actually occurring air flows in existing buildings [3,4].

The benefit of the simulation works as presented below seems to be the possibility of obtaining quantitative relationships between ventilation driving forces of infiltration and resulting ventilation air streams. It may be useful in better predicting ventilation efficiency.

5. CONCLUSIONS

Natural air change in multi-family buildings essentially depends on infiltration what means, among others, that it is formed spontaneously and is imperfectly predictable.

Providing ventilation ducts of a suitable flow rates cannot be the only way to satisfy proper conditions for air change in natural ventilation. The results reported above show that as early as initial design stage one should take account of location of a building in a given area considering its wind rose. Of a great importance is also the layout of flats within a storey and the location of window openings in definite walls of a building. It is necessary to resign applying gas-fired water heaters because it is not always in the system of natural ventilation that one can be sure of the entire exhausting of flue gases.

An important aspect of quantification of flow processes and natural exchange of the air is the matter of wind modelling as being one of the deciding driving factors of those phenomena. Development of simulation models should tend to dynamical calculations, the climate generators being chosen adequately. Averaging the results yielded in this means seems to be more reliable than that in the case of the calculations made for steady climate conditions.

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   COW, No.8, 1992.

The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Effectiveness of Various Means of Extract Ventilation at Removing Moisture from a Kitchen

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Effectiveness of various means of extract ventilation at removing moisture from a kitchen

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Synopsis

A kitchen is one of the major moisture producing areas in a dwelling. In order to prevent condensation and mould growth the relative humidity should not be too high. This paper describes a set of experiments comparing methods of kitchen ventilation and their effectiveness at moisture removal.

The three methods of extract ventilation were:

1. A mechanical extract fan of extract rate 60 ls^{-1}
2. A passive stack ventilation system
3. An open-flued gas boiler

The extract fan was only activated during moisture production, whereas the other (passive) methods of extract ventilation were permanently operational. Parameters were varied to assess their influence on both the ventilation rate of the kitchen and on the distribution of moisture about the dwelling. The effect of the kitchen door being either open or closed was investigated as well as the effect of an air-brick and trickle vent.

The three types of extract ventilation operated at differing rates throughout the test cycle. High ventilation rate for a short period, slow continuous ventilation, or an intermittent heat driven cycle all gave similar average kitchen ventilation rates. All three methods of extract ventilation can provide a satisfactory solution to moisture control in kitchens in a temperate climate. Closing the door while cooking is as important as providing extract ventilation because it prevents a large proportion of the moisture from migrating upstairs.

1. Introduction

The kitchen is one of the major moisture producing rooms of a house. In order to prevent problems of mould growth the level of relative humidity should not exceed 70% for prolonged periods [1]. Extract ventilation reduces the humidity level and is therefore desirable; it is included in Approved Document F to the Building Regulations [2]

The Approved Document F coming into force on 1 July 1995 provides for both mechanical extract ventilation and passive stack ventilation or appropriate open-flued combustion appliances.

Open-flued combustion appliances and passive stack ventilation systems (PSV) are similar in that they both have a permanent passage for airflow from inside to outside via a vertical duct. They both maintain their flow because of the temperature difference between outside and inside, and the effect of the wind blowing on the roof terminal. The combustion appliance also provides a heat input which increases the flow rate up the flue. A PSV has the advantage that its opening is at ceiling height, meaning that warmer moist air near the ceiling can be ventilated away more easily.

Work by BRE on the flow rates in PSV systems and their potential for ventilation of kitchens and bathrooms has previously been presented [3].

This paper describes some experiments in which three types of kitchen extract ventilation were compared with the case of no purpose provided ventilation. The three ventilation types were: mechanical extract fan; PSV; open-flued gas boiler. Additional variables tested were the kitchen door and an air vent, which both had two conditions: open and closed.

2. Experimental Design and Set-up

Similar houses at either end of a terrace were used for the experiments. The kitchen of House One had a PSV system (diameter 150 mm) and an extract fan (flow 60 l/s). House Two had an open-flued gas boiler with a 125 mm diameter twin walled stainless steel flue.

The experiment was designed to measure the ventilation rate of the kitchen and the change in moisture content of the kitchen and other rooms under different ventilation regimes. To ensure the data collected was not dominated by the weather, the six different experiments were repeated at least six times. Each test lasted for around 24 hours. During an experiment in which the fan was being tested, the PSV was sealed and vice versa.

At the start of each test 4 to 5 litres of water was boiled off into the kitchen in the space of two hours. This level of moisture production is a severe case and represents a heavy two hour cooking and washing session in the kitchen. The six ventilation cases were:

Condition	House One	House Two
1	No Purpose Provided Ventilation	No Purpose Provided Ventilation
2	PSV open	Flue open, boiler off
3	Fan on during moisture input	Flue open, boiler on
4	Door open, (no other ventilation)	Door open, (no other ventilation)
5	Door open, PSV open	Door open, flue open
6	Door open fan on for moisture input	Door open, boiler on

Table 1: The six ventilation conditions

Conditions were matched so that similar experiments went on at the same time in the two houses. 'Flue open, boiler off' was considered analogous to 'PSV open' (passive ventilation); 'flue open, boiler on' was the equivalent of 'fan on' (forced ventilation). The extract fan was only activated for the two hour period of moisture input, whereas the other ventilation measures were unchanged over a 24 hour test period. This assumes that people operate extract fans during the period of moisture generation, and turn them off as soon as the cooking or washing is finished. The reality may be that fans are used less than this [4].

The ventilation rate of the two kitchens was monitored continuously using a constant concentration of the tracer gas sulphur hexaflouride, SF_6. The tracer gas concentration in the hall and one bedroom was also monitored, to give the rate of air flow to each of those rooms. The relative humidity and temperature in these rooms and at two other locations in the house as well as outside were monitored every 10 minutes. Wind speed and direction were also logged.

3. Example results

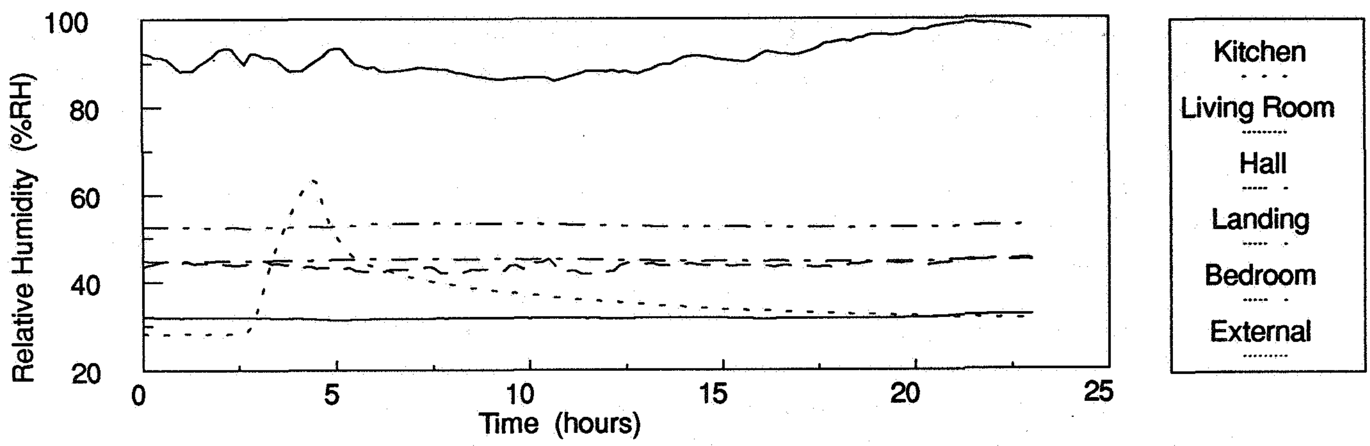

Figure 1: Plot of relative humidity vs time in House Two with the kitchen door closed

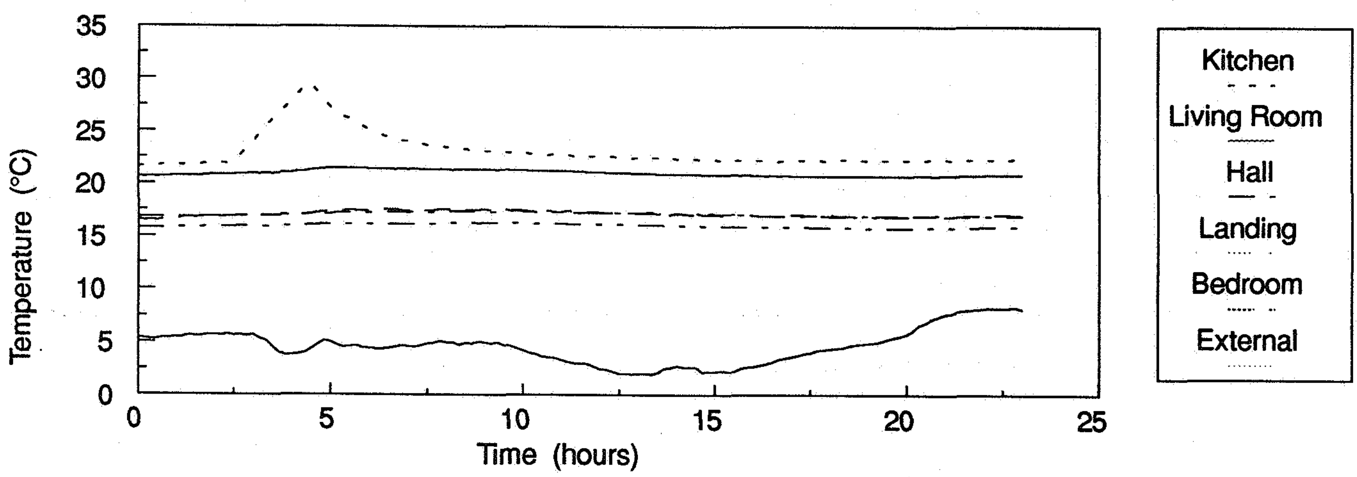

Figure 2: Plot of temperature against time in House Two with the kitchen door closed

Figure 1 shows a typical development of the relative humidities (RH) over time for a condition 2 test in House Two (kitchen door closed, flue open). Figure 2 shows the temperatures for the same test. These both show the experiment progressing, with both RH and temperature rising as the water is boiled on the cooker, reaching a peak as the two hour heat input is finished. From this time on the RH and temperature then decrease as hot moist air is replaced by cooler, dryer air from outside.

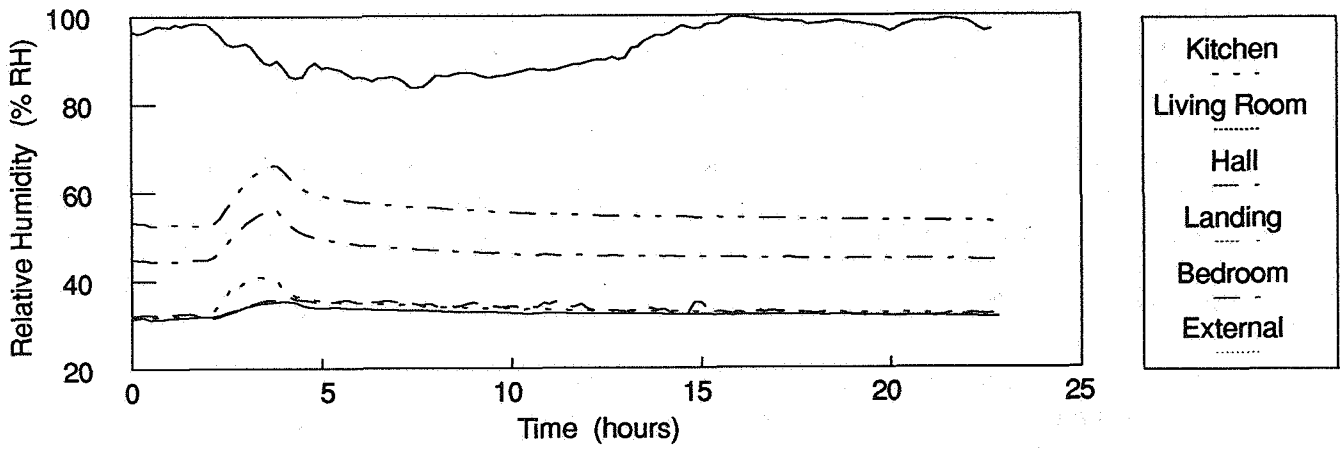

Figure 3: Plot of relative humidities vs time in House Two with the kitchen door open

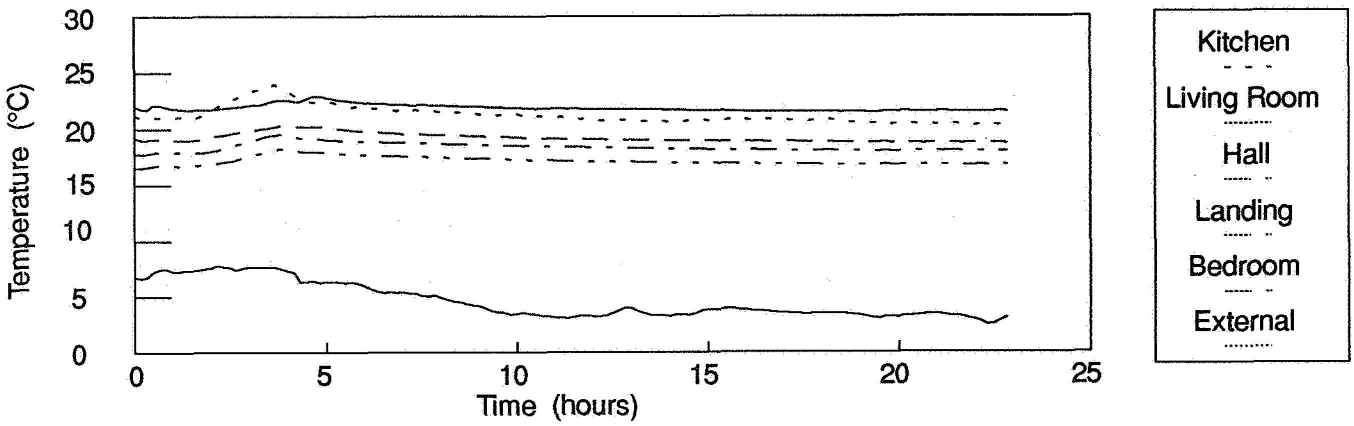

Figure 4: Plot of temperatures against time in House Two with the kitchen door open

Figure 3 shows the impact that opening the kitchen door has on the relative humidity levels in the house. Compared to figure 1 it shows that much more moisture reaches upstairs with the door open, which results in high relative humidities in the bedroom. Figure 4 shows that opening the kitchen door raises the temperatures throughout the house, but lowers that in the kitchen. Because the temperature is lower upstairs than downstairs in the house the RH is actually higher in the bedroom than in the kitchen, meaning that in this case the greatest risk of mould growth is not in the moisture producing room.

Buffer effect

When moisture is produced, the relative humidity of the atmosphere increases. When the relative humidity in a room exceeds the surface relative humidity of the walls, furnishings and fabric of the room, moisture begins to be absorbed into these surfaces, and this restrains the rate of increase of the RH of the air. After moisture production stops, the relative humidity of the room air will fall as the high moisture content air is ventilated away and dryer air replaces it. When the RH falls below the surface RH of the fabric of the room then moisture is desorbed out of the surfaces and into the air. Hence the fabric of the room acts as a buffer, reducing the rate of change of the humidity of the room air.

Because of this an extract method which is switched off when cooking finishes will not reduce the moisture levels by as much as one which continues afterwards. An extra period of extraction would remove water given off by the room fabric as the air RH decreases.

4. Analysis

In a paper to be presented elsewhere [5] the data are analyzed using analysis of variance. This method shows which changes have the greatest effect on the ventilation rate and moisture levels, using statistics. It shows that while an extract fan clearly gives the highest ventilation rate when it is running, over a 24 hour period there is little difference between a fan, a passive stack or an open flue in terms of mean air change rate or effective removal of moisture. The open or closed state of the kitchen door emerges as the key variable. In this paper we have chosen to look at the same data using transfer indices.

4.1 Transfer indices

A transfer index is a measure of the amount of contaminant released at one point found at another. The higher the concentration and/or the longer it stays in the atmosphere at that point, the higher the transfer index. Generally, the further from the source that a sample is taken the lower the transfer index for any contaminant because it is found in lower concentrations. Because of the different methods used to introduce water and SF_6 the transfer indices T_{pn} for the point p due to a source at n are defined differently [6].

For the introduction of a set volume of pollutant V_{cn}(water) it is the integral of the concentration C_p at the sample point n over time, divided by the input volume:

$$T_{Pn} = \frac{\int C_p(t) \cdot dt}{V_{cn}}$$

The units of T_{Pn} are s/m^3.

When a constant concentration of tracer gas is maintained in one room there is no need for the integral, so the transfer index is defined by:

$$T_{Pn} = \frac{C_p(\infty)}{q_n}$$

q_n is the input rate of pollutant
C_p is the equilibrium concentration at point p

SF_6 was used as a tracer gas to measure the ventilation rate of the kitchen. Unlike water vapour, SF_6 does not get absorbed into the fabric of the room, and therefore the levels of SF_6 in a room give the amount of air that has come from the source room. The transfer index for SF_6 is a measure of this, for a given experimental set up.

Comparing the transfer indices of SF_6 and water vapour gives information about the absorption of water by the building. If no water was absorbed by the building the transfer indices would be the same. In the following tables the average transfer indices for tests under different conditions are given. They were calculated by taking the average of all the transfer indices for all of the tests for the same condition.

House 1	Door	Kitchen	Living	Landing	Bedroom	Hall
SF_6 TI	closed open	218.9 52.0	not measured	not measured	41.1 45.2	44.9 41.9
Water TI	closed open	77.5 16.9	8.9 12.8	not measured	6.2 15.3	7.2 12.6
House 2						
SF_6 TI	closed open	245.3 70.7	not measured	not measured	65.9 61.6	65.7 57.6
Water TI	closed open	69.0 10.3	2.9 7.5	2.9 14.1	2.7 15.9	2.0 8.1

Table 2: Average transfer indices when no additional ventilation was provided

Looking first at the Transfer indices (TI) for the kitchen, the ratio of the door closed value to the door open case shows that the kitchen ventilation rate increases just over 4 times in House One, and by 3½ times in House Two, when the door is open.

Despite the greater ventilation rate of kitchen One, more moisture is detected in that room's atmosphere than in kitchen Two. This is probably due to the difference in the internal surfaces of the two kitchens with more moisture being absorbed into the surfaces in House Two's kitchen. The walls in house Two were only plasterboard and had not been painted, whereas those in House One had been painted. House One had lino on the kitchen floor which absorbs less moisture than the carpet in House Two's kitchen.

In both houses the transfer index for SF_6 in the hall and bedroom is approximately the same when the door is open and when it is closed. When the kitchen door is closed less SF_6 is released to keep the kitchen at a concentration of 10 ppm than when it is open. Because the kitchen ventilation rate is lower less SF_6 leaves the kitchen and the SF_6 concentration in other rooms is low (generally between 1 and 2 ppm). However as a proportion of the SF_6 released into the kitchen that found in the other rooms is the same.

The amount of water vapour which reaches other rooms is far greater when the kitchen door is open. This is because any moisture which leaves the room while the door is closed has to leave via small cracks in the fabric. With small cracks there is more chance of the water vapour coming into contact with a surface and being absorbed. When the door is open there is the possibility of bulk air movement, meaning that moist air will pass through the doorway without coming into contact with any surfaces.

Comparing methods of ventilation: kitchen door closed

House 1	Ventilation	Kitchen	Living	Landing	Bedroom	Hall
SF_6 TI	none psv open fan on	218.9 84.1 68.0	not measured	not measured	41.1 5.4 14.6	44.9 6.3 15.5
Water TI	none psv open fan on	77.5 41.4 37.2	8.9 1.4 3.2	not measured	6.2 2.2 2.6	7.2 1.8 2.4

Table 3: Comparing ventilation measures, kitchen door closed, House One Transfer Indices

The kitchen ventilation rate increases when both the psv and the fan are used meaning that the SF_6 transfer indices go down. There is also a large drop in the moisture measured in the kitchen and other rooms when extract ventilation is provided. The SF_6 transfer index for the bedroom and hall when the fan is used is larger than when the psv is used. This is because the fan only changes the kitchen conditions for its two hour period of operation.

The fan and PSV both reduce the amount of moisture in the atmosphere. That in the kitchen is reduced by about half and in the other rooms by even more. The extract fan and PSV reduce the water vapour transfer indices by similar amounts. This is because the extract fan is operating at the time when the moisture is being produced. For a fan operating for two hours, and a contaminant being produced continuously (as modelled by the SF_6) then the PSV would have removed a greater quantity of it than the fan.

House 2	Flue	Kitchen	Living	Landing	Bedroom	Hall
SF6 TI	none flue open boiler on	245.3 75.9 68.3	not measured	not measured	65.9 6.3 6.9	65.7 6.6 7.0
Water TI	none flue open boiler on	68.9 37.2 37.2	2.9 1.0 1.2	2.9 1.2 1.8	2.7 1.2 1.8	2.0 0.9 1.1

Table 4: Comparing ventilation measures, kitchen door closed, House Two Transfer Indices

The open-flued gas boiler behaves in a similar way to the psv. Turning the boiler on whichshould increase the draught up the flue does not seem to have a significant effect on the amount of contaminant in the rooms. The effect from this could be masked by weather and experimental error, but these data suggest it is not a significant factor.

Comparing methods of ventilation: kitchen door open

In the door open case the transfer indices for the fan and the PSV are almost exactly the same, for both SF_6 and moisture. This suggests they have equal effect on the average ventilation rate over the test period. It is also significant that the SF_6 TI is reduced by both the fan and PSV compared to no ventilation, but that the water vapour TI is not. This probably results from the heat input of the moisture source, causing the moisture to leave via a plume near the ceiling making it less affected by changes in the room ventilation.

House 1: Door open	Ventilation	Kitchen	Bedroom	Hall
SF_6 TI	none	52.0	45.2	41.9
	psv open	40.0	31.4	29.4
	fan on	40.7	31.8	31.6
Water TI	none	16.9	15.3	12.6
	psv open	17.5	15.6	12.0
	fan on	17.5	15.5	12.6

5. Conclusions

Experimental studies, and analysis of their results using transfer indices has shown that:

Each of an extract fan, passive stack ventilation and open flue are equally effective means of ventilating a kitchen to reduce the build up of excess levels of moisture.

Whether the kitchen door is open or closed makes the largest difference to the movement of moisture within the house. It is more important than the choice of ventilation system and none of the ventilation methods is able to prevent the movement of moisture with the door open. As a result kitchen doors should be kept closed while large amounts of moisture are being produced in the kitchen to prevent the migration of moisture upstairs where it will condense on cold surfaces.

Future work could investigate the optimum time for fan use, the impact of open windows, and the effectiveness of humidity controlled fans and passive stack devices.

6. Acknowledgements

This work was supported by the Building Regulations Division of the Department of the Environment (DoE), and published with their permission. Views expressed are those of the authors, not necessarily of the DoE.

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Water Evaporation of 5 Common Indoor Plants Under Various Climate Conditions

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Water evaporation of 5 common indoor plants under various climate conditions

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Synopsis

In recent years plants have increasingly become an integral part of building interior design. Greened office space and large enclosures can provide a better human environment not only because of psychological reasons. Due to photosynthesis, plants interact with the "aerial" environment. Water evaporation affects room air humidity and temperature. Water uptake rates of five common plants in typical indoor climate conditions have been studied. Water evaporation of these plants can now be predicted in architectural design studies. Simulation of a typical office room in summer and winter show that intensive planting can significantly increase air humidity. As a conclusion, this extra humidity should be removed by natural ventilation in summer while in winter it helps to provide comfortable air conditions. The study shows that the effect of indoor plants' water evaporation on air temperature is little.

The plant family

Dizygotheca véitchii "Castor", common name false aralie; petit boux calédonien

Family Araliaceae with approx. 20 species, originated in Australia and some Pacific Islands. Requires a minimum air temperature of 18°C and a minimum illuminance level of 1200 lux.

Dracaena deremensis "Warnekii", common name Warneckeii Dracaena

Family Agavaceae with 40 species known originated on the Canary islands, in the tropical and subtropical parts of Africa, Asia and the south-east Asian island. Warnekii Dracaena originated in east Africa.
Requires a minimum air temperature of 13°C and a minimum illuminance level of 400 lux.

Ficus benjamina, common name ficus

Family Moraceae with approx. 2000 species. Species cultivated as indoor plants are originated in South-East Asia.
Requires a minimum air temperature of 12°C and a minimum illuminance level of 1000 lux.

Hedera hélix, common name English ivy

Family Araliaceae with approx. 7 species originated in Europe and Asia.
Requires a minimum air temperature of 18°C and a minimum illuminance level of 1000 lux.

Philodendron imperial, common name philodendron

Family Araceae with approx. 275 species originated in the tropical forests of Latin America.
Requires a minimum air temperature of 15°C and a minimum illuminance level of 500 lux.

Mechanism of transpiration

The plant loses taken up water by transpiration through the stomata. Less than 1% is converted into carbohydrates. Transpiration occurs at the foliage skin boundary where atmospheric air is not saturated. Intercellular water diffuses through the stomata and then enters the atmospheric boundary layer in gaseous form. The thickness of this layer depends on occurring air movements. It varies from a few millimetres in calm to zero in wind situations. The boundary layer structure also depends on the foliage hair structure.

The 1st Fick diffusion equation describes the mechanism of water transpiration

$$\frac{dm}{dt} = -D \cdot q \cdot \frac{\partial c}{\partial x}$$

Transpiration (dm/dt) most intense in conditions with high concentration gradients ($\partial c / \partial x$) and large surface (q), the diffusion constant D being component specific.

The quantity of water transpiration is controlled by the stomata which can adjust their surface according to the momentaneous plant's need. With fully opened stomata, foliage can lose up to 50-70 % of the evaporation from an open surface. One has to take into account that all total stomata surface covers only 1-2% of the foliage.

Water transpiration can be determined by measuring water uptake with a scale. As less than 1% of the water uptake is converted into carbohydrates, this method is satisfying, especially during short survey periods.

Experimental set-up

The test cabins

3 test cabins (phytotrons) at the Swiss Federal Research Station for Fruit-Growing, Viticulture and Horticulture were used for our measuring campaign during 3 weeks in June 1993. Each phytotron was equipped with 2 sets of our plant family making a total of 10 plants. Plant arrangement provided similar light exposure. The climate in each phytotron was controlled in order to obtain constant conditions by the use of heating/ cooling/ humidification/ dehumidification devices. The experiment was set-up in order to get uptake figures of each plant depending on the main parameters air temperature, relative humidity and illuminance.

During the measuring campaign, temperatures in each phytotrons were set-up at constant levels (15°C; 20°C; 25°C). Relative humidity (40%; 60%; 80%) and illuminance (dark; level I; level I+II) were kept constant for a "day period" of 12 hours followed by a 12 hour "night period".

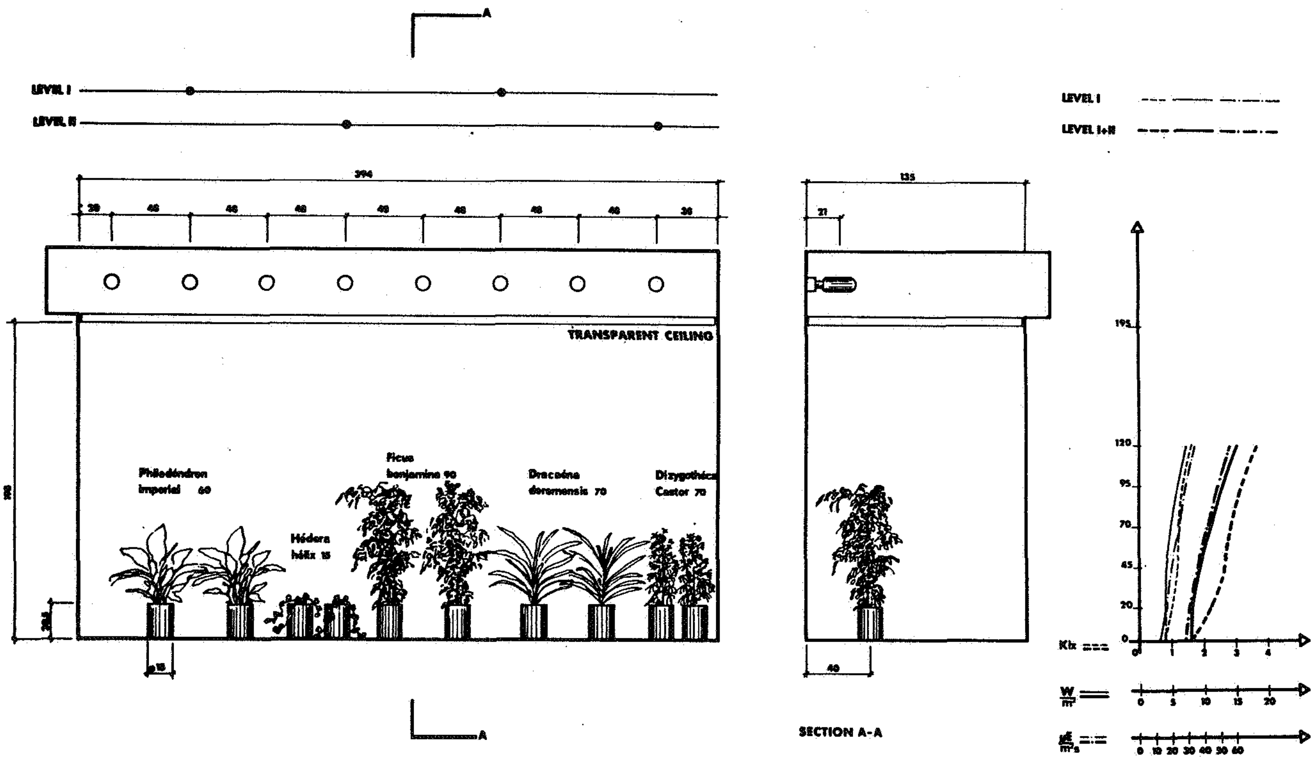


Fig. 1 Plant arrangement and light levels in the phytotrons

Light quality

The phytotrons were equipped with high-pressure metal halide lamps (EYE-Clean Ace MT 400 DL) with its favourable spectral energy distribution. The lamps were arranged so that horizontal light level distribution was even.

Hydroculture planting system

Our plants were placed in geometrically identical hydroculture pots supported in Leca, a lightweight expanded clay aggregate of pale brown colour. For irrigation and nutrition, water enriched with N: 60 mg/l, P: 22 mg/l, K: 96 mg/l, Mg: 16 mg/l, Fe: 7 mg/l was used.

Water uptake measurement

During 12-hour periods where climate was kept constant, water uptake of each plant pot was monitored. To obtain net figures of the plant without the pot's evaporation, uptake of "unplanted" pots of the same size and filled with the same amount of Leca granulate were included. Measurements were carried out with the precision scale Mettler-Toledo PM 6000. This scale has a range of 6000g with an accuracy of 0.01g. Pots had a maximal weight of 2400g and values were monitored with an accuracy of 0.1g.

Leaf area and leaf area index (LAI)

Total leaf area of each individual plant was measured in order to obtain comparable results. A LI-COR areameter which measures light transmission reduction was fed with the plant's total foliage by a conveyer. The following table shows average leaf area values:

Dizygotheca véitchii "Castor"	approx.	0.300	m^2
Dracaena deremensis "Warnekii"		0.200	
Ficus benjamina		0.750	
Hedera hélix		0.110	
Philodendron Imperial		0.450	

The characteristics of a plant's foliage density can be described with the leaf area index (Leaf Area Index = LAI):

$$LAI = \frac{total\ leaf\ area}{plant\ ground\ area}$$

Values of LAI ranges from 0.45 (ground cover plants) to 14 (bushes). The plants we used had the following LAI values:

Dizygotheca véitchii "Castor"	approx.	5
Dracaena deremensis "Warnekii"		0.7
Ficus benjamina		2
Hedera hélix		-
Philodendron Imperial		1.4

Results

The following diagrams (fig 2 to 6) show net water uptake rates in g/m^2 h depending on air temperature, air relative humidity and illuminance. Each value is the arithmetical average obtained from 2 plants exposed to the same climate. Figure 7 shows average values of the whole plant family.

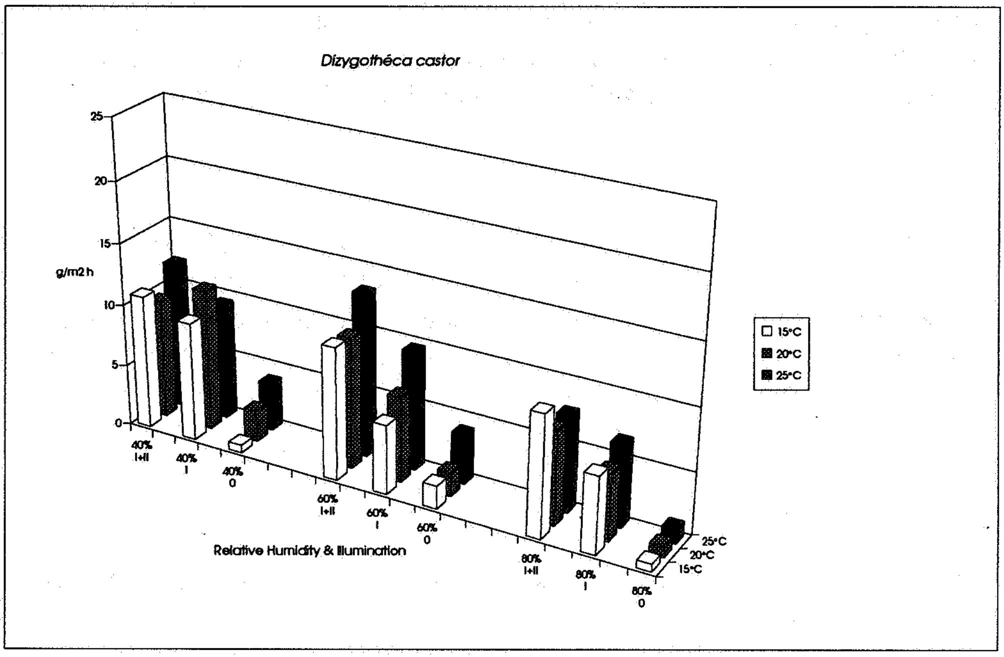


Fig. 2

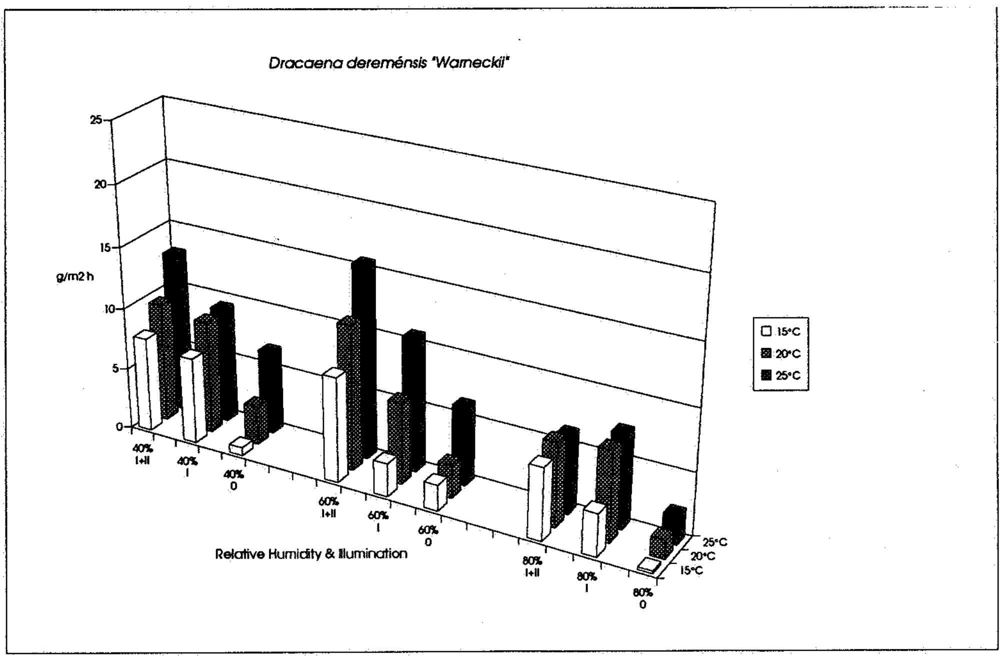
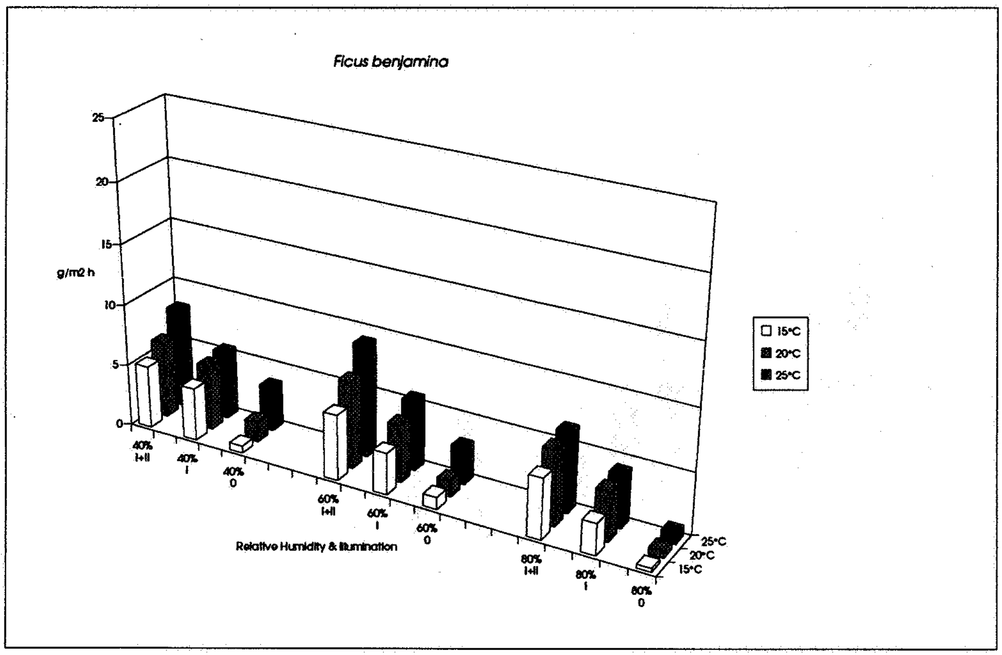
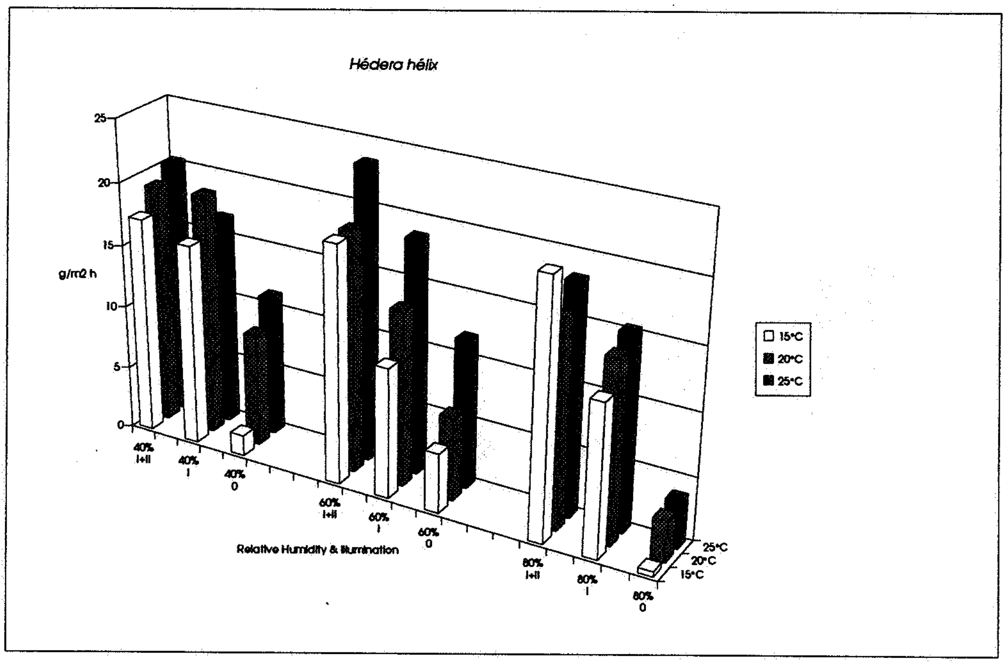


Fig. 3



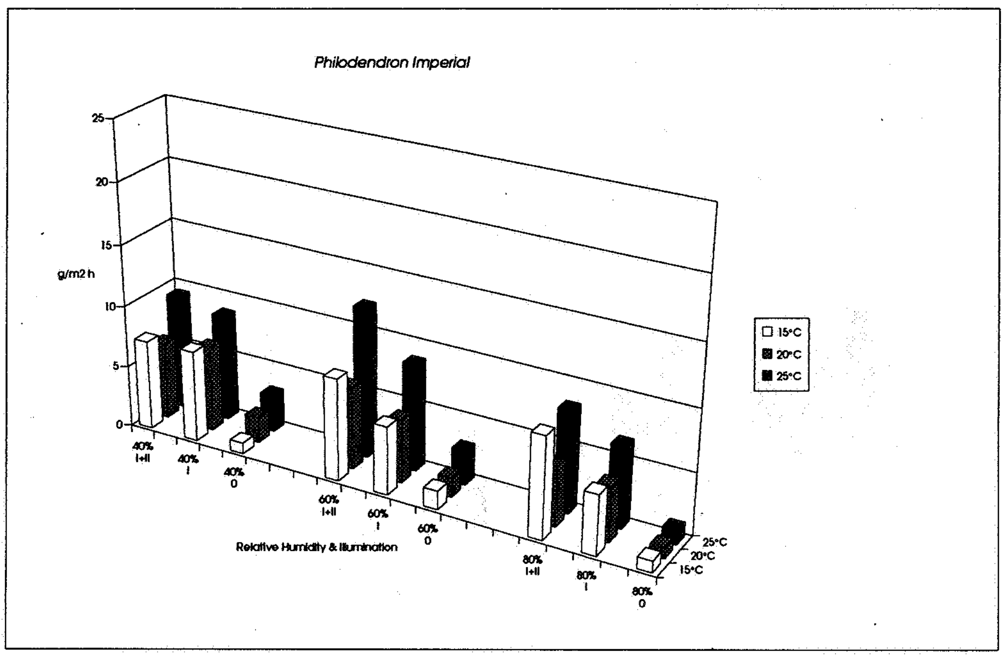
Ficus benjamina
25
20
15
10
5
0
g/m2 h
Relative Humidity & Illumination
40% I+II
40% I
40% 0
60% I+II
60% I
60% 0
80% I+II
80% I
80% 0
25°C
20°C
15°C
15°C
20°C
25°C

Fig. 4



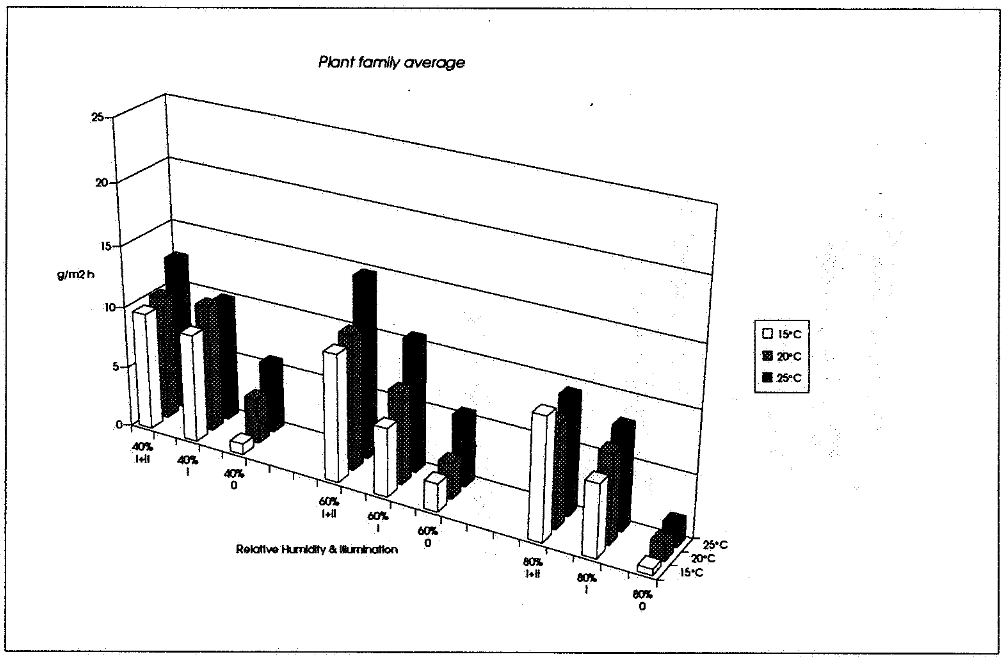
Hédera hélix
25
20
15
10
5
0
g/m2 h
Relative Humidity & Illumination
40% I+II
40% I
40% 0
60% I+II
60% I
60% 0
80% I+II
80% I
80% 0
25°C
20°C
15°C
15°C
20°C
25°C

Fig. 5



Philodendron Imperial
g/m2 h
Relative Humidity & Illumination
15°C
20°C
25°C

Fig. 6



Plant family average
g/m2 h
Relative Humidity & Illumination
15°C
20°C
25°C

Fig. 7

Plant effects on indoor air humidity and temperature

The model room

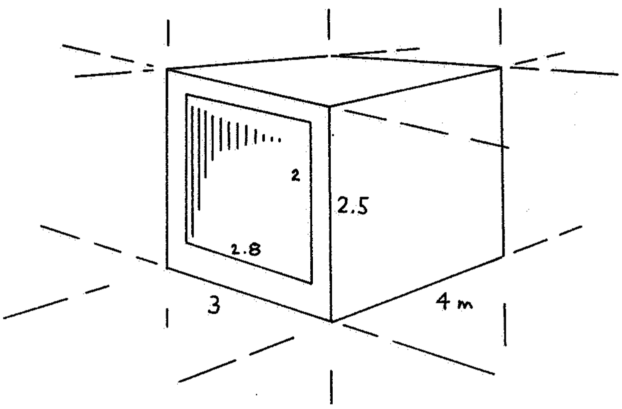


Fig. 8 Model room geometry

Outdoor climate

Our model room was exposed to climate extreme both in winter and summer. For the town of Zurich, detailed weather data are available in the form of the Design Reference Year (DRY) weather file. With this weather file, together with the model room's thermal parameters and properties, data were obtained by the use of the PC program SUNCODE V5.6. Figures 9 to 12 show predicted humidity and air temperature levels for the coldest (26 January) and the hottest (28 July) day.

Plants in the model room

According to our study in the phytotron, of the plants we used, English ivy has the highest evaporation rate. Advantages of this plant are it needs little extra space when used as a wall covering and has modest requirements of light levels and temperature. For our thermal model we fixed the following values:

Water evaporation of $4m^2$ English Ivy	day	night
in summer	72g	16g
in winter	72g	24g
(evaporation cooling power)	(50W)	

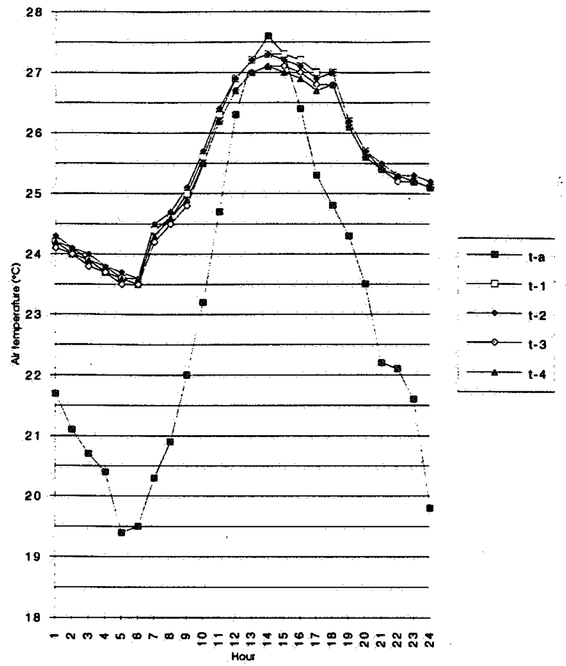


Fig. 9 Change in air temperature (T) on 28 July

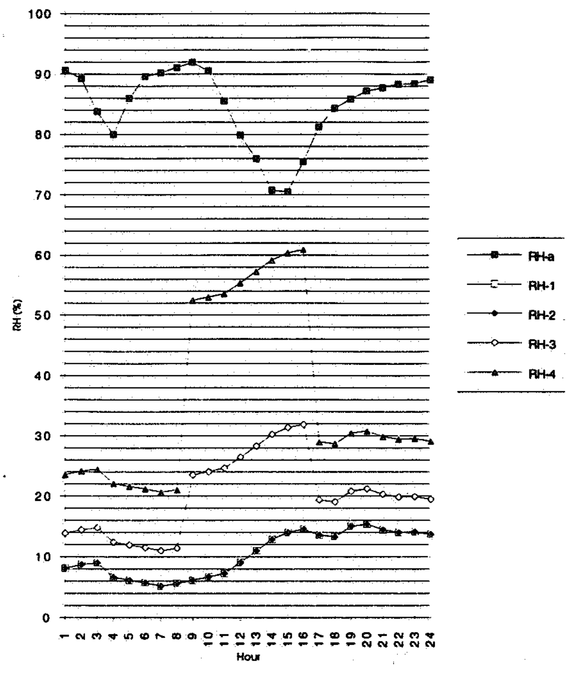


Fig. 10 Change in relative humidity (RH) on 26 January

a outdoor conditions

-1 indoor conditions without plants, air change rate n=0.8 1/h

-2 indoor conditions without plants, air change rate n=0.3 1/h

-3 indoor conditions with plants, air change rate n=0.8 1/h

-4 indoor conditions with plants, air change rate n=0.3 1/h

Summary and discussion

- *Plant water evaporation rates vary highly depending on plant species and indoor climate conditions. Evaporation rate do not depend linearly on humidity levels. Most plants will tolerate humidity levels as low as 40% but usually prefer levels between 60-80%.*

- *Humidity levels can only be increased in situations where effective plants provide a high foliage surface. Air change rates should be well below 1/h.*

- *The effect of evaporative cooling by plants is negligible.*

- *Intensive indoor planting requires an integral design method in order to provide a pleasant appearance, long plant life and comfortable room conditions both in winter and summer. A greened atrium can be a humidity source in winter but in summer, extra humidity has to be removed from the building.*

Addresses of "indoor climate gardeners"

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19 July 1994

The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Role and Tasks of Ventilation in Modern Buildings: A Case Study for Silesian Dwelling Houses

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Role and Tasks of Ventilation in Modern Buildings: Case Study of Silesian Dwelling Houses

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SYNOPSIS

The paper presents some selected results of evalution of improvement effectivenes of thermal insulation and tightness of multifamily dwelling houses located in the region of Silesia. The effect of the modernization work on heat consumption (to heat the buildings) and ventilation performance is discused. Attention is paid mainly to the sensations of the flat users connected with air flows and change. Prospects for effective implementation of thermorenovation of buildings are evaluated in the conclusions when taking into account predominating role of ventilation.
The present results of the pilot tests are a basic element of suggested changes of standards. They also represent a new approach to ventilation of dwellings.

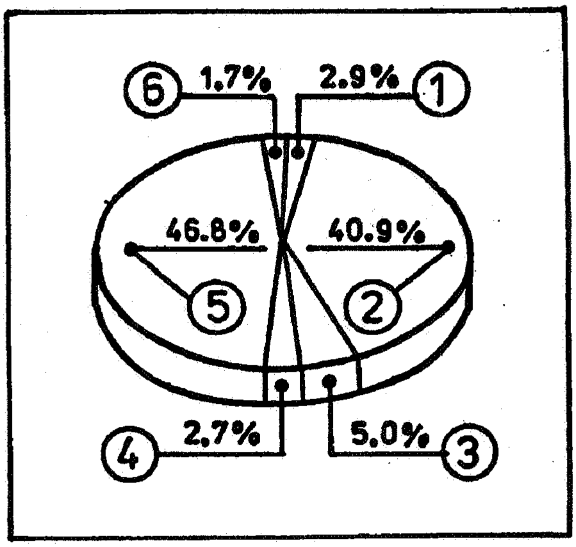
1. INTRODUCTION

Global energy use for heating, ventilation and hot water is presently about 45% of the energy balance of urban-industrial agglomerations of Silesia whereas heat losses of buildings are, on average, about 75% of the energy use [1]. The share of heat demand for ventilation purposes varies between about 25% and over 50%, depending on the size of building and thermal insulation of its walls.

Potential of energy use rationalization in buildings is significant. The evaluation carried out has proved that when thermal insulation and airtightness of buildings are improved, the energy use may be reduced by more than 45% (look Fig.1). In order to make use of that possibility, some thermorenovation work was carried out in a great number of buildings. At the same time, since 1983, new standards have been introduced successively. The standards have limited heat and air flow through the external partitions of buildings (e.g. the values of thermal conductivy coefficients for walls should not exceed 0.55 W/m^2K whereas before 1983 the upper limit was 1.16 W/m^2K).

Another characteristic example may be setting the maximum vallues of air leakage coefficients for windows (there were no limits before 1983). Presently the above coefficients should not exceed 2×10^{-4} m^3/ms at 1 Pa.

Better thermal insulation of external walls (including windows) is very much desired. However, reduction of air chanege rate by over - airtightening of external walls (above all windows) may be controversial [2].



1 - improvement of boiler house efficiency
2 - substitution of solid fuels with gas
3 - reduction of losses of district heating networks
4 - automation and regulation
5 - improvement of thermal insulation
6 - reduction of heating water losses

Fig. 1. Elements of energy saving potential

Unfortunately, the subsequent versions of the standards regarding thermal protection of buildings have not been accompanied by legat regulations in regard of ventilation needs. The results presented in the further part of the paper and, their discussion are the first in Silesia attempt to define to which extent the thermorenovation work already begun may results in undesirable sensations of the building users or even be harmful for their health.

2. BUILDINGS AND METHODS USED

The tests begun in 1984 have been carried out in multifamily buildings used for 5 to 40 years. The buidings are of different shapes sizes and the inner structure. All the buildings have natural ventilation systems (ventilation ducts without fans) and individual water heating systems (gas heaters placed in windowless bathrooms, connected to waste gas ducts). Nine of the tested buildings are identical 11 storey houses of the same construction and with the same installations. Since they are typical of multifamily buildings in Silesia; most of the data presented in the paper refer to those nine houses.
The measurements consisted in continuous registration of thermal energy supplied to the building (in their heat centres). At the same time parameters of the outdoor climate and air temperature in about 75% of the indoor spaces were recorded. Thermovision and air flow visualization were also applied. That made it possible to select rooms for tests of airtightness (pressure method) and air change rate (tracer gas method with N_2O and SF_6 used as tracer gases). The data acquired was then used for computer simulation of air flows between seperate zones where the author's own computer code SYMWENT was applied. The tests were done before and after thermorenovation of the buildings. Inquiries were carried out in the similar way.
Two questionnaire types were used, worked out by local experts. The first one included more than 30 questions regarding thermal comfort, performance of heating and ventilation equipment, indoor air quality, etc. The other one asked the users to state the health problems related to the indoor environment conditions.

Detailed decription of the buildings, methods applied and results obtained were presented in some expert reports financed by the local administrative authorities and reports of the Silesian Technical University [3,4,5]. The paper presents only a part of the tests.

3. SOME SELECTED RESULTS AND COMPARISONS

Characteristic ranges of changes of heat loss balance parameters for the tested buildings are shown in Table 1. Apparently, significant reduction in air leakage coefficients for the windows has been acquired. The increase in airtightness of the windows (with no additional supply o-penings) resulted in 2-3 times reduction of air exchange with the outdoor environment. At the same time variability of air change rate, characteristic of natural ventilation was preserved both with time and in the indoor spaces. The variability is ilustrated by the data presented in Fig.2,3 and 4.

Table 1. Results of test of main components of the heat loss balance in the buildings

Description of parameter		Buildings built before 1950	Buildings bulit after 1980	
			Before thermo-renovation	After thermo-renovation
Thermal conductivity coefficients for walls (without windows), W/m^2K		$\frac{0.98}{1.46}$;(1.22)	$\frac{0.62}{1.70}$;(1.12)	$\frac{0.42}{1.08}$;(0.59)
Thermal conductivy coefficients for windows, W/m^2K		$\frac{2.90}{4.01}$;(3.20)	$\frac{2.48}{3.18}$;(2.80)	$\frac{2.22}{2.96}$;(2.68)
Air leakage coefficients for windows, m^3/mh dla 1 daPa		$\frac{5.41}{12.60}$;(8.61)	$\frac{2.16}{7.96}$;(4.62)	;$\frac{0.31}{0.92}$(0.48)
Air change rate, 1/h	-20°C, 0 m/s	1.76	1.6	0.49
	0°C, 0 m/s	1.48	0.91	0.37
	+10°C, 0 m/s	2.3	0.89	0.32
	+12°C, 0 m/s	0.99	0.48	0.21

Note:

(1) Thermal conductivy and air leakage coefficients - min/max (average)
(2) Air leakage coefficients include slots in the walls where the windows are mounted
(3) Air change rate is the results of numerical air flow simulation assuming real values of air leakage coefficient

In practice, the problem of air change rate variability is much more complex. The data shown in Fig 2,3 and 4 do not take into account room airing by opening the windows whereas in

practice due to window opening thermorenovation of the walls may be ineffective. This can be proved by the results shown schematicaly in Fig.5.
Apparently, the expected 35% reduction in the heat use was not achieved. The main reason for that was the necessity of permanent airing of the rooms which is also confirmed by the questionnaires.

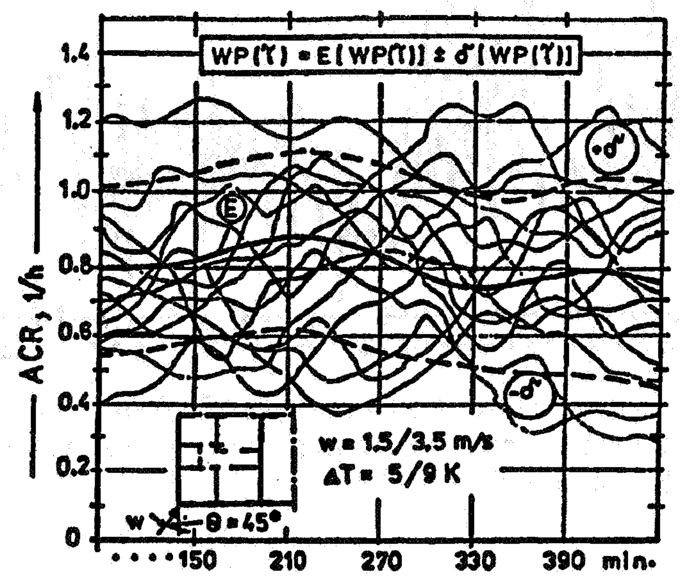


Fig.2. Variability of air change rate in a flat on the third floor of the tested building (measuremement results employing the method of NO_2 constant concentration and concentration and covering 14 four-hour period before thermorenovation)

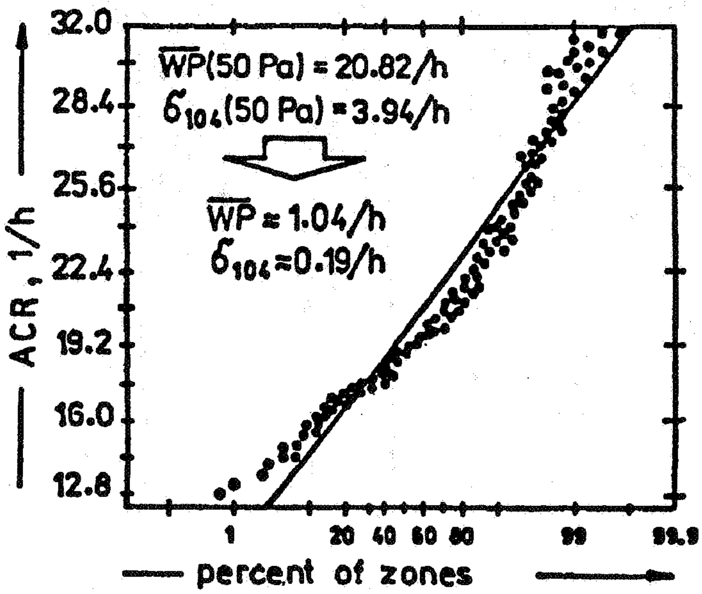


Fig.3. Normal distribution of air change rate for 104 zones located in one of the tested buildings (results of pressurization tests at Δp about 50 Pa - before termorenovation)

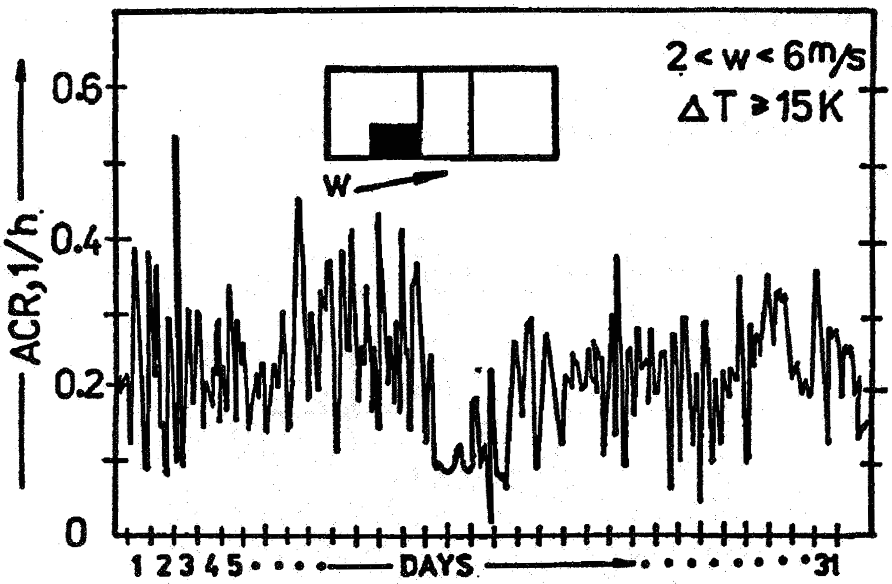


Fig.4. Variability of air change rate in a room on the fifth floor of one of the tested buildings after its thermorenovation (measurement results employing the method of constant SF_6 emission)

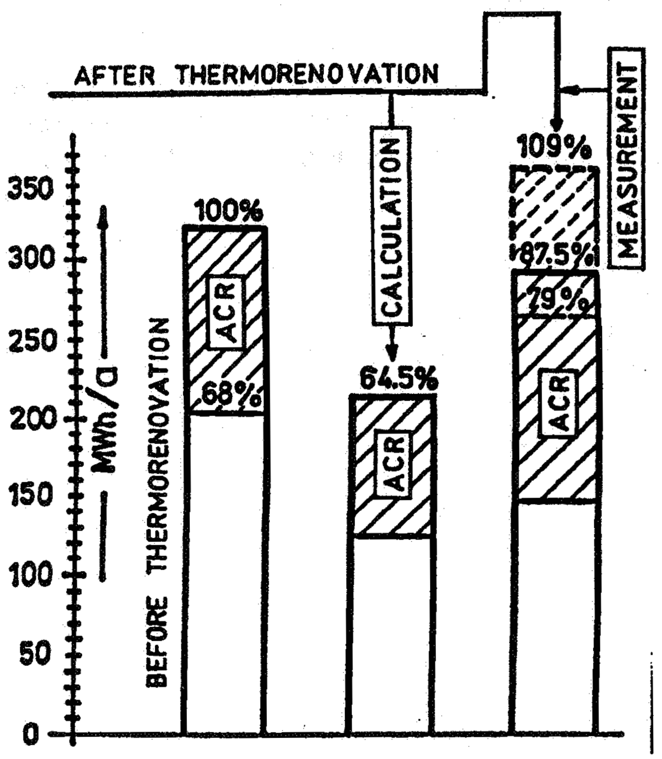


Fig5. Comparision of heat amounts used to heat 9 identical multifamily buildings .(calculations were made by means of TRYNSYS and SYMWENT)

Table 2. Number of total health symptoms and their occurrences reported by inhabitants (building with sealed windows)

SYMPTOM	DWELLING HOUSES			
	Built before 1950 (481 respondents)		Bulit after 1980* (658 respondents)	
Eye irritation	(63)	13,1%	(228)	34,6%
Dry/sore infection	(8)	1,7%	(50)	7,6%
Irriative cough	(37)	7,7%	(161)	24,5%
Excessive phlegm	(59)	12,3%	(108)	16,4%
Sinus infection	(2)	0,4%	(16)	2,4%
Bronchial pneumonia	(17)	3,5%	(21)	3,2%
Asthmatic attacks	(41)	8,5%	(54)	8,2%
Headaches	(216)	44,9%	(410)	62,3%
Dizziness	(193)	40,1%	(272)	41,3%
Unusual fatigue	(182)	37,8%	(200)	30,4%
Difficulty in sleeping	(31)	6,4%	(112)	17,0%
Nasal irritation	(50)	10,4%	(108)	16,4%
Nosebleed	(3)	0,6%	(21)	3,2%
Nausea	(40)	8,3%	(103)	15,7%
Vomiting	(18)	3,7%	(68)	10,3%
Abdominal irritation	(40)	0,8%	(18)	2,7%
Whole body ache	(23)	4,8%	(71)	10,8%
Fever	(11)	2,3%	(38)	5,8%
Stuffy/"bad" air	(33)	6,9%	(59)	9,0%
Average number of symptoms per person	2.1.		3.2.	

More than 70% of the whole number of the respondents mentioned the unpleasant sensation of overheating of rooms and lack of ventilation (air change). It should be pointed out that the percentage was less that 25% in the same buildings but before thermorenovation.
An increase in the number of health symptoms reported by the inhabitants of the buildings of reduced heat losses is also a characteristic change. Table 2 shows a comparisons of the health symptoms and includes data regarding more than 45 multifamily buildings.
The comparision shows that the total number of symptoms per person, reported by the questioned inhabitants is higher in modern (or modernized) buildings than in old dwelling houses. The highest frequencies are reported for headaches, dizziness, eye irriatation and even vomiting. The last symptom refers mainly to the users of buildings with windowless bathrooms with gas heaters of water. When air leakage coefficients for windows are low, phenomena of

backward operation of ventilation and waste gas ducts, characteristic of natural ventilation, increases. It has been confirmed by the results of air flow simulation employing the allthor's own simulation codes SYMWENT and SYMPOLL [5,6]. In result, the risk for the occupants' health increases significantly.

4. DISCUSSION

Most people spend major part of their lives in closed spaces. It is therefore of special importance to maintain air quality which does not impair comfort and health. Progressive increase in the cost of heating within the last years has become a reason for seeking ways to reduce heat use. On of the simplest ways may be increase of thermal insulation of buildings by thermorenovation of their walls.

When different building types are considered, dwelling houses and especially multifamily houses, present particular problems. Those problems are very complex and require simulataneous solution of seyeral components.

The test show that in spite of significant improvement of thermal insulation and airtightness of the walls, the energy use to heat the buildings is only slightly lower or even is not reduced at all. At the same time both the heat losses and the indoor environment conditions very much depend on air change rate and ventilation processes. A characteristic feature of basic heat losses of modern (or modernized) dwelling houses, presented in Fig.6, is that it is dominated by

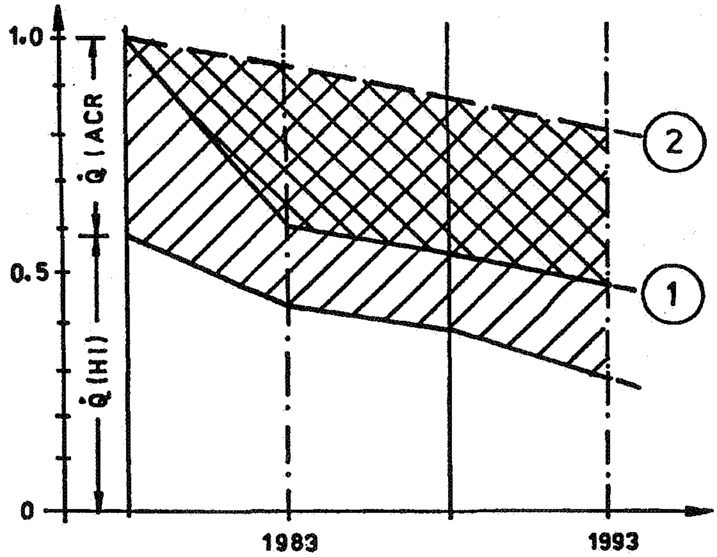


Fig.6. Changes of ventilation heat demands in the balance of basic heat losses (1983, 1993 - years when new standards were introduced)
Notations: (HI) - heat insulation, (ACR) - air change rate; 1 - according to the calculation for standard values of air leakage coefficients; 2 - as above

ventilation heat demand (see line 2). Ventilation plays particular rote in dilution and removal of pollutants within occupied spaces. Minimum requirements can be set to meet the metabolic needs of occupants. In real situation, added to such requirement are those need to dilute other sources of pollutants such as moisture, tobacco smoke, cooking, building materials, heating

appliances, furnishings, fittings, etc. At the same time, increased awareness of the potential health risks associated with outdoor (most important problems in Silesia) and indoor air pollutants should stimulate studies of ventilation systems and ventilation functioning in Poland.
The test show that the use of natural ventilation not only in small detached houses but also in multifamily (often high) buildings is inappropriate as a basis for indoor air quality and health control.

5. CONCLUSION

The attention paid to ventilation and indoor air quality in Silesia is rather new. For the last five years, research concerning building airtightness from energy point of view has been done. Basing on the results of measurements and questionnaires it may be concluded:

- Air change rate in flats and rooms located in modern or modernized dwelling houses are very small; this is a result of excessive airtightness of external walls, especially windows.
- The most significant health symptoms are headaches, dizziness, unusual fatique and eye irritation; the number of these symptoms per person appears to increase with decreasing heat consumption of the building.

In the next stage of the study, the research will include on-site measurements of air polltants in new dwelling houses. Presently research has been started on natural and mechanical ventilation efficiency both in situ and at laboratory stands. It is also expected that the research on ventilating air quality and elimination of the risks occurring in existing and actually buit buildings will be continued.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

The Role of Ventilation in Controlling the Dispersion of Radon Gas from a Cellar in a Domestic House

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ABSTRACT

In certain parts of the United Kingdom where radon gas seeps from the ground into the basement of domestic housing, normal methods of removing this gas by using under floor extract ventilation is not appropriate. In this situation the radon gas enters the basement through the side walls of the cellar and hence into the house.

Using mechanical ventilation to either pressurise or de-pressurise the cellar may be an appropriate solution to this problem, however before installing such a system in a house a ventilation strategy must be established.

This paper sets out a ventilation strategy for minimising the ingress of radon into a domestic house, which has been established by simulating the air movements within a domestic house using Breeze Version 6 for a range of environmental conditions.

The results of this analysis show that in winter when the emission of radon gas is most strong de-pressurisation of the basement can improve the ingress of the gas to the rest of the house.

1. BACKGROUND

The Building Research Establishment are currently investigating methods of reducing radon levels in housing in high risk areas of the UK. In some houses affected by radon there are basements which will allow the gas to enter through both the floor and walls. In such situations it would not be feasible to use under floor negative ventilation as this would create a negative pressure in the basement thus inducing further flows to enter through the walls. conversely by pressurising the basement the heat requirement of the house will be increased due to the more external (cold) air being induced into the house.

As part of the ongoing investigations to establish the optimum air flow rates to minimise the ingress of radon gas a contract was awarded to the Building Science Research Unit, School of Architectural Studies to investigate the this problem by firstly carrying out a computer investigation into the likely performance of an actual house and then to carry out full scale tests to validate the computer simulations.

This paper therefore presents the results of the initial computer investigation into the performance of the house under varying induced ventilation rates. It is recognised that in the real situation the air flows will be generated from both buoyancy forces due to the central heating system and the pressure driven wind forces. Both these have been taken into account in this analysis.

2. METHOD

In order to establish the appropriate ventilation strategy it was decided to use the computer model Breeze Version 6. This model is a later development of the initial Breeze model developed by the Building Research Establishment and includes a Runge-Kutta routine for tracing pollutant concentration histories in all cells of the simulated building.

As the full-scale monitoring will be carried out on an existing house details of the house had to be established in order to simulate the appropriate air movements. This was carried out by determining from a site visit the various flow paths connecting the various rooms in the house. It was not possible to carry out a full pressurisation test on the house and therefore the flow characteristics for the identified flow paths were established from data published in the AIVC Air Infiltration Calculation Techniques Guide (1).

Figures 1a-d show the layout of the house.

In order to calculate the air flows between the basement and the ground floor through the timber floor a it was decided to use a shaft connecting these two zones where the entry and exit apperatures reflect the leakage area.

2.1 Pressure Coefficients

The house was modelled using Breeze for a range of wind speeds and directions. The Cp values used were obtained from Dr. Perera (2).

2.2 Radon Concentration

In order to estimate the concentration of radon in each of the rooms a seeding concentration in the basement of 960 Bq/m3 was used.

2.3 Ventilation Rates

As the main point of this investigation was to establish a ventilation strategy for the house a fan system was incorporated into the basement which gave either positive or negative ventilation rates. The ventilation rates were varied from 1 to 10 air changes per hour.

Superimposed on to induced ventilation rates were pressure and buoyancy driven flows. This meant that the solutions obtained were appropriate to what would be expected in the real environment.

2.4 Leakage Data Used in the Analysis

The discharge coefficients for small and large openings was set to 0.65 and 1.0 respectively. The exponent for small openings was set to 0.66 and for large openings 0.5.

The diffuse leakage through the building envelope is specified by three parameters, Leakage Coefficient, Leakage Exponent and Leakage Test Temperature. As a full air leakage test could not be carried out the values were chosen for the most general case as given below:-

a) Leakage Coefficient 0.0757
b) Leakage exponent 0.6
c) Leakage Test Temperature 4 ^{o}C

2.4 Set Internal and external temperatures

The analysis was carried out for the following temperatures:

a) Basement 10 ^{o}C
b) Ground Floor 18 ^{o}C
c) First Floor 18 ^{o}C
d) Second Floor 16 ^{o}C

3. RESULTS

The results are presented in graphical form in respect to the reduction in radon concentration in the basement and the subsequent increase in concentration in selected other rooms in the house. It is not felt necessary to present the results for every room in the house as this would not add to the understanding of the likely air movements.

In winter when radon concentrations tend to be high it was felt necessary to estimate the performance of the house with the prevailing wind directions found at this time of year.

3.1 North Wind - Effect on Basement

3.1.1 Low Wind Speed (1 m/s)

Figure 2 shows that the efficiency of reduction in the level of radon in the basement is a function of the air extraction rate with the greatest efficiency (87.1%) occurring at an extraction rate of 6.6 air changes per hour.

3.1.2 High Wind Speed (5 m/s)

Figure 3 shows that at a high wind speed the, which results in the house being pressurised due to the relatively high leakage area there is a tendency for the basement to be pressurised and the reduction in the efficiency of radon reduction is small (12.7%) at an air change rate of 0.92 (+ve) to 84.7 % at 5.5 air changes per hour.

3.2 Effect on Levels in the House for North Wind at Average Conditions

The above analysis has indicated that the efficiency of radon removal is a function of wind speed, ventilation rate and whether +ve or -ve pressurisation to the cellar is used.

The rooms selected were directly above the cellar and the values in the hall are given as this is the main area for vertical transport due to stack ventilation.

3.2.1 Basement Pressurised

Figure 4 shows the effect on the radon reduction or increase in selected rooms in the house when the basement is pressurised with air change rates varying from 1 to 9 per hour. The wind speed used in this simulation was the mean speed of 3 m/s.

As the ventilation rate is increased the reduction in radon concentration in the basement increases while at low ventilation rates the increase in the other rooms increased until about 4 air changes is reached and then falls.

3.2.2 Basement De-Pressurised

From the earlier analysis it would appear that the better solution would be to de-pressurise the basement as the resultant increase in the concentrations in other rooms would be minimised. the following analysis was therefore carried out for different wind directions to establish the robustness of this assumption. Two simulations were carried out, the first for a low fan setting and then for a high fan setting (resulting in higher extraction rates), buoyancy and wind forces were still acting on the building.

3.2.1 Low Fan Setting

Figure 5 shows the results of this analysis and it is clearly seen that there is an increase in the concentrations in other rooms of the house. This can be explained by realising that at a low fan setting the wind pressure and buoyancy forces are stronger than the fan extraction force and therefore for most orientations although the fan is set to extract air, air is actually entering the basement. This has the effect of dispersing the radon to the other rooms in the house.

3.2.2 High Fan Setting

In this situation, see Figure 6, the fan pressure overcomes the wind and buoyancy pressures and therefore is more effective in removing radon from the basement. for most wind directions there is no increase in the radon concentration in the rest of the house. This is encouraging although there will be an energy penalty to pay for in winter when more heating will be required.

4. CONCLUSIONS

The computer simulation of the dispersal of radon gas from a cellar in a domestic house has shown that under certain conditions extract rates of approximately 6 air changes per hour are sufficient to make a significant contribution the control of the levels measured in other parts of the house.

The full scale testing of these findings will be started in the Autumn of 1994 and various ventilation strategies will be tested over a two year period.

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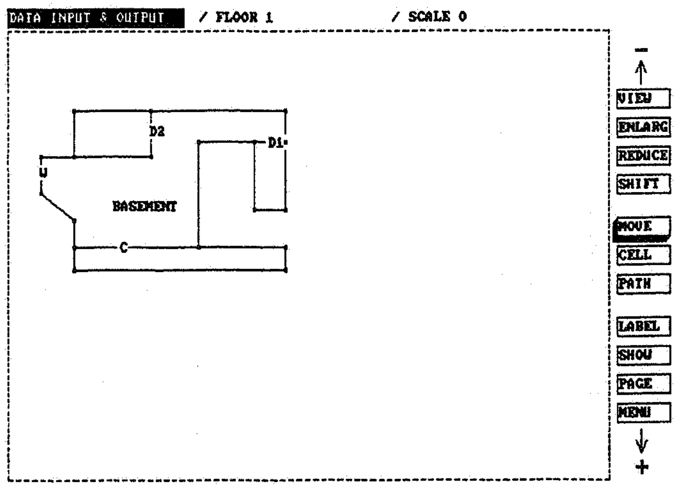
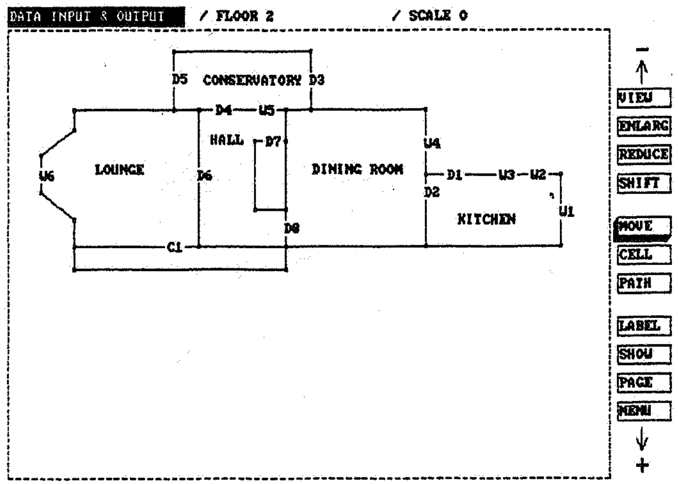
ACKNOWLEDGEMENT

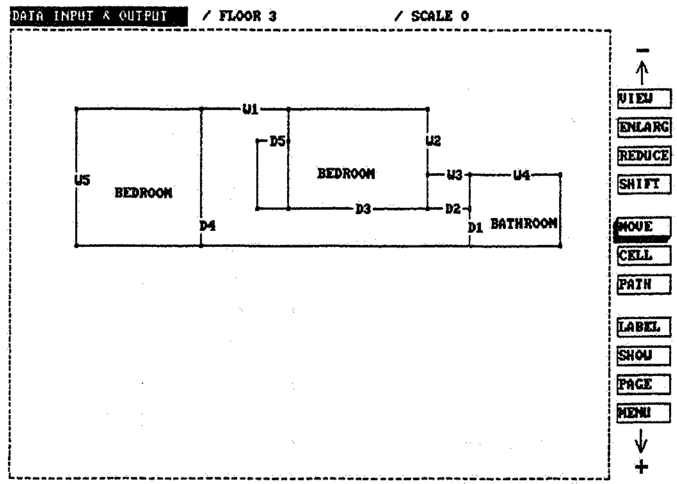
The authors would like to thank the Building Research Establishment, Department of the Environment for the financial support to carry out the computer simulations presented above.



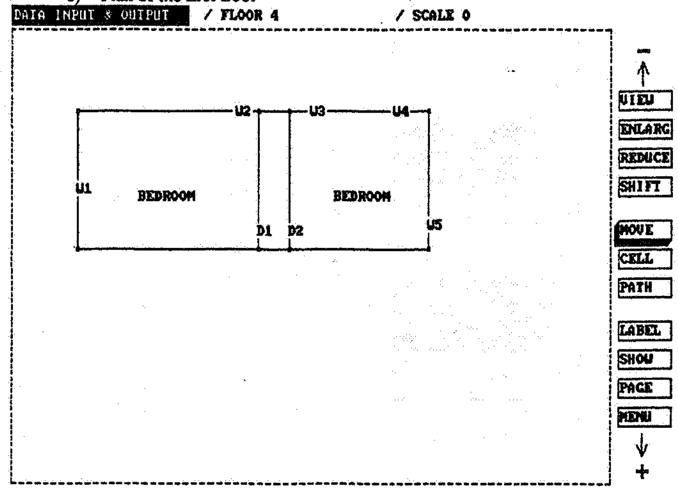
a) Basement plan



b) Ground floor plan



c) Plan of the first floor



d) Plan of The second floor (top floor)

Figure 1 House Plans as Defined in Breeze

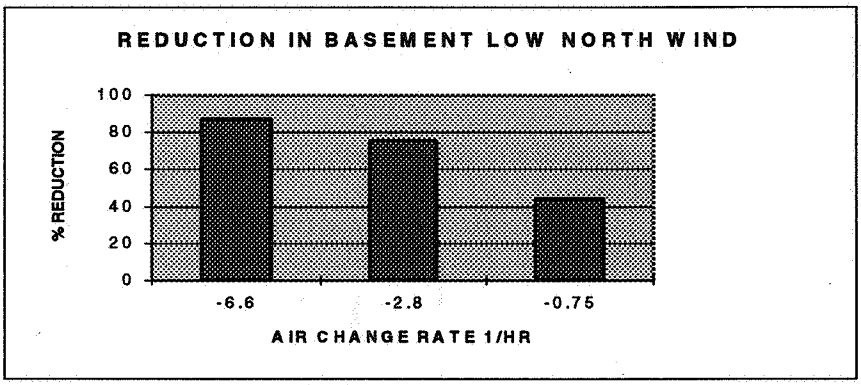


Figure 2 Reduction in Radon Level in Basement

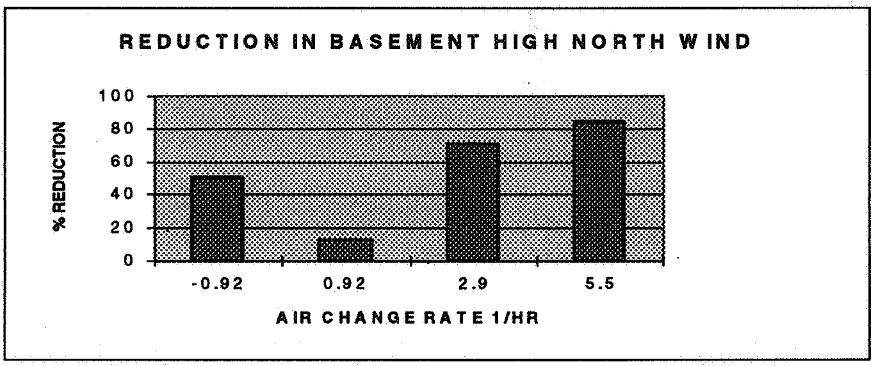


Figure 3 Reduction in Radon Level in Basement

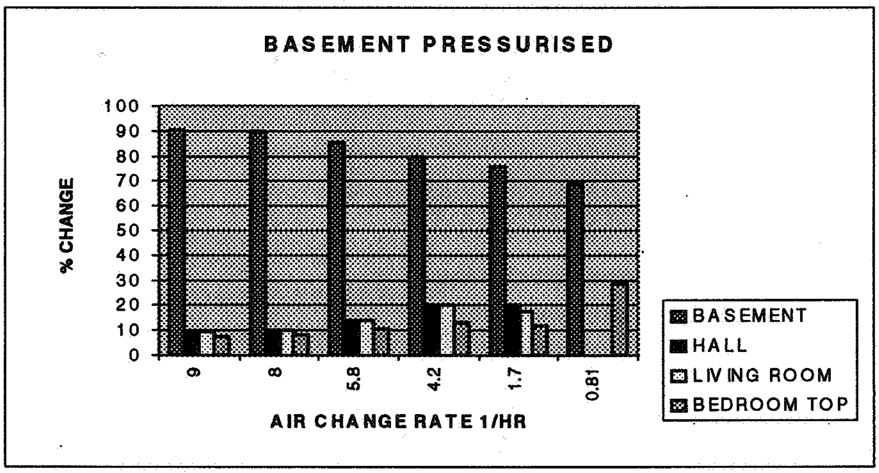


Figure 4 Reduction in Radon Level in Basement and Increase in other Rooms for Mean Wind Speed of 3 m/s

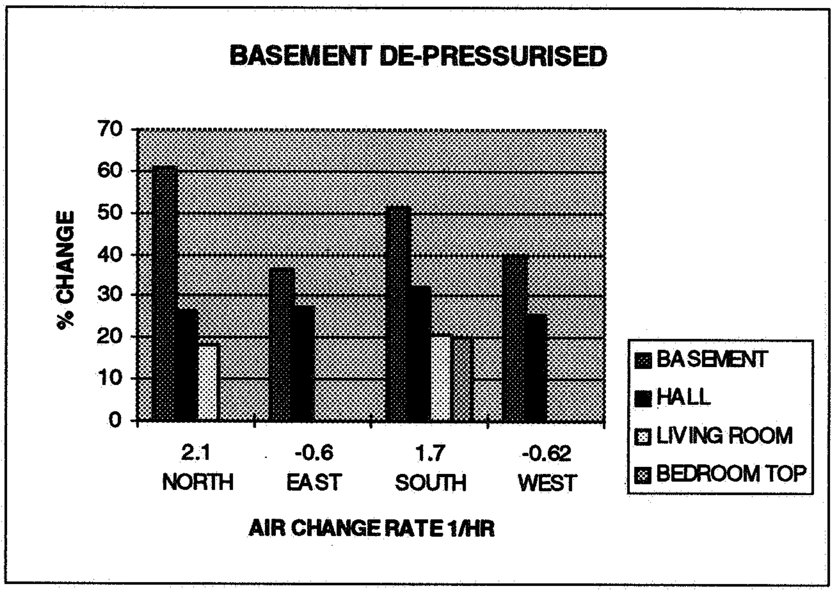
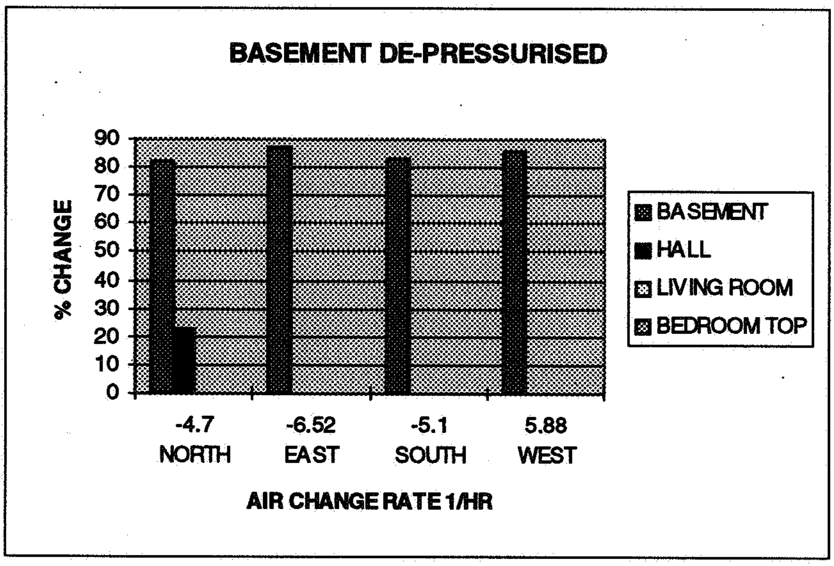


Figure 5 Reduction in Radon Level in Basement and Increase in other Rooms for Mean Wind Speed of 3 m/s and Low Fan Speed



Figur 6 Reduction in Radon Level in Basement and Increase in other Rooms for Mean Wind Speed of 3 m/s and High Fan Speed

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Detection and Mitigation of Occupational Radon Exposure in Underground Workplaces

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DETECTION AND MITIGATION OF OCCUPATIONAL RADON EXPOSURE IN UNDERGROUND WORKPLACES

ABSTRACT

The aim of this study was to unravel the occupational exposure to radon among underground workers. The possibility for radon mitigation by improving ventilation or by sealing was also investigated.

65 workrooms in 19 workplaces has been investigated in the ground floor, in basements and in underground spaces in southern Finland and in middle Finland. Radon concentration varied from 15 to 1636 Bq/m^3 during working hours resulting in annual dose of 0.09 to 10.3 mSv. The average radon concentration in all places studied was 359 Bq/m^3 (2.2 mSv) and the average in working time was 293 Bq/m^3 (1.8 mSv). Concentration of radon exceeding 400 Bq/m^3 (which is the occupational limit on annual average radon exposure in Finland) was detected in 13 (20 %) workrooms.

The mitigation methods utilized were enhanced ventilation, adjustment of exploitation time, and sealing yielding reduction of 45 - 75 % in radon exposure levels.

1. INTRODUCTION

Monitoring of occupational radon levels in workplaces has been a common practice in mines, but rarely accomplished in other working environment. A new Finnish law and regulations of radiation became valid in the beginning of 1992. These regulations /1/ includes all workplaces except mines and quarrys, which have their own previous regulations. In normal work (8 hours/day), the average annual radon levels shall not be over 400 Bq/m^3 in indoor air.These regulations are based on ICRP's recommendations. The newest regulations, which determine the occupational annual limit of radon to be 5 mSv, has been published by ICRP in 1993 /2/.

In China, Deng et al. /3/ have studied radon in 51 underground buildings, including hotels, restaurants, entertainment halls, shops and factories. Mean concentration of radon varied from 3 to 616 Bq/m^3 in these buildings. Increasing of depth and fissures in walls seemed to increase radon levels. In another Chinese study /4/ mean indoor air concentration of radon was 75 Bq/m^3 in underground buildings including a shop, a restaurant and a market. The annual average radon concentration was 93 Bq/m^3 in 74 subway stations in Korea /5/. The concentrations varied within a wide range from undetectable to 677 Bq/m^3. Radon levels in three adjacent rooms situated in the lower ground level of a multistorey office building were found to be from 73 to 130 Bq/m^3 in Switzerland /6/. The Finnish Centre for Radiation and Nuclear Safety (STUK) began a survey of radon levels in Finnish workplaces in 1993 /7/. So far, they have investigated about 3500 workplaces in cities, situating in areas, where radon levels have amounts exceeded 400 Bq/m^3 in dwellings. According to Annanmäki about in 30 % of studied workplaces radon levels were over 300 Bq/m^3.

Sealing, effective ventilation, pressurization and subslab depressurization have been found out to be effective mitigation methods for radon. In United States /8,9,10/ some studies have been conducted on radon problems and mitigation methods in schools. Those indicated that HVAC pressurization has the capability to provide both radon reduction and improved indoor air quality in new and existing buildings. According to Cohilis et al. /11/, effective ventilation with pressurization and sealing the constructions significantly (14 - 98 %) decreased radon levels in Belgian schools. Similar findings have been found by our group /12,13/.

2. MATERIALS AND METHODS

2.1 The workplaces

Radon concentrations vary largely in different parts of Finland. High radon concentrations have been detected in southern Finland, especially in areas with weathered granite and eskers. The underground spaces studied situated in southern Finland (15 places) and in middle Finland (4 places). Workplaces studied included different kind of offices and servicing rooms in schools, office buildings, telecommunication centres and military forces. Volumes of workplaces varied largely, from small office rooms of 20 m^3 to large research halls of 17 200 m^3.

2.2 Measurement techniques

Volumes, exploitation times of ventilation and ground work of workplaces, working hours and amount of employees were inquired by questionnaire. Radon levels were analyzed at breathing zone continuously by using the Lucas cell method /14/ with a Pylon AB-5 assembly which includes detector, photomultiplier and data collection system based on a microprocessor. The output data of the Pylon detector were processed with SP-55 software run on a PC. The interval of continuous measurement with the pump flow rate of 0.4 l/min was 30 minutes (averaged to one hour). The integrated long-term level of indoor radon was detected by the alpha track etch films /15/, which gives the average radon level during one month, including also nights and weekends, when the HVAC-system is not as effective as in the daytime. Alfa track etch films were analyzed by the Finnish Centre for Radiation and Nuclear Safety. The pressure difference across the wall was monitored by an electronic manometer together with a datataker averaging three minute intervals to one hour and the data was run on and analyzed by a PC. During daytime working hours air-exchange rates were measured by tracer gas technique and dilution method using Freon as a tracer gas and an infrared spectophotometer, Miran 1-A, as an analyzer. Doses were calculated by using working time of 2000 hours and equilibrium factor of 0.4 /2/.

3. RESULTS

3.1 Radon consentrations

Radon levels varied from 12 to 5000 Bq/m^3 and radon levels in working hours varied from 15 to 1636 Bq/m^3. Annual average doses were respectively from 0.09 to 31.5 mSv and from 0.09 to 10.3 mSv. Radon levels exceeded 400 Bq/m^3 daily in six workplaces in 16 working rooms of places (21 %) and during working hours in 13 working rooms of places (20 %) (table 1.).

	Measured time		In working hours	
Bq/m^3	frekv.	%	frekv.	%
0 - 199	41	63.1	39	60.0
200 - 399	8	12.3	7	10.8
400 - 799	10	15.2	7	10.8
800 -	6	6.1	6	9.2
Missing	-	-	6	9.2
Total	65	100.0	65	100.0

Table 1. Radon levels (Bq/m^3) with gategories.

	N= 65	Ave	Med	Min	Max	Stddev	Work.
Groundwork:							
Crawling space	1	1082	1082	1082	1082	-	1636
Ground floor	22	134	104	25	508	122	135
Basement	17	663	210	53	5000	1177	427
Cave	25	293	140	12	1647	376	316
Depth:							
0-3 m	32	385	117	25	5000	896	232
4-20 m	12	547	442	12	1647	503	736
21- m	15	140	128	30	412	103	120
Ventilation:							
Natural ventilation	2	2625	2625	303	5000	3321	303
M. exhaust	10	292	125	53	1802	534	294
M.exh. & suppl.	51	291	131	12	1647	354	300
Else	2	126	126	84	168	59	123
Whole data	65	359	133	12	5000	693	293

Table 2. The mean, median, minimum, maximum and standard deviation of measured radon levels (Bq/m^3) in 65 workrooms. 5 results are measured by alpha track etch film and the rest are measured by Pylon AB-5. (M.exhaust = mechanical exhaust, M.exh.&suppl.= mechanical exhaust and supply, work. = average in working hours).

3.1 Mitigation methods

There was six workplaces including 13 workrooms, where concentration of radon in working hours was over 400 Bq/m^3. Places were two telecommunication centres (including 8 rooms), a conservation hall of museum (2 rooms), an office of airport (1 room), an office of school (1 room) and an office of archives (1 room). The places had mechanical ventilation, except the school, which had mere mechanical exhaust. The other telecommunication centre located in depth of 17 meters, the other located in a basement, the conservation hall of museum located in cave, the office of airport and office of school located in groundfloor and the office of archives located in depth of 17 meters. The telecommunication centre in a basement (figure 1), the airport office (figure 2) and the school (figure 3) were mitigated during this study.

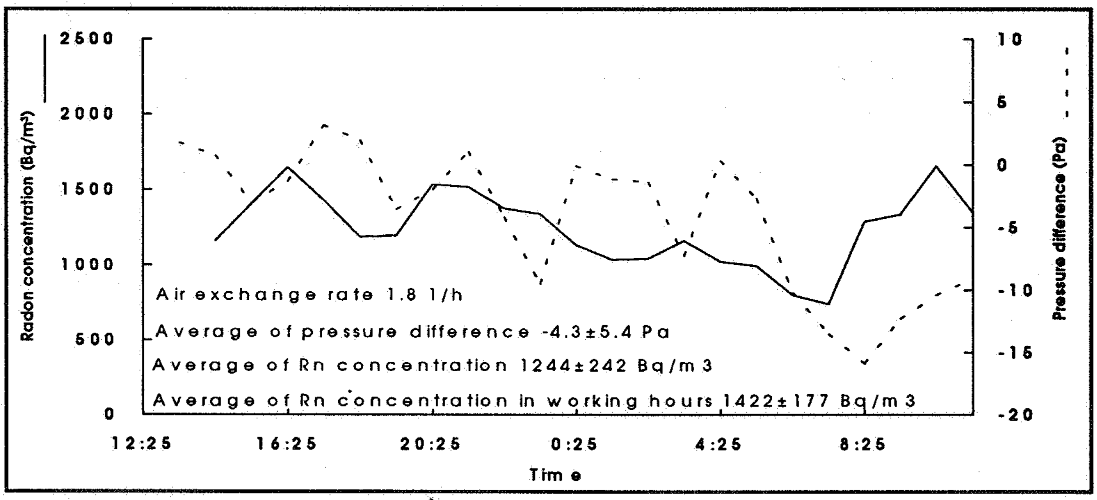

Figure 1a) Radon concentration (—) and pressure difference (- - -) in telecommunication centre, in a cross-linking room before mitigation

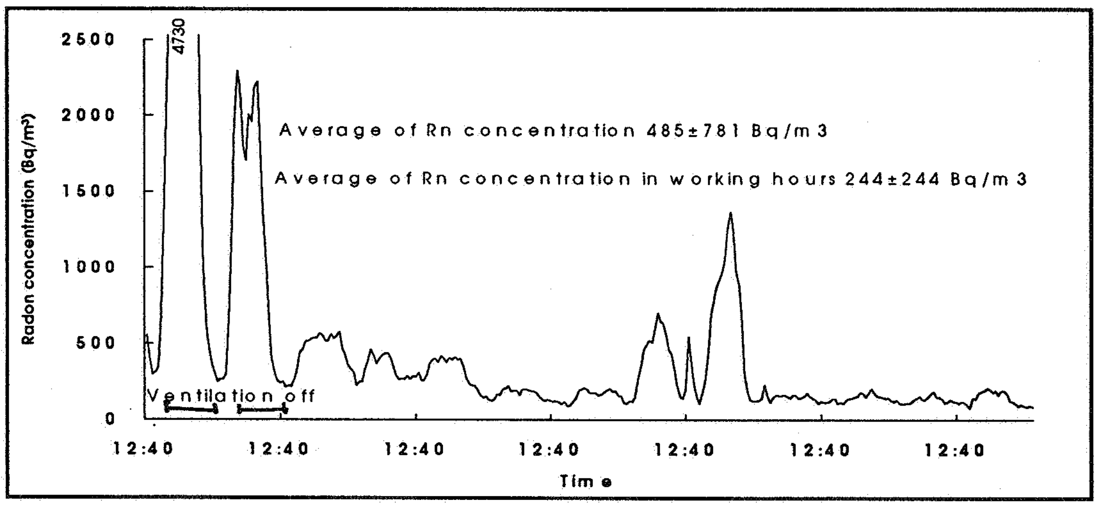

Figure 1b) Radon concentration in the cross-linking room after mitigation. The floor of the next room, where radon concentration was 5000 Bq/m^3 (measured by alfa track etch film), was sealed by a plastic carpet, and an additional exhaust air fan was installed in the room.

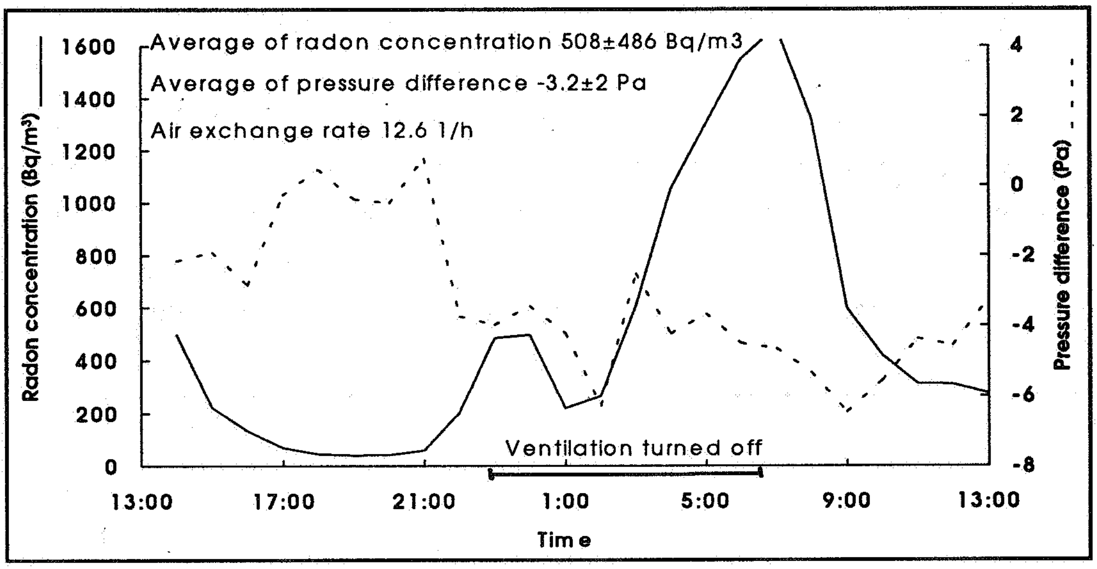

Figure 2a) Radon concentration (—) and pressure difference (- - -) in the office of airport before mitigation.

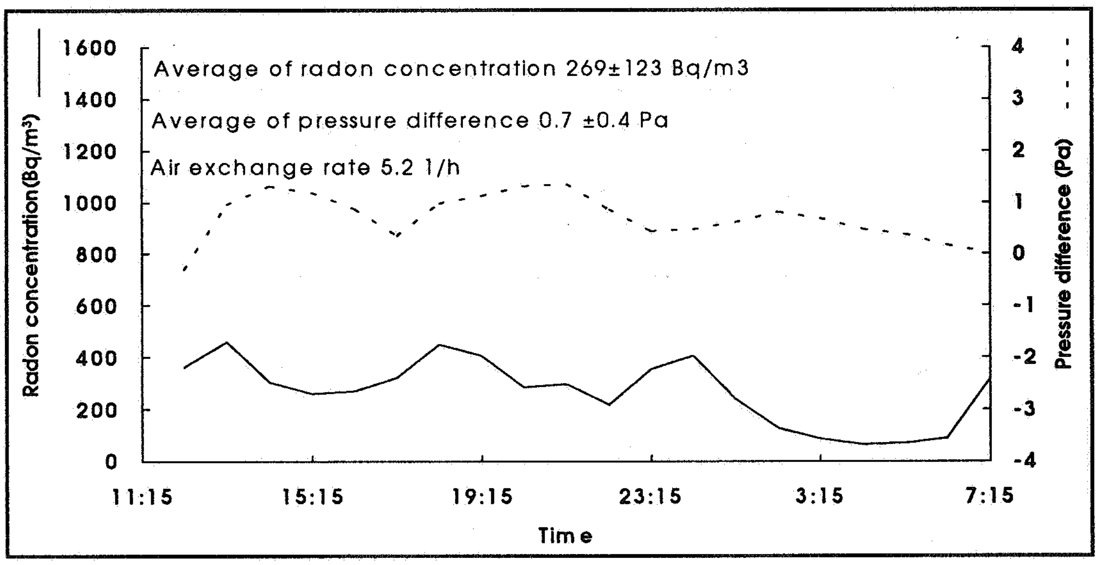

Figure 2b) Radon concentration (—) and pressure difference (- - -) in the office of airport after mitigation. Working hours in this room was 24 hours. The exploitation time of ventilation was adjusted to operate also for 24 hours.

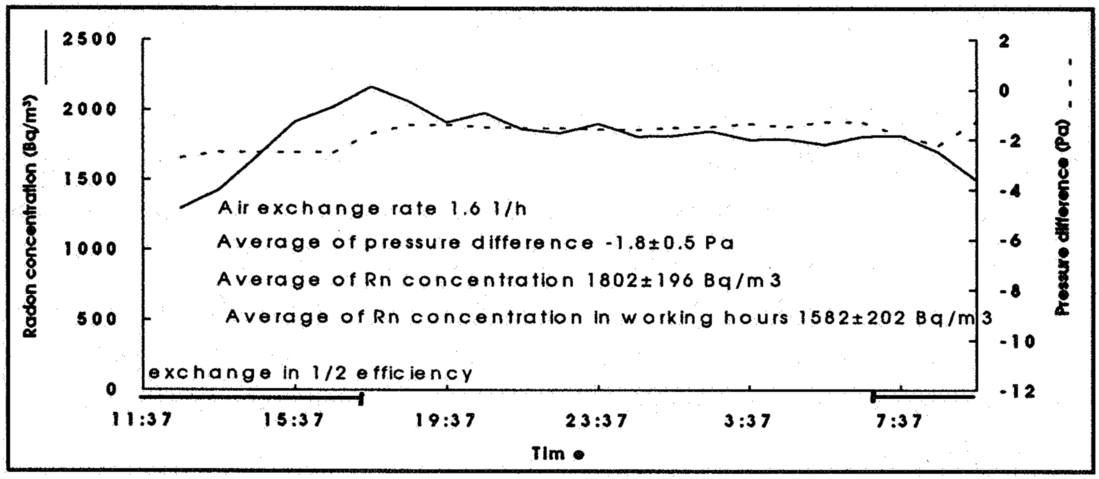

Figure 3a) Radon concentration (—) and pressure difference (- - -) in the office of school before mitigation

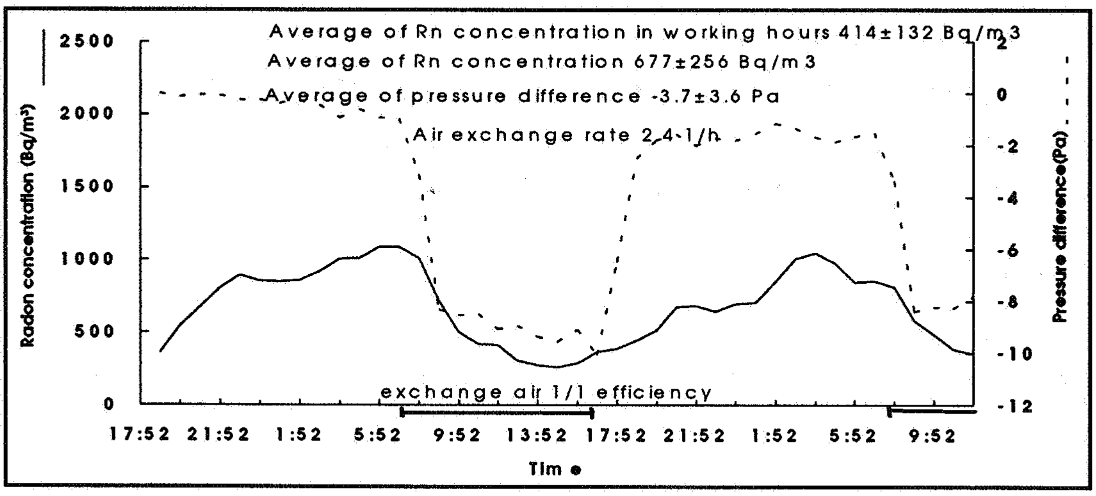

Figure 3b) Radon concentration (—) and pressure difference (- - -) in the office of school after mitigation. The mechanical exhaust ventilation was adjusted to operate more effectively, and the crawling space was naturally ventilated.

4. CONCLUSIONS

There was not any markable statistical associations between depth or ventilation system and radon concentration. The correlation between radon concentration and ground work of basement was positive but not high.

To effect the existing ventilation in underground workplaces was found to be a cheap and an easy and effective technique to mitigate radon in indoor air. Radon gas drift to indoor air is suggested to decrease, when amounts of exhaust and supply air are in balance, and constructions are sealed. The source sites have to be ventilated separately and the constructions between spaces have to be sealed. The exploitation time of ventilation should be enough long to decrease radon concentration to the action level in the beginning of working hours after weekends and nights, when ventilation is usually turned off. Ventilation of a crawling space caused a substitutional air coming from outdoor air and diluted radon gas under the school office. An optimal combination of these mitigation methods reduced the radon concentrations 45 - 75 %.

ICRP's recommendations for the occupational annual radon exposure is 20 mSv per year averaged over a period of 5 year with the effective dose not exceeding 50 mSv in any single year /2/, were not exceeded in the workplaces studied after mitigations.

ACKNOWLEDGEMENTS

The research is supported by the Foundation of Occupational Health of Finnish Ministry of Finance. The authors would like to thank Eng Matti Konttinen and Jussi Niilonen for their skilled technical assistance during measurements.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

The Mechanical Ventilation of Suspended Timber Floors for Radon Remediation - A Simple Analysis

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THE MECHANICAL VENTILATION OF SUSPENDED TIMBER FLOORS FOR RADON REMEDIATION - A SIMPLE ANALYSIS

By M Woolliscroft

SUMMARY

Mechanical ventilation of the underfloor space is one of the most effective ways of reducing radon levels in buildings with suspended timber floors. There is a question however whether this ventilation should be supply or extract, sometimes extract is more effective, sometimes supply is more effective. This report presents a simple analysis of the problem and suggests the hypothesis that the relative effectiveness of supply or extract ventilation to the underfloor space depends on the relative airtightness of the floor and the soil or oversite surface. The analysis suggests that if the floor is relatively tight then supply ventilation may be more effective whereas if the floor is relatively leaky or there is oversite concrete then extract may be better. It is suggested that in either case it is better to keep the underfloor pressure low and that when mechanical ventilation is provided to the underfloor space it may be necessary to increase the number of airbricks.

This paper is to be submitted to the AIVC Conference September 1994

July 1994

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THE MECHANICAL VENTILATION OF SUSPENDED TIMBER FLOORS FOR RADON REMEDIATION

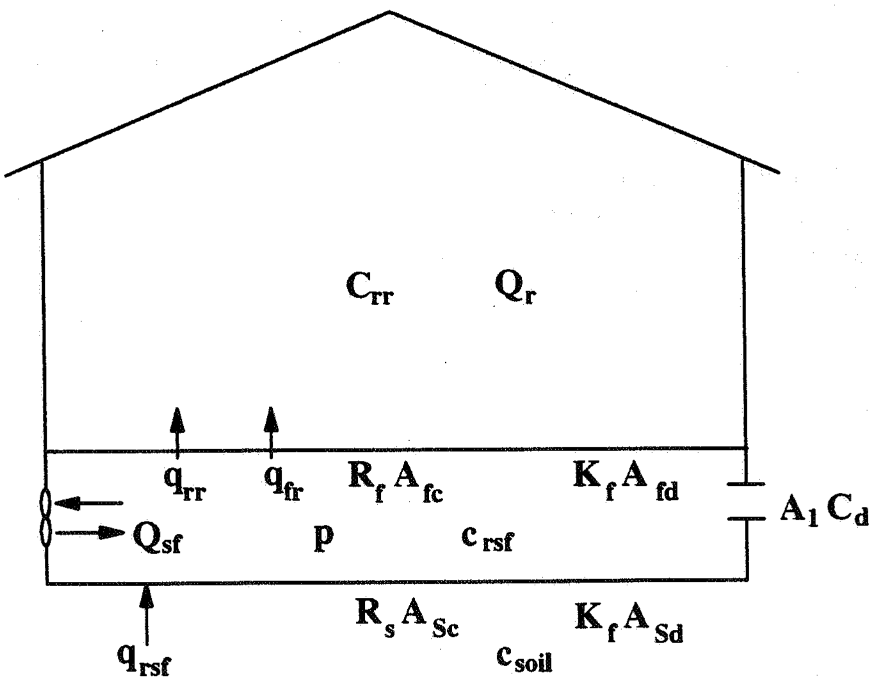
A note on the merits of Extract v Supply Ventilation.
by M Woolliscroft

INTRODUCTION

Mechanical ventilation of the underfloor space is generally one of the most effective ways of reducing radon levels in buildings with suspended timber floors. Where there is no concrete oversite it is often the only way. There is a question however whether this ventilation should be extract or supply. Whilst extract from the underfloor space will reverse the pressure gradient across the floor thus preventing radon laden air from entering the room it will also draw more radon in from the soil thus increasing the radon concentration in the underfloor space thus any radon which does get into the room will be of higher concentration. On the other hand supply ventilation by reversing the pressure gradient across the soil will prevent radon from getting into the underfloor space but any radon which does get into the underfloor space will be blown into the room. The purpose of this note is to try to determine analytically a broad outline for the conditions under which either supply ventilation on the one hand or extract ventilation on the other is the optimum solution.

THE ANALYTICAL MODEL

The simplified analytical model is shown diagrammatically in figure 1 below. Flow may be extract or supply.



SYMBOLS

p	is the pressure in the underfloor space with respect to outside
ρ	is the density of air
q_{rsf}	is the radon flow rate from the soil to the underfloor space
c_{soil}	is the radon concentration in the soil
K_s	is a diffusion flow constant soil to underfloor space. A diffusion flow constant $K = \frac{D}{L}$ has been defined where D is the diffusion coefficient, because although the characteristic length L is easy to define in the case of the floor (thickness of boards), it is more difficult to define in the case of the soil.
A_{sd}	is the flow area of the soil to diffusion
A_{sc}	is the flow area of the soil to pressure driven flow
c_{rsf}	is the radon concentration in the underfloor space
Q_{sf}	is the fan supply or extract rate to the underfloor space
A_1	is the opening area to the underfloor space
C_d	is the discharge coefficient of the air inlet/outlet to the underfloor space
q_{rr}	is the radon flow rate from the underfloor space into the room
K_f	is a diffusion flow constant for flow from the underfloor space to the room
A_{fd}	is the flow area of the floor to diffusion
A_{fc}	is the flow area of the floor to pressure driven flow
c_{rr}	is the radon concentration in the room
Q_r	is the ventilation flow rate in the room
R_s	is the flow resistivity of the soil
R_f	is the flow resistivity of the floor
q_{fr}	is the airflow rate from the underfloor space into the room

In the case of extract ventilation the process assumed is that radon enters the underfloor space under pressure driven flow from the soil. It is then diluted by the ventilation of the underfloor space but some radon enters the room by diffusion. Where pressure driven flow exists across a boundary diffusion flow is assumed to be much smaller and is ignored.

In the case of supply ventilation of the underfloor space radon is assumed to enter the underfloor space by diffusion from the soil. It is then diluted in the underfloor space and radon at a lower concentration is then blown into the room through cracks in the floor by pressure driven flow. Flows are taken to be steady state throughout.

EXTRACT

$$p = -\tfrac{1}{2}\,\rho\left(\frac{Q_{sf}}{A_1 C_d}\right)^2 \qquad (1)$$

$$q_{rsf} = \frac{-pA_{sc}}{R_s} \cdot c_{soil} \quad \text{pressure driven flow from soil to underfloor space} \qquad (2)$$

assuming flow is laminar i.e. Darcy

substituting equation (1) in (2)

$$q_{rsf} = \tfrac{1}{2}\rho \left(\frac{Q_{sf}}{A_1 C_d}\right)^2 \frac{A_{sc}}{R_s} \cdot c_{soil} \tag{3}$$

By definition

$$c_{rsf} = \frac{q_{rsf}}{Q_{sf}} \tag{4}$$

assuming that the ambient radon level C_o is relatively small and
that $q_{rsf} << Q_{sf}$ i.e. the flow of radon gas is much less than the underfloor ventilation rate.

Substituting equation (3) in (4)

$$c_{rsf} = \tfrac{1}{2}\rho \frac{Q_{sf}}{A_1^2 C_d^2} \cdot \frac{A_{sc}}{R_s} \cdot c_{soil} \tag{5}$$

Diffusion across the floor (It is assumed that flow of radon across the floor in the extract case is by diffusion only because the pressure gradient is negative)

$$q_{rr} = K_f A_{fd} \,(c_{rsf} - c_{rr}) \tag{6}$$

By definition

$$c_{rr} = \frac{q_{rr}}{Q_r} \tag{7}$$

again assuming that the ambient radon level C_o is relatively small

eliminating q_{rr} between equations (6) and (7)

$$Q_r \, c_{rr} = K_f A_{fd} (c_{rsf} - c_{rr}) \tag{8}$$

hence

$$c_{rr} = \frac{K_f \, A_{fd} \, c_{rsf}}{Q_r + K_f \, A_{fd}} \tag{9}$$

substituting for c_{rsf} from equation (5)

$$\underset{extract}{c_{rr}} = \frac{K_f A_{fd} \, \tfrac{1}{2}\rho \dfrac{Q_{sf}}{A_1^2 C_d^2} \cdot \dfrac{A_{sc}}{R_s} \cdot c_{soil}}{Q_r + K_f A_{fd}} \tag{10}$$

SUPPLY

Flow into underfloor space by diffusion

$$q_{rsf} = K_s A_{sd} \ (c_{soil} - c_{rsf}) \tag{11}$$

By definition

$$c_{rsf} = \frac{q_{rsf}}{Q_{sf}} \tag{12}$$

substituting (12) in (11)

$$q_{rsf} = K_s A_{sd} (c_{soil} - \frac{q_{rsf}}{Q_{sf}}) \tag{13}$$

rearranging

$$q_{rsf} \ (\frac{1}{K_s A_{sd}} + \frac{1}{Q_{sf}}) = c_{soil} \tag{14}$$

$$\therefore \ q_{rsf} = c_{soil} \ (\frac{K_s A_{sd} \ Q_{sf}}{K_s A_{sd} + Q_{sf}}) \tag{15}$$

substitute (15) in (12)

$$c_{rsf} = \frac{c_{soil} \ \frac{K_s A_{sd} \ Q_{sf}}{K_s A_{sd} + Q_{sf}}}{Q_{sf}} \tag{16}$$

$$= c_{soil} (\frac{K_s A_{sd}}{K_s A_{sd} + Q_{sf}}) \tag{17}$$

Flow into room by pressure

$$p = ½\rho \ (\frac{Q_{sf}}{A_1 C_D})^2 \tag{18}$$

The pressure is assumed uniform throughout the underfloor space.

The house will generally be at a pressure lower than atmospheric. Therefore the pressure difference between the underfloor space and the house will be greater than p say γp where $\gamma \geq 1$

assume flow is laminar

$$\text{then } q_{fr} = \gamma \frac{p . A_{fc}}{R_f} \tag{19}$$

substituting equation (18) into (19)

$$\text{then } q_{fr} = \gamma \tfrac{1}{2} \rho \left(\frac{Q_{sf}}{A_1 C_d} \right)^2 . \frac{A_{fc}}{R_f} \tag{20}$$

Flow of radon is airflow x concentration

$$\therefore q_{rr} = q_{fr} . c_{rsf} \tag{21}$$

substituting (20) in (21)

$$q_{rr} = \gamma \tfrac{1}{2} \rho \left(\frac{Q_{sf}}{A_1 C_d} \right)^2 . \frac{A_{fc}}{R_f} c_{rsf} \tag{22}$$

substituting for c_{rsf} from equation (17)

$$q_{rr} = \gamma \tfrac{1}{2} \rho \left(\frac{Q_{sf}}{A_1 C_d} \right)^2 . \frac{A_{fc}}{R_f} c_{soil} \left(\frac{K_s A_{sd}}{K_s A_{sd} + Q_{sf}} \right) \tag{23}$$

$$\text{By definition } C_{rr} = \frac{q_{rr}}{Q_r} \text{ again assuming } q_{rr} << Q_r$$

$$\therefore \underset{supply}{C_{rr}} = \gamma \tfrac{1}{2} \rho \left(\frac{Q_{sf}}{A_1 C_d} \right)^2 . \frac{A_{fc}}{R_f} \frac{C_{soil}}{Q_r} \left(\frac{K_s A_{sd}}{K_s A_{sd} + Q_{sf}} \right) \tag{24}$$

Let us now look at the ratio of the room radon concentration for supply compared with that for extract.

$$\frac{C_{rr}\ supply}{C_{rr}\ extract} = \frac{\gamma \tfrac{1}{2}\rho \left(\frac{Q_{sf}}{A_1 C_d}\right)^2 . \frac{A_{fc}}{R_f} . \frac{C_{soil}}{Qr} \left(\frac{K_s A_{sd}}{K_s A_{sd}+Q_{sf}}\right)}{K_f A_{fd}\ \tfrac{1}{2}\rho \frac{Q_{sf}}{A_1^2 C_d^2} . \frac{A_{sc}}{R_s} . \frac{1}{Q_r + K_f A_{fd}} . C_{soil}} \tag{25}$$

Now pressure driven flows are >> diffusion flow thus $Q_{sf} >> K_s A_{sd}$., $Q_r >> K_f A_{fd}$.

$$\therefore \frac{C_{rr}\ supply}{C_{rr}\ extract} = \frac{\gamma \left(\frac{Q_{sf}}{A_1 C_d}\right)^2 . \frac{A_{fc}}{R_f} . \frac{K_s A_{sd}}{Q_r . Q_{sf}}}{K_f A_{fd} \frac{Q_{sf}}{A_1^2 C_D^2} . \frac{A_{sc}}{R_s} . \frac{1}{Q_r}} \tag{26}$$

$$= \gamma \frac{K_s}{K_f} \frac{R_s}{R_f} . \frac{A_{fc}}{A_{fd}} \frac{A_{sd}}{A_{sc}} \tag{27}$$

Let us assume for the moment that $A_{fc} = A_{fd}$ and $A_{sd} = A_{sc}$. Now $K \alpha \frac{1}{L}$ and $R \alpha L$, which will be true for a given crack width

Thus

$$K \alpha \frac{1}{R} \tag{28}$$

$$Thus \quad \frac{K_s}{K_f} = \frac{R_f}{R_s} \tag{29}$$

$$thus\ \frac{C_{rr}\ supply}{C_{rr}\ extract} = \gamma \geq 1 \tag{30}$$

Thus we might expect extract ventilation of subfloor spaces to be more effective than supply. This agrees with the generally accepted wisdom, Henschel (Ref 1). However we have had practical cases where supply ventilation is more effective than extract. Diffusion flow of radon is not overwhelmingly through the same cracks as convective flow; diffusion through concrete for example can be significant, Rogers and Nielson (Ref 2). Rogers and Nielson also state that for concrete, diffusion flow through the slab is much greater than advection through the slab. The same may be true for timber. Thus it is entirely possible that $A_{fd} > A_{fc}$.

$$thus\ \frac{C_{rr}\ supply}{C_{rr}\ extract} = \gamma\ \frac{A_{fc}}{A_{fd}}\ assuming\ \frac{A_{sd}}{A_{sc}} = 1 \tag{31}$$

$$If\ \frac{A_{fd}}{A_{fc}} > \gamma\ then\ C_{rr}\ supply < C_{rr}\ extract.$$

Such a situation might arise with a fairly tight timber boarded floor with no floor covering. Such a case arose in a school with a polished floor. Supply ventilation to the underfloor space proved much more effective than extract.

$$The\ ratio\ \frac{A_{sd}}{A_{sc}}$$

is likely to be affected by the presence or otherwise of concrete oversite. If there is oversite then it is is likely that $A_{sd} > A_{sc}$ thus it is more likely that C_{rr} supply > C_{rr} extract.

PRACTICAL IMPLICATIONS

It is unlikely that in any particular case one is going to be able to measure the relative tightness or flow resistance of the floor on the one hand or the ground, soil or oversite on the other or the diffusion coefficient of the floor. This analysis does however suggest that where there is a tight bare floor perhaps a tongued and grooved floor and simply bare earth below then supply ventilation will be more effective. In perhaps the majority of cases however one might expect extract ventilation from the underfloor space to be more effective, particularly where there is concrete oversite. In most practical situations the case for supply or extract will not be clear cut and it would seem advisable to choose fans where the flow can easily be reversed.

Looking directly at the equation for extract ventilation equation (11) and supply ventilation equation (25) some hypotheses are suggested for the level of extract and supply. Looking first at extract equation (11) it would appear that the radon level is minimised by reducing Q_{sf} the underfloor ventilation rate and maximising the room ventilation rate Q_r. However if we substitute

$$p = \tfrac{1}{2}\rho\ \frac{Q_{sf}{}^2}{A_1^2 C_d^2} \tag{32}$$

in equation (11) we get:

$$\underset{extract}{C_{r_r}} = \frac{K_1 A_{fd}\frac{p}{Q_{s_f}} \cdot \frac{A_2}{R_s} \cdot C_{soil}}{Q_r + K_1 A_{fd}} \tag{33}$$

Thus it can be seen that the room radon concentration is reduced by; low pressure, high subfloor ventilation rate and high room ventilation rate. This agrees with the analysis of Hartless and Gardiner (Ref 3). Thus the right approach would seem to be to have a high subfloor ventilation rate at lower pressure which means a large area of ventilation openings to the underfloor space.

Similarly for supply ventilation to the underfloor space equation (24).
we get

$$\underset{supply}{C_{rr}} = \gamma p \frac{A_{fc}}{R_f} \frac{C_{soil}}{Q_r} \left(\frac{K_s A_{sc}}{K_s A_{sc} + Q_{ss}} \right) \tag{34}$$

Again this suggests minimising p and maximising Q_{sf} and Q_r

There is probably a limit to the reduction of the underfloor pressure however whether suction or positive. If this pressure is made too small then there will in the case of extract be areas of positive pressure driving radon into the room and in the case of supply, areas of negative pressure sucking up radon from the ground, due to wind effects. This suggests that the pressure generated under the floor should probably be about the same as that which would occur due to wind. Thus when fitting a fan to an underfloor space it may be advisable to increase the number of airbricks. The requirement in both supply and extract for a high flow rate at low pressure suggests the use of axial flow fans where practicable.

CONCLUSION

A simple analysis has been presented of mechanical ventilation under a suspended timber floor which suggests that the relative efficacy of supply and extract ventilation depends on the relative airtightness of the floor and the ground underneath and the permeability of the floor. The relatively tighter the floor the more effective supply ventilation as against extract. This is in accordance with some limited practical experience but the hypothesis needs systematic experimental verification. In both cases it is suggested that a high flow rate at low pressure will give the best results.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Using Pressure Extension Tests to Improve Radon Protection of UK Housing

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Using pressure extension tests to improve radon protection in housing.

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Synopsis

In order to demonstrate conformity with the current Building Regulations, many house builders are incorporating sumps beneath the ground floor construction of houses within the designated Radon Affected Areas. These sumps will allow for later depressurisation of the below ground floor construction and thereby prevent radon passage to the internal building environment. There are concerns regarding the costs of these measures and also the potential for these sumps to be used by vermin as nesting sites as well as their effectiveness.

This paper reports on an ongoing study into the effect of different fill materials on sub- slab depressurization. In each test suction is applied at the centre of a floor slab and the resulting pressure field and flow rate is measured. These data give a good indication of the way in which a sump would be expected to perform. The results show that there is significant variation from fill materials described as being the same.

Symbols

Q = Air flow rate	m^3/s
t = thickness of fill layer	m
P_i = Pressure at point i	Pa
r_i = distance from centre of suction hole to point i	m
k = permeability of fill	m^2
μ = dynamic viscosity	Pa.s
v = Darcy velocity	m/s
ΔP = Linear pressure difference	Pa
L = linear distance	m

Introduction

Radon is recognised by the Government as a significant public health risk. Background information on radon can be found in the Householders Guide to Radon [1]. The Building Regulations [2] require that builders achieve radon concentrations in dwellings that are as low as reasonably practicable. There is compulsory restriction of radon in all other buildings under health and safety legislation; exposure to radon is restricted by the Ionising Radiation Regulations 1985 [3]. Guidance on how to protect against radon in new buildings is given by BRE [4].

The National Radiological Protection Board (NRPB) estimates that some 100,000 homes in th UK may be above the designated Action Level of 200 Bq/m^3 and 10,000 places of work subject to the Ionising Radiation Regulations.

There are a number of ways of ensuring that radon levels in a dwelling are maintained below the Action Level. These fall into three main categories:

1) Dilute any radon below floor level (sub–floor ventilation).

2) Prevent any radon migration across the ground floor to the dwelling interior. (Barrier method or sub–slab depressurization)

3) Ensure that any radon entering the dwelling is removed almost immediately by adequate levels of whole house ventilation. (Interior ventilation method).

All these methods are practical in different circumstances and are used. Wimpey have in the past undertaken some investigations into the performance of radon barriers and the consequences of their use on builders and purchasers alike Ref [5].

This project concerns a programme of work to investigate subfloor ventilation of ground supported floors. It is being funded by the Department of The Environment through the Building Research Establishment. The programme is being undertaken by Wimpey Environmental Ltd, using, where practical, floor slabs constructed by Wimpey Homes Holdings Ltd. It was originally intended to undertake tests on dwellings in affected areas only, however, the relatively low number of dwellings currently being constructed and a general preference for the construction of block and beam floors in the affected areas has meant that a number of tests have been undertaken on floors outside of the designated affected areas, although they have been selected from areas close by.

Principals and test procedure

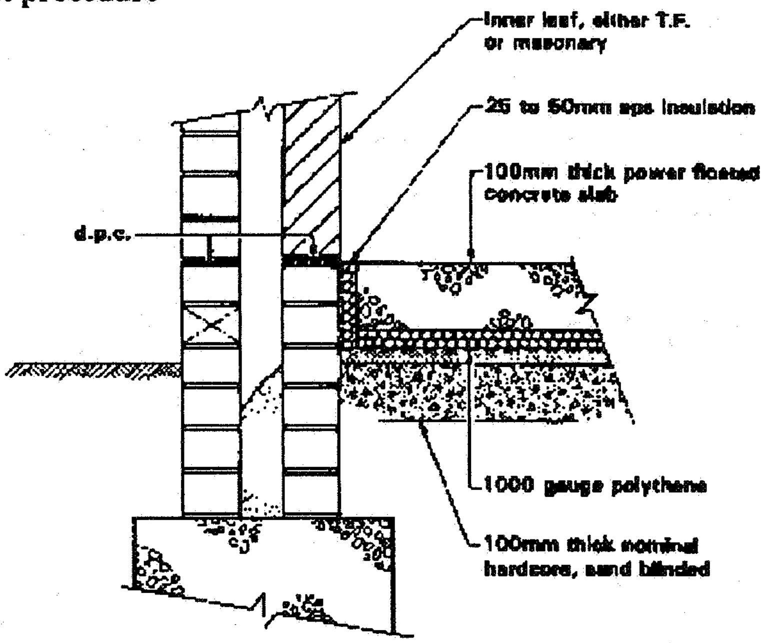

Figure 1: The key elements of a ground supported floor.

Depressurising the aggregate fill below the concrete floor, so that it is at a lower pressure than the internal air of the dwelling, will prevent the flow of radon between the fill and the dwelling. This is because the direction of flow of air is downwards through any cracks in the floor slab rather than upwards. It is this upwards flow which carries radon into a dwelling, and which the method aims to prevent.

A suspended floor has a sub–floor void (effectively a sump matched to the size of the floor), and the resistance to air flow in the void is very low. The whole of the void will effectively have the same negative pressure and the method is likely to work, either by depressurisation or by ventilation. However even in these circumstances short circuits and unventilated dead zones may be formed.

With a solid floor the underfloor void is filled with an aggregate material which restricts air movement . For a uniformly–sized fill material the resistance to the air flow will increase as the size of the aggregate is reduced because of the effect of reducing the porosity and increasing the surface area. Similarly the improved packing of aggregates with a large range of sizes results in increased resistance to air flow. For this reason many builders specify that only large, uniform aggregates should be used as floor fills in Radon Affected areas. Wimpey Homes Holdings recommend that ground supported floors include a ventilation layer which is a minimum of 150 mm thick and is made up of 75 mm single size aggregate.

The purpose of this programme is to establish, in Affected Areas, the resistance of typical floor fills to the flow of air, and how this varies with the different grades of the aggregates used for floor fills. From these results it should be possible to optimise the specification of aggregate to meet the radon remediation and constructional requirements. It is also an objective to establish whether the performance of the floor can be characterised by a simple test of this type and whether this can then be used as an indicator of the suitability of ventilation to the fill in the remediation of radon in an existing building.

It was decided to make best use of Wimpey Environmental's position within the Wimpey Group of companies and agreement in principal was obtained from Wimpey Homes Holdings Ltd to undertake tests on floor slabs under construction on their sites within radon affected areas. It is the practice for batches of floor slabs to be built and walls constructed to up to the damp proof course before overbuilding the remainder of the dwelling. It was decided that tests could be best completed at this stage with less likelihood of damage to internal finishes and the lack of internal paint work etc making general remediation easier. The disadvantage of this approach was that weather became a factor and many tests have been aborted because of high wind speeds.

Test procedure

The test procedure is a relatively simple one and involves drilling a 38 mm diameter hole in the middle of a floor slab. Checks are made to ensure that the suction hole has penetrated the damp–proof membrane. Into this hole is introduced a specially constructed suction tip which incorporates a dust filter and pressure tap. The perimeter between the suction tip and floor slab is sealed with a mastic seal or foamed sealant. To this is fixed the hose from a powerful domestic vacuum cleaner (VAX 4000) which incorporates an in– line air flow meter along the hose length.

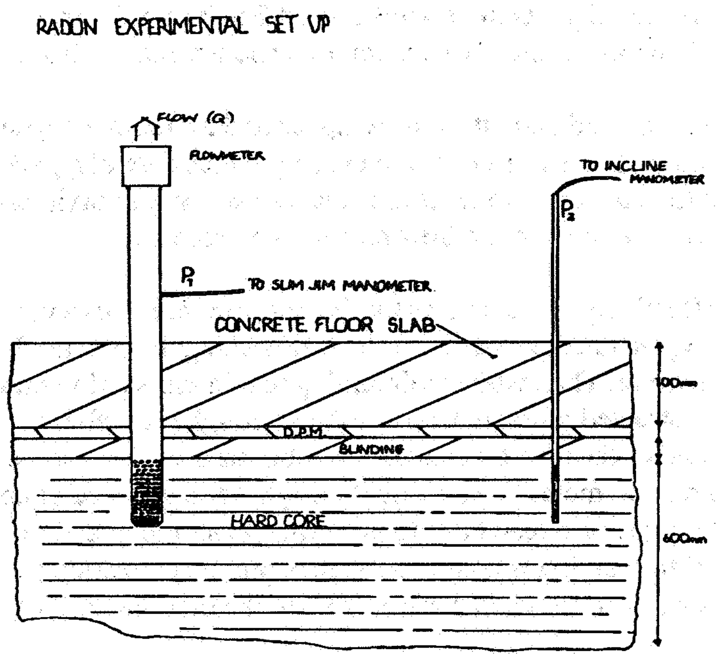

Figure 2: Schematic of experiment

Additional 10 mm diameter holes are then drilled radially from the suction hole in a diagonal line towards the floor corner, see Figure 2. Where there are internal footings and structural partition walls, for garages etc then these are avoided, as much as possible, in order to limit the influences on the measured pressures and flows. In general the smaller monitoring holes are drilled in the pattern shown with the separations doubling as the distance from the suction hole is increased; starting at roughly 150–200 mm from the centre of the main suction hole, with the next holes at 300 mm, 600 mm to a maximum separation of 1 m until the edge of the floor slab is reached

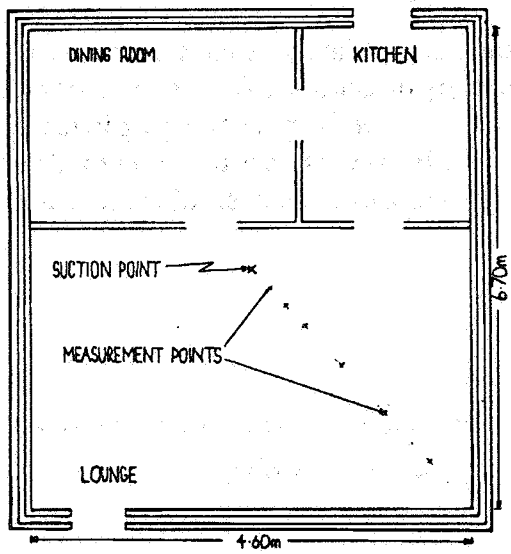

Figure 3: Diagram of suction and test holes

The largest floors encountered generally do not exceed 7 to 8 m, with many floors substantially smaller than this. Typically between five and ten sample holes are used.

The vacuum cleaner is switched on at its slowest speed and air removed from the central suction hole. At this stage all the 10 mm diameter sample holes are plugged with small rubber bungs. The suction of the vacuum cleaner is increased slowly until maximum, measurable pressure is attained; the pressure and the flow rate are then recorded.

This base resistance effectively characterises the fill performance. However it is important to assess the extent of the generated pressure field in establishing the potential for this technique as a radon remedial measure. Then while maintaining maximum suction on the vacuum cleaner, a specially constructed pressure tap is pushed through the slab into the fill. In order to prevent blocking the end of the tube by contact with the surface of any large aggregates, the tap is perforated by 3 mm diameter holes along its length for a distance of about 75 mm, as shown in figure 3. Tubing is then lead back to an incline manometer and the pressure generated as a result of the suction is measured at each prepared position. This is repeated for each of the prepared sample holes while keeping any of the other holes well sealed with the rubber bungs.

In radon Affected Areas Wimpey Homes are installing radon sumps within all ground supported floors. Consequently an adaptor was prepared which allows suction to be applied from the vacuum cleaner at the sump outlet. Measurements were then made in exactly the same way as described earlier with the sump as the source of suction. In some cases, because of the density of the slab reinforcements, it was not possible to penetrate the slab with a 38 mm diameter drill, in these cases only measurements with suction applied through the sump could be made.

Samples of the floor aggregate and details of type and supplier were obtained for each site and have been examined for particle size distribution in accordance with BS1377:Part 2:1990 [6].

The programme is still ongoing and to date tests have been undertaken on a total of 40 slabs. The programme target is currently on schedule for a total of 100 slabs to be tested in this way. The results to date are presented here and are encouraging in respect of the generated extensions to the pressure fields. However this is only part of the story with BRE and Wimpey undertaking a more rigorous mathematical analysis of the results.

Results

Total flow resistance

The simplest way to present the result of the test is to introduce a total flow resistance R (Pa.s/m^3), defined by assuming that the flow obeys:

$$Q = P / R$$

Where Q is the total flow in m^3/h, P is the total pressure difference generated by the vacuum in

Pa. The results suggest that this floor resistance varies markedly and for the nine sites examined to date we have seen variations between 6 x 10^2 Pa.s/m^3 at one site in Northampton up to 6.8 x 10^6 Pa.s/m^3 at a second site close by, see table 1. Results to date suggest that this simple test parameter may provide sufficient information in assessing the probable performance of sumps in radon remediation.

Site	Number of Tests	Average value for suction pressure divided by flow rate (Pa.s/m^3)x10^6
A	11	0.2122
B	1	0.0006
C	3	0.0054
D	7	4.7236
E	5	2.0630
F	1	1.7600
G	2	0.3139
H	2	0.7609
I	1	6.7500
Mean		1.8433
Maximum		6.7500
Minimum		0.0006

Table 1: Resistances to flow measured for 33 floor slabs

Pressure fields

A plot of measured pressure difference against the distance from the hole is shown in figure 4 for one/two of the test sites. One line can be seen to fit well to a logarithmic decay, whilst the other shows more variation from it. This reflects the variability in permeability below floor slabs.

Two measured examples of how pressure falls with distance from the suction point.

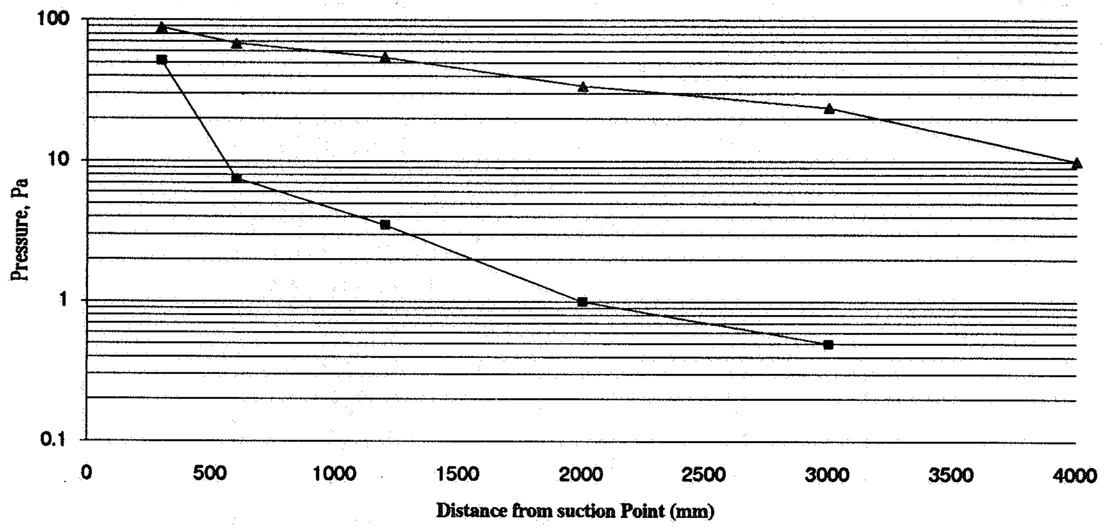

Figure 4: Graph of pressure at measurement point against distance from suction point

At the simplest level an assessment of the performance can be made by examining how far the pressure field does extend and this could be used as a criterion for the use of sumps. For example, perhaps the pressure should not fall below 5 Pa at the edge of the floor slab when a suction pressure of 9000 Pa is applied. However this does suggest that testing would be required in every case. If so, this could be simplified by specifying a minimum value for the resistance of the floor material.

The flow of fluids in soils and aggregate materials under the influence of a pressure difference P across a distance L is generally described by Darcy s Law:–

$$v = k/\mu.\Delta P/L \qquad (1)$$

where k is a constant for the aggregate called the permeability and the other variables are as defined earlier. Therefore, according to Darcy, the velocity of the air in a fill material will be linearly related to the pressure difference. This does apply in most conditions, but not at high velocities. If Darcy's Law is combined with the continuity equation ($\nabla.v = 0$) then the pressure is governed by Laplace's Equation:

$$\nabla^2 P = 0 \qquad (2)$$

Because the floor of the house and the soil below the fill are much less permeable than the fill it is reasonable to assume no flow through them. The suction hole extends down to the base of the fill material, so it is possible to assume no vertical variation in pressure. Solving (2) in cylindrical co-ordinates, and assuming radial symmetry gives:

$$P = (P_1 - P_2) . (\ln r/r_2) / \ln (r_1/r_2) + P_2 \qquad (3)$$

Where

P_1 is the pressure at radius r_1,
P_2 is the pressure at radius r_2
P is the pressure at the general point r

Differentiating (3) with respect to r, using Darcy's Law (1) and then multiplying by the area of flow $2\pi r.t$ gives the flow at any radius r

$$Q = 2\pi kt/\mu\, (P_1 - P_2)/\ln(r_1/r_2) \qquad (4)$$

Where P_1 and P_2 are the pressures measured between each sample radii. Hence if the theory above is reasonable, taking the result from any pair of measurement points, together with the total flow, (which does not change) should give the value of k.

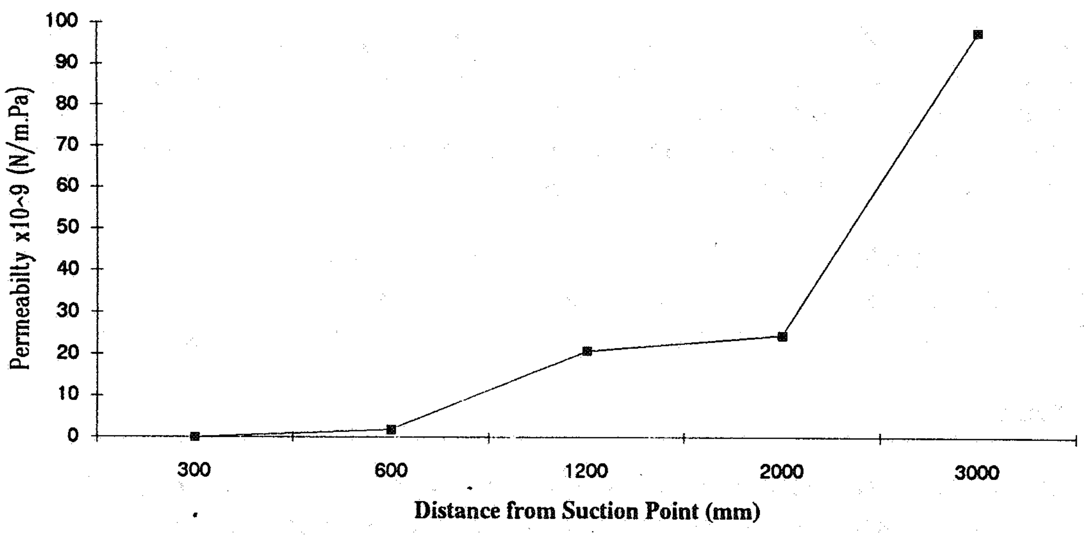

Figure 5: Variation of calculated k with distance from suction point

An example is given here of a graph of calculated permeability over viscosity for site A against distance from the suction point. It can be seen that the permeability calculated from (4) is not a constant for the fill but increases exponentially with distance from the suction point.

This behaviour is not common for every floor examined, but deserves further investigation. It results from one of a number of possible effects:

a) Darcy's Law does not apply, because the flow speeds are too high

b) The cylindrical symmetry is poor, so edge effects change the result

c) The flow is not the same at all radii because some leaves the fill at each radius

Each of these points could explain an increased effective permeability away from the suction point. If a) is significant the non-linear Darcy-Forcheimer Law for pressure loss could be used Ref [7]. The effect of b) is hard to calculate, probably requiring a numerical model. c) is also difficult to apply. The argument is that the same flow has been assumed at all radii, that is the flow which is measured at the suction point. However as some of this flow comes from the ground or through the slab the actual flow at any radius will be less than this. Hence the ratio of the central flow over the pressure difference is higher than it would be for the correct flow. To evaluate this effect will involve more computational effort than is possible now.

These are early days, but it is hoped that as the analysis progresses and more floors examined a better understanding of the ventilation performance of floor fill materials will be developed to increase confidence in radon remediation.

Conclusions

40 of a planned series of around 100 tests of the air flow through fill materials have been carried out, and a preliminary analysis of the results made. These show that there is a wide variety in the resistance of the fill materials to the air flow caused by sucking from the centre of a floor slab. The extent of the pressure extension also varies considerably, and this is the effect which matters most directly to radon remediation.

In future work the details of the make up of the fill materials will be considered along with the reasons why the permeability appears to increase with distance. In addition the use of the total resistance as a measure of the probable success of a sump needs to be considered further.

Acknowledgements

This work was supported by the Building Regulations Division of the Department of the Environment (DoE), and published with their permission. Views expressed are those of the authors, not necessarily of the DoE.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Modelling Fluctuating Air Flows Through Building Cracks

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Synopsis

The paper summarises an approach to determining the equations governing the air flow through simple cracks subject to fluctuating pressures. To this end, an experimental arrangement has been developed that enables the laboratory simulation of fluctuating driving pressure signals. A standard straight crack was subjected to this signal, which fluctuates in both magnitude and frequency.
An air control system permits a high level of fluctuating pressure control. Pressure fluctuations are imposed on a $1m^2$ section of wall with a centrally mounted crack and the overall volumetric air flow rate, pressure distribution and flow direction are monitored. In addition to the experimental testing, computational fluid dynamics (CFD) analyses were carried out for the crack type and specific flow condition. An analysis was made of initial physical test results and a comparison made between these findings and the preliminary CFD models developed.
A background to the work is given and the methods used in simulating the necessary conditions for fluctuating crack flow measurement and the preliminary results of the physical testing and CFD analyses are presented.

1. Introduction

The work presented in the paper augments research in this field which includes that of Baker et al. [1987], Sahin et al. [1988], Mokhtarzadeh-Dehghan et al. [1992] and Riberon et al. [1990]. Work by Baker et al. into non-fluctuating pressures suggested that the power law equation for crack flow,

$$\Delta P = \alpha \cdot Q^{\beta},$$

developed by Etheridge should be replaced by a quadratic form,

$\Delta P = A \cdot Q + B \cdot Q^2$. They found that the relationship

$$\frac{1}{C_d^2} = k.\frac{k}{D_h}.\frac{1}{\mathrm{Re}_h} + C$$

satisfactorily described the crack flow for a wide range of $z/Re_h.D_h$ values but that it was inadequate at higher values. As an alternative, the theoretical quadratic model of crack flow of the form:

$$\Delta P = \frac{12\mu z}{Ld^3}Q + \frac{\rho C}{2d^2L^2}Q^2$$

was suggested for non-fluctuating flow. Research by Riberon et al. into the effect of wind pressure on air movements in buildings challenged the acceptance of traditional natural ventilation design codes which use a non-fluctuating pressure/flow model. Effects were studied using a numerical model including air compressibility and a wind tunnel model, using pollutant concentration as the indicator. Theoretical solutions were compared with field data from the BOUIN test house. Numerical studies on a 2-room house revealed that transitory wind fluctuations cannot be ignored but that air compressibility can. Tests on CO_2 concentration showed a 35% difference between analyses with constant wind pressure and those with temporal wind fluctuations. However, numerical studies on a 1-room house with a single opening showed that compressibility is a factor. Similar studies on a 7-floor building indicated that taking into account wind fluctuations considerably modifies the instantaneous flow rates. The work concludes by stating the necessity of considering temporal wind fluctuations.

Work by Rao and Haghighat into wind induced fluctuating airflow in buildings similarly challenged acceptance of steady-state, non-fluctuating airflow models. They proposed a new model; employing spectral analysis and statistical linearisation to model the pulsating flow. This splits fluctuating flow into 2 categories - *eddy flow*, which represents additional air exchange through openings due to penetration of eddies, and *pulsating flow,* which is the result of bulk flow (pressure induced) across the opening. Their results showed that coherence between wind pressure and flow is low at high fluctuation frequencies, but that this is not critical as most power densities of the wind pressure occur at low frequencies. Work presented here addresses these low frequency fluctuations. In general airflow through openings can be summarised as follows, with the areas addressed by this work given in italics:

Table 1 Summary of Airflow Through Openings

Causes	*1. Wind Induced Pressure*			2.Thermal buoyancy	3.Mech. systems
Types	Steady-state	***Fluctuating***			
Generators	Mean pressure differences	***Temporal variations in wind-induced pressures***			
Types		***a) Pulsating flow***	b) eddies		
Due to:		***Wind fluctuation and compressibility of air***	turbulence in air stream		

2. Method

The approach to developing an appropriate equation for fluctuating airflow is founded on the pulsating airflow model developed by Haghighat et al. who suggested summing up all the airflows caused by pressures of single sine-wave frequencies. Flow due to compressibility of air is introduced as the third element of the airflow system. This is developed into the modified fluctuating airflow model which includes an eddy-flow component. The aim was to develop a satisfactory CFD model of fluctuating flow through the crack which could be used to model divers pressure fluctuations and hence develop a fluctuating crack flow equation. However, physical validation of CFD simulations of fluctuating crack flow was considered a prerequisite to the development of an appropriate model.

2.1 Physical Testing

To find the airflow at various fluctuating pressures a test rig was developed (see Fig 1). The principal supplementary components to the steady-state system summarised by Baker et al. are the 4 variable speed fans. These permit a more extensive range of pressures to be applied to the crack surface. Also a damper was introduced to control volumetric airflow. The damper was a variable air volume regulator unit (LTG Lufttechnische VRE/SS/250) controlled by a servo drive damper motor (CMR DM10). The maximum rotation angle of the elliptical damper blade was 60°, supplying a volumetric air flow proportional to the percentage opening of the damper at higher pressures. The servo motor was driven by a compact PCB board which had an overload protection fitted to halt the servo motor without the use of end limit switches. This allowed input signals to be applied safely to the flow control unit. Two relays were utilised on the PCB board for the overall driving signal, served by a function generator. This provided a sinusoidal signal to drive the damper.

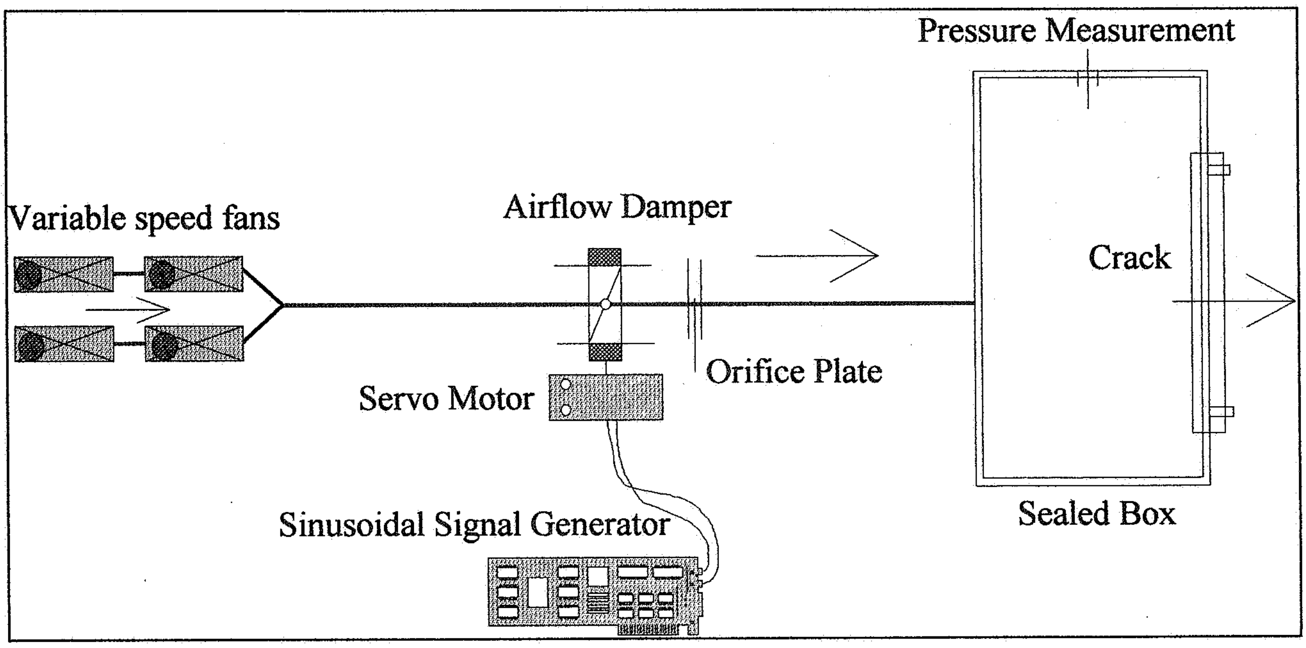

Fig 1 Schematic of Test Rig

The variable speed fans were ran at different power settings and the resultant pressure fluctuations were monitored at 0.1 second intervals for an initial driving signal of 0.66Hz. Fig 2 shows the resultant pressure fluctuations on the face of the wall section at various fan settings (Series 2-6 indicating progressively lower fan power) and records the consistency in pressure patterns. The flow through the crack was calculated from pressure readings from the orifice plate.

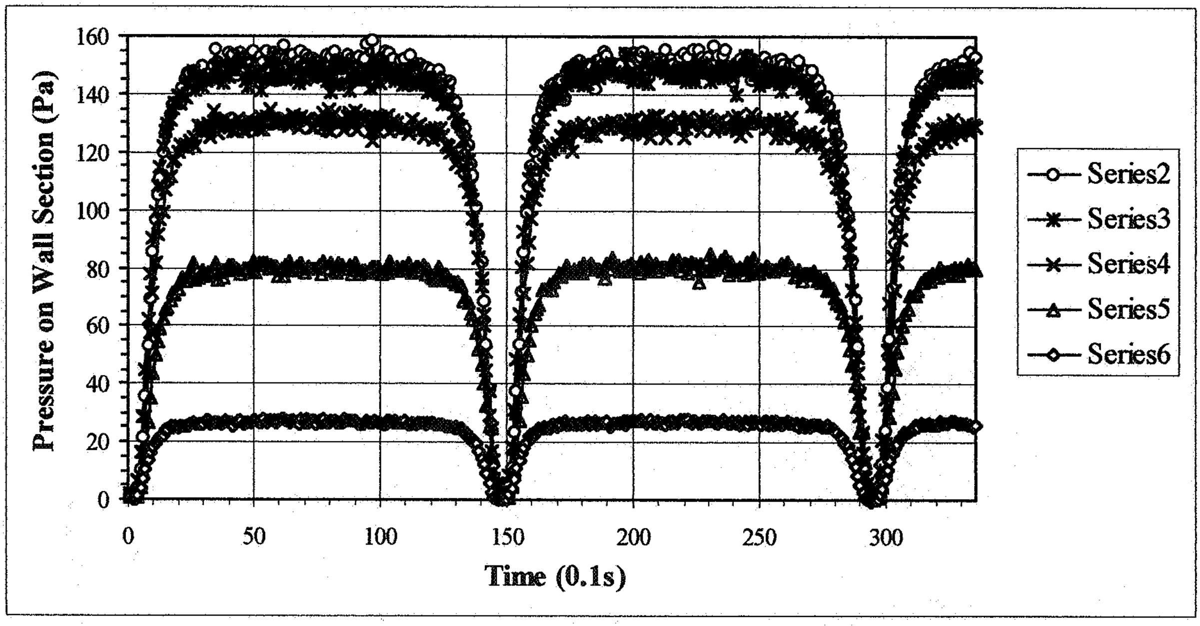

Fig.2 Resultant Pressure Fluctuations on Wall Section

2.2 CFD Modelling

A commercially available finite volume CFD code was used, FLOVENT. This had the capability to model airflow as laminar or turbulent and offered a k-ε solution. The FLOVENT solution method utilised in the analyses of the flow behaviour was fully validated both physically and mathematically before any fluctuating crack flow simulations were made [Tinker and Palmer.1994]. A three-dimensional model was developed in FLOVENT, using simple Cartesian co-ordinates and analyses executed in rectangular cuboid solution domains, where variables were solved using a Gauss-Seidel iteration For convergence to occur, maximum acceptable errors were defined that were realistically attainable and for final modelling these errors were characterised as very small proportions of the overall value of the variable in question. For example, the admissible termination residual for velocities, for which the error is found in the momentum equations, was designated as 1% of the sum of all momentum inflows. However, when overall values were very small the figure of 1% was eased due to the difficulties in achieving the minute values required.

Because temperatures anywhere in the rig were not significantly above the external conditions, density was kept constant throughout the space at $1.19 kgm^{-3}$, which allowed for a RH value of approximately 50% at STP. Also, the variations in air conditions experienced in the rig were relatively small and the specific heat of air in the CFD model was kept constant, at $1.005 x 10^3 Jkg^{-1} K^{-1}$, which again was consistent with air at STP. Under the same conditions the laminar dynamic viscosity was set constant to $1.84 x 10^{-5} kgm^{-1} s^{-1}$.

To expedite the solutions, after an acceptable resolution of each flow case had been found, overall flow boundary conditions were slightly adjusted and a new simulation was made based on antecedent results, i.e. values at individual grid cells were pre-set to existing values from the previous model. Although the initial variable values at each cell only superficially affected the final solution, they had a considerable influence on the rate of convergence of the model. Overall mesh sizes were limited by the computational power and data capacity of the hardware to 10-15k grid cells, depending on the type of solution. Analyses using a k-ε model requires greater capacity due to the larger number of parameters in the solution and so all preliminary grid definitions were kept as small as possible whilst providing an appropriate soluble model. All key surfaces were identified and grid lines defined along them. Up to 20 cells were defined for the fourth dimension, time, which allowed the fluctuating flows to be modelled.

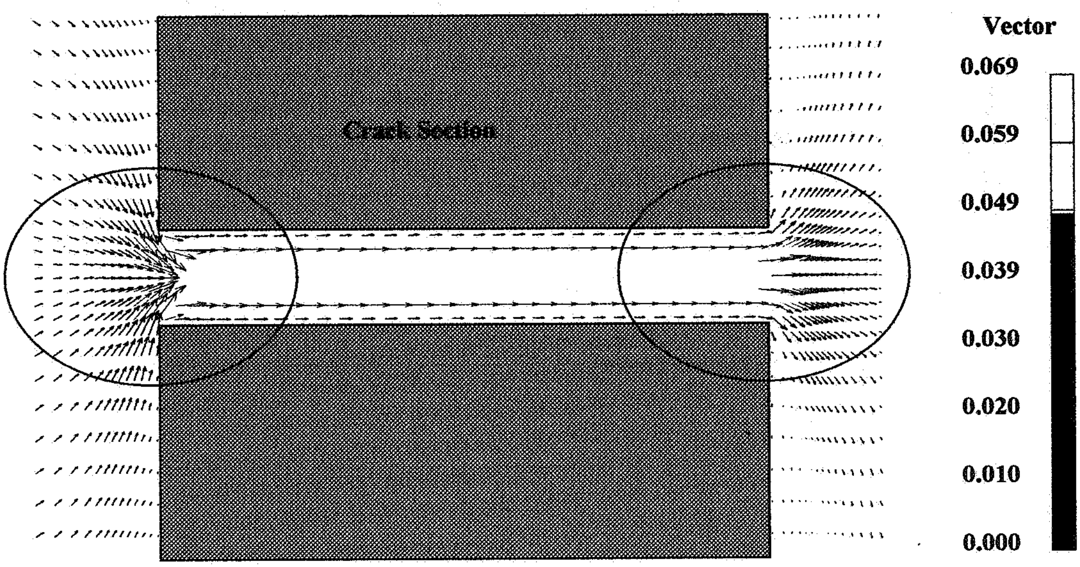

Fig 3 Section of a CFD Simulation of Wall Showing Crack Flow

An appropriate crack model was developed from a comparison of simulation results using various flow boundary parameters with physical tests. This would then enable simulations of models with sinusoidal pressure fluctuations. Fig 3 shows the CFD simulation of the straight crack and shows the airflow pattern at the inlet and outlet from the crack which proved to be an important indicator of appropriate solutions. Fig.4 shows different predicted volumetric flow rates and air distribution patterns for different pressure differentials applied across the same crack. The static pressure was varied according to measurements taken from the test rig and volumetric flow was then calculated from average outlet velocities and crack size.

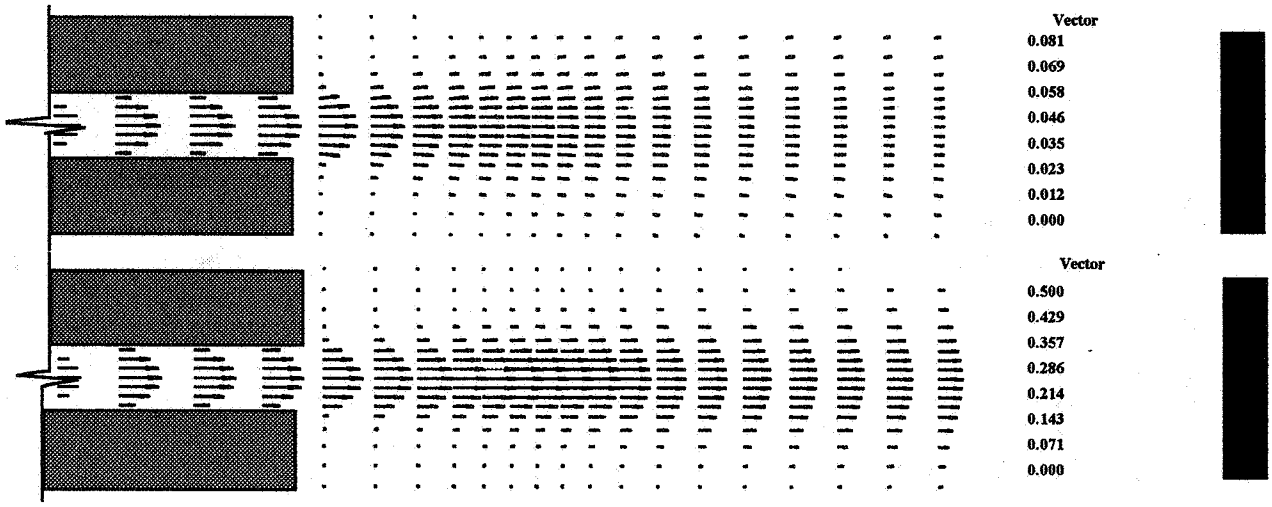

Fig.4 Outlet Conditions for Straight Crack - 20Pa and 140Pa Pressure Differentials

3. Results

For the preliminary test case, measurement of the pressure drop across the crack revealed a fluctuating pressure signal shown in Fig.5, varying from 0Pa to approximately 130Pa. A simultaneous measurement of the pressure differential across the orifice plate indicated flow through the crack succeeded the pressure fluctuation by approximately 0.4 seconds for this test apparatus. The hysteresis is illustrated in Fig.6 which shows a graph of the pressure differential across the crack versus the volumetric airflow through the crack.

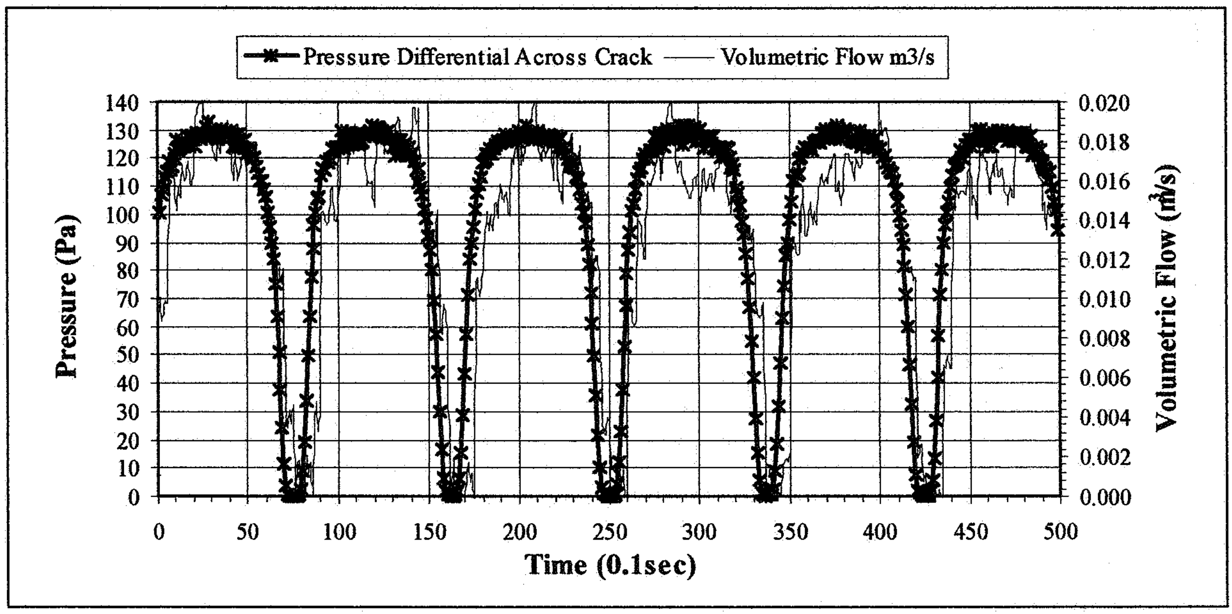

Fig.5 Pressure Differential Across Crack and Volumetric Flow Rate Through Crack vs Time

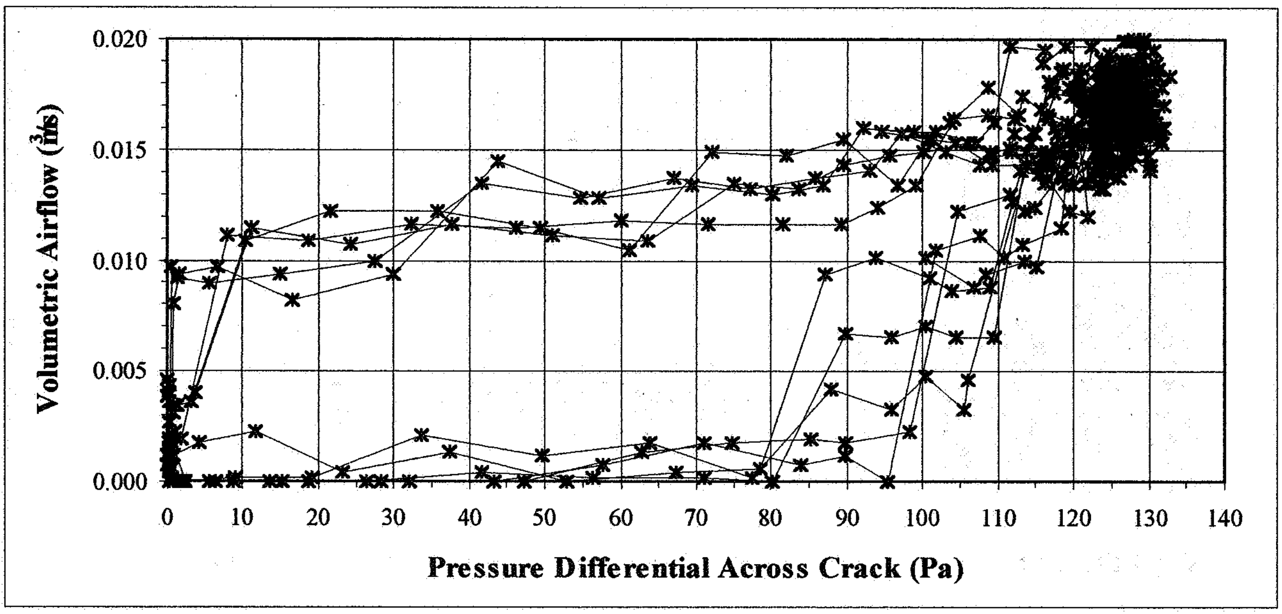

Fig.6 Pressure Differential Across Crack vs Volumetric Flow Rate

With the hysteresis effect removed, a comparison of experimentally found volumetric airflow against applied pressure differential is made for the fluctuating flow experimental results and the steady-state theoretical model and steady-state experimental results. This is shown in Fig.7.

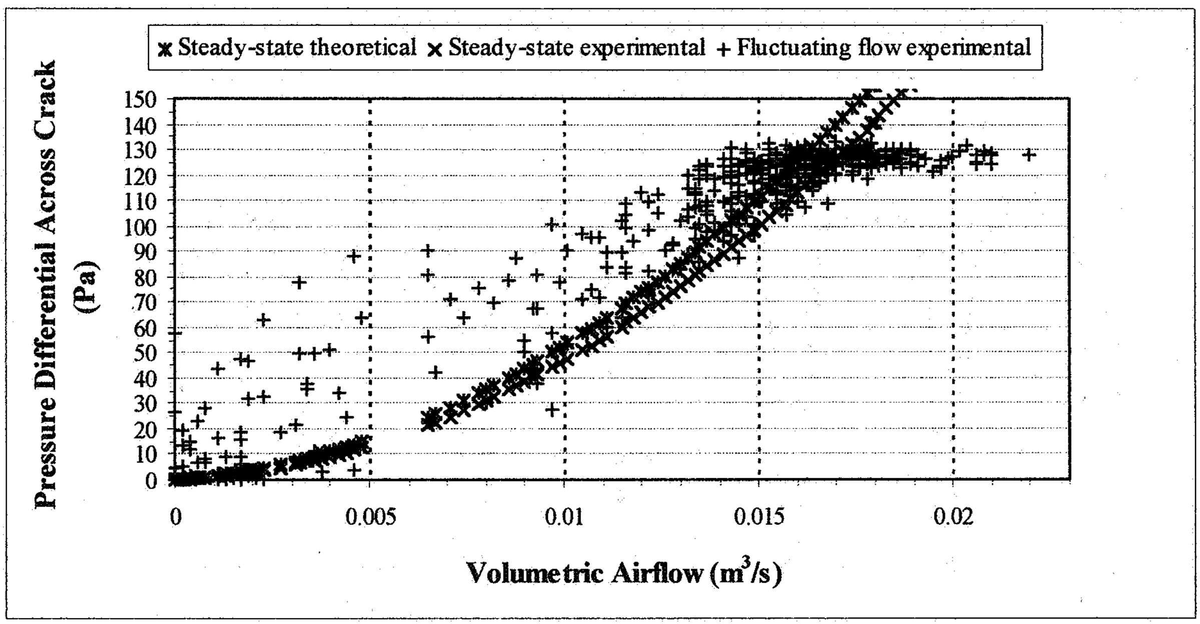

Fig 7 Volumetric Airflow Through Crack vs Applied Pressure Differential for Steady-state Theoretical and Experimental and for Fluctuating Experimental

The results indicate that, for lower pressures, volumetric airflow is lower in the fluctuating flow tests than it is for steady-state cases. However, the flow catches up with pressure fluctuations at the higher end of the pressure range and for no increase in applied pressure across the crack there continues to be an increase in volumetric flow. Initial CFD simulations show that appropriate crack flow modelling requires careful definition of the overall pressure on the face of the wall section. This was most correctly modelled using a static pressure inside the box. The pressure on the outside should be defined as ambient and a head loss of one dynamic head assumed. Defining volumetric inflow and outflow rates tended to over-estimate flow through the crack by in effect forcing air through without consideration of resistance of the crack itself. Initial fluctuating flow simulations show a correlation with experimental results for instantaneous maximum and minimum pressure differentials, but transient flow has proved more difficult to model. Further simulations are being carried out to prepare a model that can confidently model sinusoidal pressure fluctuations.

4. Conclusions

A methodology has been presented as an approach to evaluating air flow through simple cracks subject to fluctuating pressures. Initial results indicate that fluctuating pressures of varying frequency and magnitude can be simulated using this approach and that the resultant volumetric flows can be quantified. Similarly, comparative studies of results from the test rig have led to the formation of a reliable CFD model which can form the basis for further pressure simulations and ultimately sinusoidal fluctuations.

The results show the existence of considerable hysteresis in flow due to fluctuating pressure differentials and that quantification of this and overall volumetric flow should provide the basis for adjusting steady state crack flow equations for fluctuating flow.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Air-Tightness of U.S. Dwellings

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AIR-TIGHTNESS OF U.S. DWELLINGS[1]

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Blower Doors are used to measure the air tightness and air leakage of building envelopes. As existing dwellings in the United States are ventilated primarily through leaks in the building shell (i.e., infiltration) rather than by whole-house mechanical ventilation systems, quantification of airtightness data is critical in order to answer the following kinds of questions: What is the Construction Quality of the Building Envelope? Where are the Air Leakage Pathways? How Tight is the Building? How Much Ventilation Does the Air Leakage Supply? How Much Energy Does the Air Leakage Loose in this Building Too Tight? Is this Building Too Loose? When Should Mechanical Ventilation be Considered? Tens of thousands of unique fan pressurization measurements have been made of U.S. dwellings over the past decade; LBL has recently been collecting available data into its air leakage database containing over 12000 measurements. This report uses that data to determine the leakage characteristics of the U.S. housing stock in terms of region, age, construction type and quality. Results indicate that US dwellings tend to be quite leaky without respect to climate.

Keywords: Infiltration, Ventilation, Air Leakage, Indoor Air Quality, Energy, Blower Door, Fan Pressurization, Measurements

Lawrence Berkeley Laboratory Report No. 35700

1. This work was supported by the Assistant Secretary for Energy Efficiency and Renewable Energy, Office of Building Technology of the U.S. Department of Energy under contract no. DE-AC03-76SF00098.

1 INTRODUCTION

Virtually all knowledge about the air tightness of buildings comes from field measurements of *fan pressurization* using *Blower Door* technology. Blower Doors measure air tightness which, in turn, is the prime building factor in determining infiltration and *air leakage*. This report summarizes the measured air leakage data for U.S. dwellings.

This report does not intend to cover issues related to the fan pressurization measurements themselves. There exist many measurement standards[11] throughout the world, but the two referenced by the ASHRAE Standards relevant to much of the work in North America are the ASTM Standard[3] and the Canadian Standard[10]. Issues of measurement uncertainty[27] and reproducibility,[20] while important, will not be discussed in detail. Both technical[7] and popular[14,13] articles are available to familiarize the reader with some of the relevant issues.

This report focuses on single-zone buildings. While fan pressurization techniques are sometimes used for component or multizone leakage measurements, the vast majority of measurements have been made for whole-building, single-zone situations, such as single-family homes. The data summarized herein will deal with single-family homes throughout the United States for a wide variety of vintages, construction types, and conditions.

2 BACKGROUND

Air leakage data is now used for a wide variety of purposes from the qualitative (e.g. construction quality control) to the quantitative (e.g. envelope tightness standards). As the key envelope property related to air flow, leakage data is used in one form or another for infiltration-related modeling. Given such diverse uses, it is not surprising that this data is often treated as a stand-alone quantity, even though air leakage values are only intermediate value.

Before proceeding on to summarize the current measurements, it may be instructive to briefly review the history of fan pressurization measurements and their relationship to air flow modeling. Blower-Door technology was first used in Sweden as a window-mounted fan to test the tightness of building envelopes.[8] The technology was brought to the U.S. by Blomsterberg and used in Princeton to help "find and fix the leaks[16]," where it became a Blower *Door.*

During this period the diagnostic potential of Blower Doors began to become apparent. Blower Doors helped to uncover hidden *bypasses*[17] that account for a much greater percentage of building leakage than did the presumed culprits of window, door, and electrical outlet leakage. The use of Blower Doors as part of retrofitting and weatherization became known as *House Doctoring*[18,12] and led to the creation of instrumented audits[15] and computerized optimizations.[31]

While it was well understood that Blower Doors could be used to measure air tightness, the use of Blower-Door data could not be generally used to estimate real-time air flows under natural conditions. When compared with tracer-gas measurements, early modeling work[9] was found wanting. Attributed to (and often denied by) Kronvall and Persily,[22] there was a rule of thumb that seemed to relate Blower-Door data to seasonal air change data in spite of its simplicity:

$$ACH \approx \frac{ACH_{50}}{20} \quad \text{(EQ 1)}$$

That is, the seasonal amount of natural air exchange could be related to air flow necessary to pressurize the building to 50 Pascals, where "ACH" is the natural air changes per hour and "ACH_{50}" are the air changes induced by a 50 Pa pressure using a fan.

To overcome the physical limitations of such rules of thumb, it is necessary to physically model the system which, in this case, means separating the leakage characteristics of the building from the (weather) driving forces. As the early versions of the ASTM Standard show, leakage is conventionally described as a power law, which was found to be empirically valid but without theoretical substantiation (until recently[21]). Using orifice flow as a physical model, the Blower-Door data can be used to estimate the Effective Leakage Area (*ELA*).

Using this orifice-flow paradigm, the LBL Infiltration model[25] was developed and validated[26] and became incorporated into the ASHRAE Handbook of Fundamentals[2]. Much of the subsequent work on quantifying infiltration is based on that model, including ASHRAE Standards 119[3,23] and 136[4]. A more detailed description of how to use fan pressurization data is currently available.[30]

While ACH_{50} is a popular single-parameter quantification of leakage, the one used most by ASHRAE is called "Normalized Leakage", *NL*, which, like ACH_{50} can be calculated from fan pressurization measurements[5] (i.e. the exponent, *n*, and the Effective Leakage Area, *ELA*) and the building geometry (i.e. the floor area, A_f, and the building height, *H*):

$$NL = 1000\frac{ELA}{Af}\left(\frac{H}{2.5m}\right)^{0.3} \quad \text{(EQ 2)}$$

Blower Doors are still used to find and fix the leaks, but more and more the values generated by the measurements are used to estimate infiltration for both indoor air quality and energy consumption estimates. These estimates in turn are used to compare to standards or to base program or policy decisions. Each specific purpose has a different set of issues associated with it as it regards the use of the Blower-Door data. An earlier work[28] describes related data sources and their use in determining energy liabilities in more detail.

3 DESCRIPTION OF LEAKAGE MEASUREMENTS

The primary kind of data used in this report is, of course, leakage data. We required that all data in our dataset be of single-family detached dwellings from known locations in the U.S. In addition to air tightness data we required that the size of the dwelling and the number of stories be known. We requested, but did not always receive, more detailed information including the leakage exponent, the year of construction, the type of construction, floor/basement type and HVAC system, the building height, and any information regarding retrofits or general building condition.

Most of the data we used was not collected by the authors but was either published or volunteered by other researchers or practitioners. The largest sources of data consisted of 10800 houses from Alaska, Alabama, Vermont and Rhode Island, from Energy Rated

Homes of America. The largest *published* dataset used was the AIVC Leakage Database[19]. Those who volunteered published or unpublished data are listed in the "ACKNOWLDEGEMENTS" . We can summarize the dataset we have in a number of ways. Included in our database are 12946 individual measurements on over 12500 houses from the listed sources, including about 450 homes from the AIVC's numerical data base.

By its very nature the sample collected is not statistically representative of the almost 75 million single-family households in the U.S. Furthermore, different constituent datasets and measurements are of different qualities and should not be treated equally. Having said that, we must realize that this data represents the best set currently available and we shall use it to summarize the important physical characteristics contained in this database. Work continues on extrapolating this dataset to be representative of the U.S. housing stock.

4 RESULTS

We analyzed the data first to determine some overall trends in the leakage dataset without regard for the building properties and then we looked to the relationship between the details of the building and its leakage. Table 1, "SUMMARY OF LEAKAGE MEASURMENTS," summarizes the overall content of the dataset and contains the year of construction, the size of the dwellings and several variables relating the leakage information.

TABLE 1. SUMMARY OF LEAKAGE MEASURMENTS

Kind and Number of Measurements		Mean	Std Dev.	Min.	Max.
Year Built	1492	1965	24.2	1850	1993
Floor Area [m^2]	12946	156.4	66.7	37	720
Normalized Leakage	12946	1.72	0.84	0.023	4.758
ACH_{50}	12902	29.7	14.5	0.47	83.6
Exponent	2224	0.649	0.084	0.336	1.276

We can use the dataset to see if there is a useful correlation between the two ways of quantifying leakage. The average ratio[1] between ACH_{50} and NL is 17.5, with a standard deviation of 2.3, indicating that a 13% extra uncertainty can be introduced when converting directly between these two quantities. In general we will use Normalized Leakage rather than air changes at 50 Pascals to make our leakage comparisons.

The leakage values in Table 1 are averages of pressurization and depressurization values whenever both existed. One question that has often been posed is whether or not there is a significant difference between the two. We analyzed all of the cases in which both were measured and found that of the 280 usable measurements pressurization tests reported 9% higher leakage on average than did depressurization. As the error of the mean was 2% this difference is significant.The 9% value was calculated from the Normalized Leakage values. We repeated the analysis using the air changes at 50 Pascals and found the same trend but a larger average (i.e. 12%) value, but with a narrower distribution.

1. It should be noted that this ratio is only a relationship between to slightly different ways of summarizing airtightness and does not relate directly to Equation 1 for the calculation of infiltration.

This result suggested that there might be a difference in exponent between pressurization and depressurization, but our analysis shows that there was no statistically significant difference. We also looked at the general distribution of exponents and they appear quite clustered, even though there were many nonphysical outliers. The average exponent for the 1973 measurements that reported exponents is 0.65 with a standard deviation of 0.08

In the collection process data was sought from all over the U.S. So one important breakdown of the data we looked at was the examination of leakage by State. Our data does not include from some states (i.e. Delaware, Florida, Hawaii, Kansas, Kentucky, Louisiana, Maryland, Michigan, Mississippi, Nebraska, New Jersey, New Mexico, North Dakota, Ohio, Pennsylvania, South Dakota, Tennessee, Texas, Utah, West Virginia, Wisconsin, and Wyoming) but breaks down as indicated in Table 2, "NORMALIZED LEAKAGE BY STATE," for the other states. The last line of the table includes data in which the exact state was unknown.

TABLE 2. NORMALIZED LEAKAGE BY STATE

State	Average Normalized Leakage	Standard Deviation	N	State	Average Normalized Leakage	Standard Deviation	N
Alabama	0.85	.33	30	Minnesota	0.38	.21	2
Alaska	1.99	1.16	2830	Missouri	1.64	.45	11
Arizona	0.66	.49	5	Montana	0.14	.11	19
Arkansas	1.95	.98	551	Nevada	0.78	.49	4
California	0.73	.30	253	New Hampshire	1.13	N/A	1
Colorado	0.87	.35	13	New York	0.73	.58	282
Connecticut	0.50	N/A	1	North Carolina	1.48	.86	187
Georgia	1.57	.29	7	Oklahoma	1.12	.70	204
Idaho	0.50	.49	56	Oregon	0.40	.21	79
Illinois	0.66	.60	179	Rhode Island	1.88	.50	6284
Iowa	0.14	.07	2	South Carolina	0.78	.36	2
Indiana	0.39	N/A	1	Vermont	1.56	.55	1186
Maine	0.40	.10	3	Virginia	0.23	.05	2
Massachusetts	0.53	.22	3	Washington	0.44	.24	199
Northeast[a]	1.26	.78	467	*Other*[a]	0.72	.39	83

a. The se homes come from three studies in which the state was not identified: one in the New England (i.e. "*Northeast*". the other two from the Pacific Northwest and Iowa (i.e. "*Other*").

In examining regional trends we attempted to use regression techniques to determine if there were any leakage trends with climate, latitude, etc. Our analysis showed no significant trends with these climate-related parameters indicating the trends in leakage are more dominated by construction quality, local practices, age distribution, etc. than they are by weather. As an example, one can examine more extreme climates such as Alaska and Vermont which appear leakier than the mild climates such as California and Oregon, but other mild climates such as North Carolina appear quite leaky.

4.1 Relationship to Building Properties

We examined the dataset in some detail to look at five building criteria that may impact leakage: number of stories; floor/basement type; thermal distribution system; retrofitting; and dwelling age. We discuss below the impact of each of these factors.

Number of Stories: Most of the U.S. Housing stock is one and two story, single-family dwellings. We looked at the entire dataset to determine if differences in construction type affects the leakage. Approximately 56% of our measurements are of multistory dwellings. We find that multistory houses are 11% leakier than single-story houses with an error of the mean near 1%. This value is, therefore, statistically significant, and we can conclude that there is a difference between single and multiple storied dwellings.

Floor/Basement Type: We restricted our consideration of this issue to two classes: those dwellings that had floor leakage to outdoors (i.e. crawlspace homes and unconditioned basements) and those that had no floor leakage to outdoors (i.e. slab-on-grade and fully conditioned basement homes). The vast majority (80%) of our dataset had floor leakage. The subset that did not was slightly (5%) tighter and this value was statistically significant.

Thermal Distribution System: Because of the current interest in the efficiency of residential thermal distributions systems, we analyzed those (1442) homes where there was knowledge about the existence (or absence) of a duct system. The surprising result was that the homes with duct systems (43% of this subset) were tighter (NL=0.7) than those homes that did not have duct systems (NL=0.9). Where duct systems were measured separately (only about 130 homes), they accounted for just under 30% of the total leakage--a finding consistent with other studies.

Retrofitting: A (465 house) subset of the houses were measured as part of retrofit or weatherization projects and had measurements both before and after the retrofits were done. From these measurements we found that the average retrofit reduced the leakage by about 25% (from NL=1.34 to NL=0.99 with the error of the mean difference being NL=0.03).

Dwelling Age: We examined that data for which the year of construction was available to see if there were leakage trends correlating to the age of the dwelling. Examining the data in detail we found a break point at the year 1980. The 628 houses built after 1980 did not show any trend with age and were tighter (NL=0.47) than average. The 869 houses built prior to 1980 showed a clear increase in leakage with increasing age and were on average leakier (NL=1.05) than new houses but still tighter than the average of the entire dataset.

5 DISCUSSION AND CONCLUSIONS

The first significant finding is that dwellings appear to be even leakier than previously estimated. This current analysis includes large datasets that represent much more comprehensive cross-sections of ordinary homes in particular locations (e.g. Rhode Island, Alaska, Vermont etc.) than had been previously studied. Although not spread evenly around the country these more intensive studies suggest that our previous leakage estimates were biases towards tighter housing, probably because more energy efficient houses have been studied in detail.

Unlike the impact of leakage, floor/basement type and the number of stories, the impact of ducts, the effect of retrofits, and year of construction information is available on only subsets of the data. Furthermore these subsets themselves appear to be tighter than the dataset as a whole, probably reflecting the fact in the larger, broader studies, less information was recorded and that the detailed studies probably tended to be on better construction.- Our previous study[28] had indicated that approximately half the U.S. would meet ASHRAE's airtightness standard[3]. This dataset, although not statistically representative, has less than 10% of the country meeting that standard-indicating that the stock may be leakier than previously estimated.

We examined the data subsets in many ways and looked at distributions of various quantities. In almost every distribution there were more outliers than would be expected from a normal distribution; some of them were nonphysical and induced most likely by measurement problems such as weather effects or mismatches between equipment capacities and dwelling conditions. Outliers may also be caused by data entry errors.

Table 2, "NORMALIZED LEAKAGE BY STATE," clearly indicates that more data is needed in certain areas of the country. Our future efforts will be to try to fill these data gaps and then use the kind of statistical techniques we have used before[28] to extrapolate the information to the country as a whole.

6 ACKNOWLDEGEMENTS

The authors would like to acknowledge the contributions of leakage and related data made by individuals and organizations. Table 3, "LIST OF DATA CONTRIBUTORS USED IN THIS REPORT," includes those sources for which data was included in our analysis. In addition to those listed below there are many individuals that have sent in data on

TABLE 3. LIST OF DATA CONTRIBUTORS USED IN THIS REPORT

CONTRIBUTOR	INSTITUTION	REGION
Ron Hughes, Evan Brown	Energy Rated Homes of America	Alaska, Arkansas, Rhode Island, and Vermont
Kenneth Wiggers	American Radon Services,	Iowa
Mark Ternes	Oak Ridge National Lab	Northeastern States, and Oklahoma
Terry Sharp	Oak Ridge National Lab	North Carolina
Rose Girer-Wilson	University of Illinois	Illinois
Bill Levins	Oak Ridge National Lab	Northeastern States
Larry Palmiter &Tami Bond	Ecotope	Pacific Northwest
Bruce Wilcox	Berkeley Solar Group	California
Victor Espanosa	Las Angeles Dept. of Water & Power	California
Peter Strunk	Synertech	New York
Bob Carver, Bob Kelly	New York State ERDA	New York
Matson, Jump, Modera	Lawrence Berkeley Labs	California
Liddament et al.	Air Infiltration and Ventilation Centre	U.S. Wide

paper only, and have not been entered in as of this time. It is our intent to continue with the data entry as time permits, starting with areas of the country which are under-represented. The data presented here represents a small fraction of the measurements taken and it is hoped that further sources will be developed.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Energy Efficient Ventilation of Bathrooms

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Synopsis

This paper reports on the findings from two extensive laboratory studies of ventilation of bathrooms of different sizes and layout of ventilation. The ventilation flow rates were varied. Moisture production were due to laundering and shower baths. In one of the studies the bathroom was provided with a drying cabinet. The extract air was forced to pass through the drying cabinet which was connected to the extract ventilation system of the house by a duct running from the cabinet to the extract air terminal device. Two alternative connections of the drying cabinet to ventilation system of the house were tested; a standard hose directly connected to the extract terminal device (direct extraction) and a hose wich ends just before the terminal device (indirect extraction). To save energy the drying cabinet was not heated but provided with a small mixing fan to enhance the mixing within the cabinet. Two alternative locations of the intake of air to the drying cabinet were tested. The humidity, air-exchange efficiency and local mean age of air were monitored during the test. The drying of the wash was recorded by continuously recording the change of weight of the wash.
Selection of layout of ventilation was based on results of recorded distribution of local mean age and the target for moisture exposure was to minimise the time of exposure of relative humidity higher than 70 %.

Introduction.

The bathroom is the main principle area of moisture and water vapour production. In Sweden people have changed their washing habits and people living in multi family houses want to do the daily washing of their cloths in their bathrooms instead of using the washhouse belong to the house. To dry the laundry they install drying cabinets. Drying of the laundry is the most energy consuming part of the washing "process". Therefore if unheated drying cabinets instead of heated cabinets are installed a substantial amount of energy will be saved (for Sweden the saving is estimated to be between 0.5 -1 Twh) . In an unheated drying cabinet the drying of the laundry is arranged by forcing the ventilation air to pass the drying cabinet. During the passage of the air through the drying cabinet its temperature is lowered. The challenge is to provide an adequate ventilation of both the room and the drying cabinet. If the ventilation does not work properly severe damage to the building may be caused by condensation on surfaces and long time exposure of high relative humidity levels.

Means to control the humidity in the bathroom

To limit the risk for mould growth etc. the aim is to keep the time of exposure to high relative humidity levels as short as possible. There is a risk for mould growth to occur when the relative humidity is higher than 70 % therefore in this investigation the target was put to minimise the time of exposure to relative humidity higher than 70 %. One has the following control methods:

-Source control

-Ventilation
Flow Rate
Distribution of air

-Temperature
Surface temperature
Room air temperature

Drying of the wash in a separate drying cabinet is one example of source control. By dilution of air with less content of humidity than in the bathroom the relative humidity will be reduced. Efficient distribution of the supplied air is a prerequisite for an energy efficient ventilation. By heating of the room air the moisture holding capacity of the air will increase and the relative humidity will be lowered. Local heating of the most exposed parts, e.g. under the bathtub, is one possibility. However, all kind of heating requires energy.

Where may the problems occur ?

The most vulnerable point in the room are the parts of the room surfaces with the lowest temperature and regions with bad ventilation. The internal surface temperatures depend on the quantity of heating, on the indoor and outdoor temperatures and on the thermal resistance of the external walls. Cold bridges does not always occur at joints. Some types of wall constructions have their lowest thermal resistance at the centre of the wall. However, the *lowest temperature* occurs usually at the lowest parts of the wall, see Figure 1. This is due that the loss of heat gives rise to a downdraught of air along the wall surface.
After a shower bath the walls are covered with a film of water which is transported towards the floor. Therefore the floor area is the part most exposed to humidity. Due to the presence of obstacles the air may have difficulties to reach under the bath tube. Room corners may be less ventilated due to that the primary air stream is deflected before it reaches the corners[2].

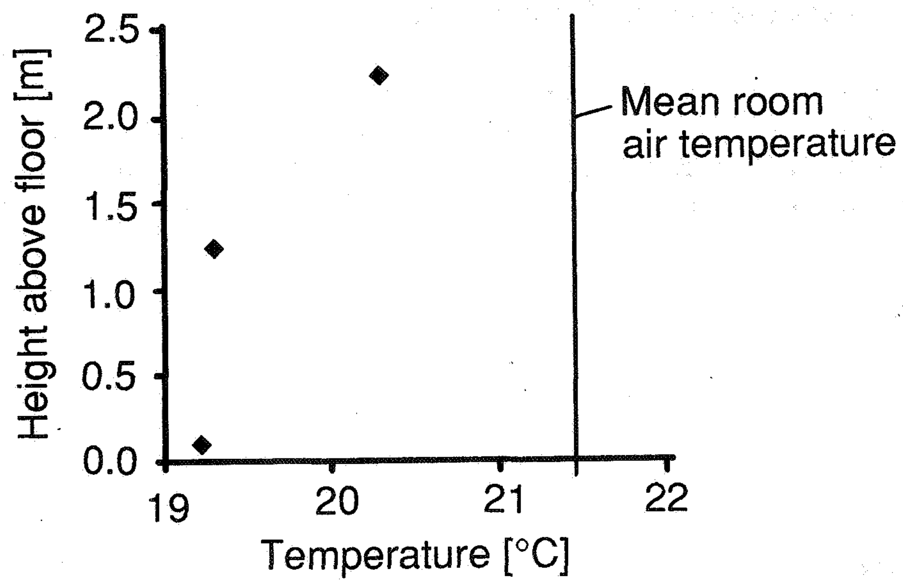

***Figure 1** Example of recorded internal surface temperature of an exterior wall. (from small bath room in Fig. 2)*

Description of test facilities

The tests were accomplished in the indoor testhouse located in the laboratory. One facade of the testhouse is the facade of the laboratory. The testhouse is described in [1]. Two sizes of bath rooms were investigated, see Fig.2.

The pertinent data for the bathrooms is given in the table below.

Bathroom	Volume $[m^3]$	Flow Rate $[m^3/h]$	Flow Rate [litre/s]	Nominal Time Constant [minutes]
Large	13.2	30	8.3	26.4
		55	15.3	14.4
Small	7.8	18	5.0	26.0
		54	15.0	8.7

The door was always closed in all tests to represent a "worst case" situation. The bathroom is ventilated by an extract fan and incoming air comes from the other rooms of the house via purpose made openings between the door and the doorframe. The bathroom shown at bottom of Fig. 2 is provided with a drying cabinet connected indirectly to the extract thermal device of the room (indirect extraction). The main idea behind using such type of connection is to make the drying cabinet independent of the ventilation system of the house in the sense that it will not give rise to any pressure drop.

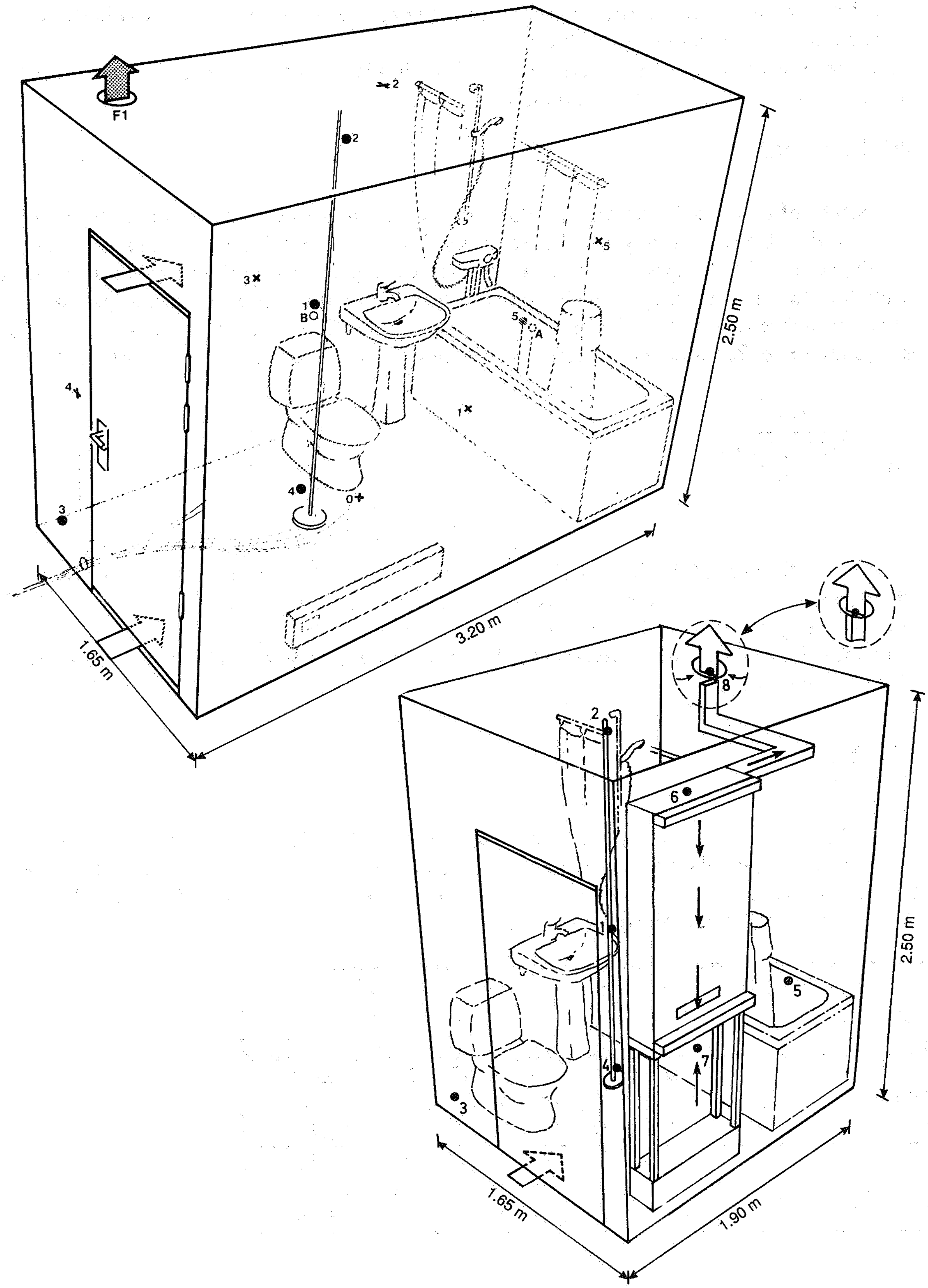

Figure 2 Large bathroom without drying cabinet
Below: Bathroom provided with drying cabinet with indirect extraction.
Inset shows direct extraction.

It will thereby not influence the balancing of the ventilation system. As an inset is shown a standard direct connection with a hose..
In the tests with the small bathroom the humidity in the remaining rooms were controlled and the target was set to 50 % relative humidity.

Air Distribution

The intake of air to the bathroom was through a 2 cm gap between the door and the doorway. This implies that one has a type of three-dimensional diffuser with width equal to the door width (71 cm). The temperature in the bathroom was higher than in the neighbouring rooms. Therefore the air streaming through the gap enters the room as a *non-isothermal jet* and is therefore influenced both by inertia and buoyancy forces. The relative strength of inertia and buoyancy forces are grouped together into the non-dimensional *Archimedes number Ar*

$$\mathrm{Ar} = \frac{g\dfrac{\Delta\theta\sqrt{A}}{(273.33+\theta)}}{U^2} \tag{1}$$

Where A is the area of the "diffuser", U is the mean velocity and $\Delta\theta$ is the temperature difference between the incoming air and the room air. With increasing distance from the supply a buoyant jet is gradually more influenced by the buoyancy forces. The transition to a buoyancy dominated flow is characterised by a quantity known in fluid mechanics as the *thermal length* l_m.

$$l_m = \frac{\sqrt{A}}{\mathrm{Ar}} \tag{2}$$

If this thermal length is much longer than the *perimeter,P,* of the room in the direction of the flow one can expect the buoyancy having minor influence on the flow. The table below gives the pertinent data of the supply device and the supply velocity U, Reynolds number Re, Archimedes number and the thermal length.

q [l/s]	*U* [m/s]	*Re* [1]	$\Delta\theta$ [°C]	*Ar* [1]	l_m [m]	l_m/P [1] Small	Large
5 l/s	0.35	2 800	0.5	0.0175	6.85	0.78	0.60
			1	0.035	3.42	0.39	0.30
			2	0.070	1.71	0.19	0.15
15 l/s	1.06	8 480	0.5	0.0019	63.0	7.16	5.52
			1	0.0038	31.5	3.58	2.76
			2	0.0076	15.7	1.78	1.38

Figure 3 shows for the large bathroom tests with different locations of the intake of air to the bathroom. The *local mean age of air* $\overline{\tau_p}$ was recorded at the points indicated by the filled circles. Standard decay method employing active tracer gas technique was used. Of special interest is measuring point No 5 which is located behind the bathtub. The result is presented in terms of the *local air-exchange index* according to the following definition

$$\varepsilon_i = \frac{\tau_n}{\overline{\tau_p}} \bullet 100 \cdot \ [\%] \qquad (3)$$

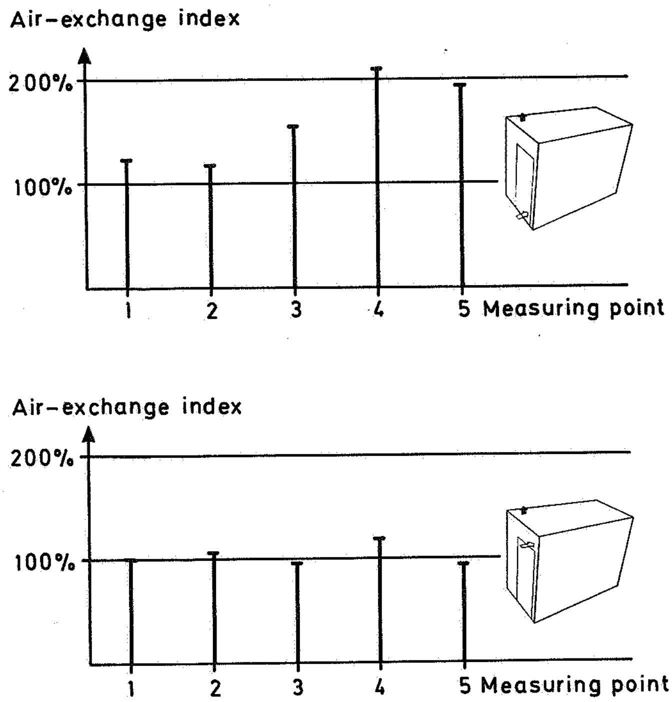

***Figure 3** Large bathroom. Recorded distribution of the local mean-age air for two locations of intake of air. Nominal time-constant 0.24 hours.*

One sees that with the intake located below the door the air arrives behind the bathtub twice as fast as with the intake above the door. Based on these results it was decided to locate the intake of air below the door.

The next figure shows the recorded air-exchange efficiency for the smaller bathroom and with an unbroken connection between the drying cabinet and the extract device of the ventilation system of the room. Therefore the intake of air to the cabinet is the extract point of the air from the room. The intake of room air to the drying cabinet could be located at two alternative positions, at the top of the cabinet or at the bottom of the cabinet. From the upper intake the room air was led to the bottom of the cabinet, see Fig. 2, and therefore the room air entered always the cabinet at its bottom.

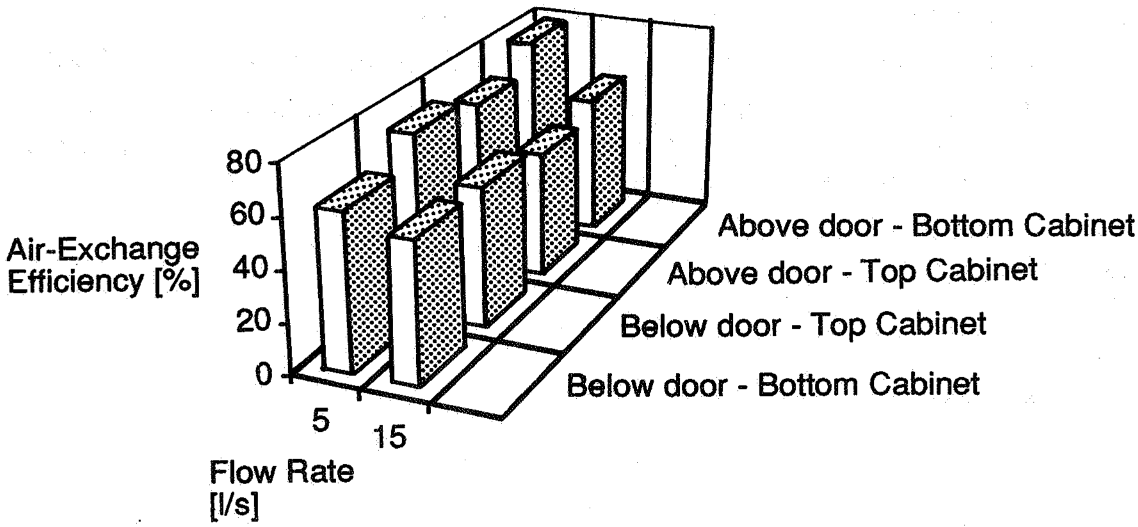

Figure 4. Recorded distribution of local mean age of air.

At the highest flow rate one obtains an even distribution of the air whereas at the lowest flow rate one obtains a more favourable air flow pattern. This can be explained by the analysis in the previous subsection. At the lowest flow rate there is an influence of the buoyancy whereas at the higher flow rate the flow is purely momentum *driven.*

Exposure to humidity

Figure 5 summaries the results obtained for the smaller bathroom provided with a drying cabinet witch was directly connected to the ventilation system of the house.

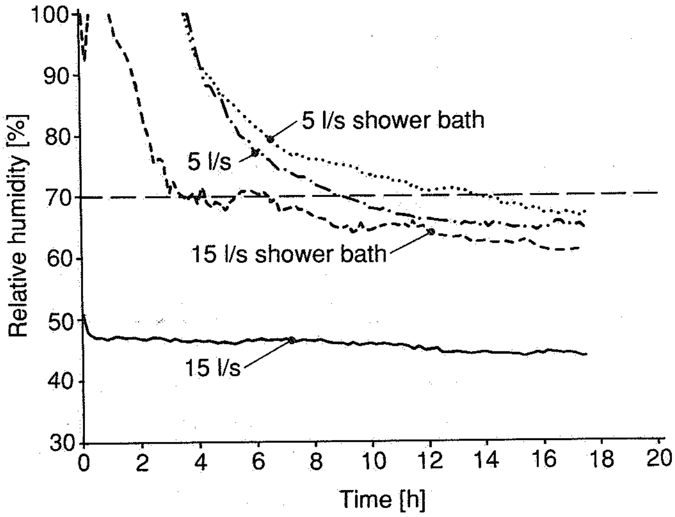

Figure 5 Recorded relative humidity behind the bath-tub at different ventilation flow rates. Duration of shower bath 5 minutes

When the flow rate amounts to 15 l/s the target is met except when taking a 5 shower bath. However the exposure to high relative humidity larger than 70 % is fairly short.

Conclusions

When drying the wash in the bathroom it was possible to meet the target set up for the maximum level of the relative humidity (70 %) behind the bathtub by taking the following steps:

- Install a drying cabinet directly connected to the extract ventilation system
- Flow rate of 15 l/s
- Intake of air to the bathroom at floor level

The drying cabinet was unheated but provided with a small mixing fan (30 watt) in order to enhance the distribution of air within the drying cabinet. The drying cabinet introduced a pressure drop of less than 20 Pascal.
When simultaneously drying the wash in the cabinet and taking a 5 minutes shower bath the air becomes saturated and the target was exceeded during a time period of about two hours.

Acknowledgement

The authors´ would like to acknowledge Anders Mellin for assisting in doing the measurements. This project has been sponsored by NUTEK under the program for technology procurement for promoting innovations.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

The Relative Energy Use of Passive Stack Ventilators and Extract Fans

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THE RELATIVE ENERGY USE OF PASSIVE STACK VENTILATORS AND EXTRACT FANS

by M Woolliscroft MSc, BSc(Econ), C Eng, M.I. Mech E, M CIBSE

SUMMARY

The relative energy use of PSV and extract fans has been a matter of considerable controversy, particularly in the UK. A steady state methodology is presented based on the approach of BS5250 and that of Professor Meyringer (Air Infiltration Review November 85). The ventilation, over and above background ventilation, required to remove moisture is shown to be affected by; the rate of moisture production in the dwelling, the moisture content of the outside air, the air temperature of the dwelling, the air tightness of the dwelling, the moisture absorption of the structure and furniture, the dwelling size, whether trickle vents are open or closed, the proportion of moisture removed in the kitchen or bathroom.

Equations are derived for the energy used by PSV, both uncontrolled and humidity controlled and by humidity controlled extract fans. Manually controlled systems, either fans or PSV, have not been considered because their use depends on human behaviour and there is, as yet, a lack of detailed reliable data. Conditions are determined for one or the other to be the greater. The effect of varying the above variables and opening or closing the kitchen door is investigated. It is shown that PSV both controlled and uncontrolled, has an energy advantage in heavily occupied, cold and small dwellings. In average dwellings there is little difference and extract fans are more energy efficient in large, warm and lightly occupied dwellings.

This paper is to be presented at the AIVC Conference
Buxton September 1994

July 1994

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THE RELATIVE ENERGY USE OF PASSIVE STACK VENTILATORS AND EXTRACT FANS

LIST OF SYMBOLS

Q_1	Airflow rate required to remove moisture m^3/day
ρ_{air}	Density of air kg/m^3
G	Moisture generation rate kg/day
g_{inside}	Moisture content of the inside air kg/kg
$g_{outside}$	Moisture content of the outside air in kg/kg
α	Proportion of moisture extracted in kitchen and bathroom by extract fan
B_f	Flow reduction factor fan = additional ventilation in dwelling/flow through fan
B_{psv}	Flow reduction factor PSV = additional ventilation in dwelling/flow through PSV
Q_2	Background ventilation rate m^3 /day
Q_3	fan flow rate m^3/day
g_{70}	moisture content of inside air corresponding to 70% RH at the inside air temperature kg/kg
ΔT	temperature difference inside to outside ° C
E_1	efficiency of boiler or heating appliance
P	fan power watts
E_2	efficiency of electricity generation and distribution %
Q_4	PSV flow rate m^3/day
γ	proportion of the time for which the fan runs
T_{in}	inside temperature °C
T_{out}	outside temperature °C
RH	relative humidity %
RH_o	RH at ref point %
T_o	inside temperature at reference point
g_o	moisture content of inside air at ref point kg/kg
A_1	open area of humidity control device cm^2
A_o	open area of humidity control device at RH_o cm^2
A_2	area of stack cm^2
k_1	loss coefficient of humidity control device
k_2	loss coefficient of rest of stack system

1. INTRODUCTION

In the UK the pollutant controlling the ventilation rate is moisture and thus the energy used in ventilation is dependent on the process of moisture transfer and removal. This is a dynamic process operating on a multi cell flow process. The analysis which will be presented in this paper will however be steady state, based on average daily moisture production rates and essentially single cell. A more complex analysis will be developed in due course. The additional ventilation required to remove moisture over and above the background ventilation of the dwelling is influenced by a number of factors, some of the most important are:

the rate of moisture production in the dwelling
the moisture content of the outside air
the air temperature of the dwelling
the air tightness of the dwelling
the moisture absorbtion of the structure and furniture
the dwelling size
whether trickle vents are open or closed
the proportion of moisture removed in the kitchen or bathroom

Manually controlled systems have not been considered because their use and hence relative performace, is dependent on human behaviour and there is little information available. An important factor in the calculations is that the effect of an extract fan or a passive stack ventilator is not simply additive to the total ventilation. The addition of the extract fan or PSV changes the pressure distribution in the dwelling. Calculations using the single cell ventilation model BREVENT (ref 1), suggest a reduction factor to the flow through the device of about 0.5 applies for PSV and 0.6 for fans in the range of interest. Initial calculations are given for uncontrolled PSV and then later for controlled PSV.

2. THE CALCULATIONS

2.1 Energy calculations have been carried out for a range of rates of moisture production up to 16kg per day, the maximum for a very wet household given in BS 5250 (reference 4). The calculations have been carried out month by month over a heating season from October to April inclusive taking average conditions of outside moisture and outside temperature for each month. Energy use has been calculated for three mean household temperatures, 14° C, 16°C and 18°C. (Results are only shown for 16°C and 18°C). These are typical figures for the UK from the English House Condition Survey (refs 2 and 3)

2.2 The calculations have been carried out on the basis of the equations below. The calculations at this stage are for uncontrolled PSV. The flow rate required to remove the moisture generated is given by:

$$Q_1 = \frac{G}{\rho_{air}(g_{70} - g_{outside})} \qquad (1)$$

from Milbank reference (5) which is the same as the methodology given in BS5250 reference (4) and Meyringer reference (8), (although the precise equation is somewhat different).

If a proportion α of the moisture is removed in the kitchen, and if the extract rate of the fan is Q_3 m^3/day then the proportion of the day for which the fan runs will be

$$\alpha \frac{Q_1}{Q_3} \qquad (2)$$

The remainder of the moisture will be absorbed and/or will spread around the rest of the dwelling to be desorbed later. This moisture may require the fan to run to provide additional ventilation. When the extract fan (or PSV) is applied to the whole house then for the reasons given earlier the additional flow is less than the actual fan flow rate. If we call the reduction factor B then if Q_2 is the background ventilation rate in m^3/day then the proportion of the day for which the extract fan will run in order to clear the rest of the moisture is given by:

$$\frac{(1-\alpha)\ Q_1 - Q_2}{B_f\ Q_3} \tag{3}$$

If the fan is operated by a humidistat this quantity cannot be negative.

The energy use arising from the fan airflow may be expressed by:

Fan airflow energy use =

$$\rho_{air}\ \frac{B_f\ Q_3}{E_1}\left[\frac{(1-\alpha)\ Q_1 - Q_2}{B_f\ Q_3} + \alpha\ \frac{Q_1}{Q_3}\right] \Delta T \quad (KJ/day) \tag{4}$$

where the specific heat of air is taken as 1. KJ/Kg °K

The primary energy used by the fan is given by

$$\frac{P}{E_2}\ 24 \left[\frac{(1\ -\ \alpha)\ Q_1\ -\ Q_2}{B_f\ Q_3} + \alpha \frac{Q_1}{Q_3}\right] 3.6\ KJ/day \tag{5}$$

The uncontrolled PSV energy use is given by:

$$\frac{\rho_{air}\ B_{psv}}{E_1}\ Q_4\ \Delta T \qquad KJ/day \tag{6}$$

These equations have been used to plot figures 1-3. The PSV system consists of a 125mm PSV in the kitchen and 100mm PSV in the bathroom. The fan system comprises a 68 litre/sec fan in the kitchen and a 25 litre/sec fan in the bathroom. Total fan power is 75W. The efficiency of electricity generation and supply is taken to be 30% and that of the boiler as 80%. The airtightness has been taken as 7 AC/hour at 50 pa in figures 1 and 2, which is fairly tight but likely to be typical under the revised UK Building Regulations and 10 AC/hour at 50 pa in fig 3. No direct account has been taken of moisture absorbtion as such although the steady state average calculation implies absorbtion and desorption. A 200m^3 dwelling has been chosen. Calculations were carried out with trickle vents closed. The corresponding whole house airflow rates were obtained from BREVENT taking a 4m/s wind speed, and background airflow rate was adjusted each month for temperature. It has been assumed that 50% of moisture is removed in the kitchen when the fan is running. It can be seen that PSV uses less energy with colder more airtight dwellings and where the rate of moisture production is high.

2.3 Another approach to the problem is to calculate the level of G, the moisture production rate at which PSV and extract fan energy use are equal. In general this will be done for seasonal averages. From equations (4), (5) and (6) PSV energy use is greater than extract fan energy use if:-

$$\frac{\rho_{air} B_{psv}}{E_1} Q_4 \Delta T > \frac{\rho_{air} B_f Q_3}{E_1} \left[\frac{(1-\alpha)\, Q_1 - Q_2}{B_f Q_3} + \alpha \frac{Q_1}{Q_3} \right] \Delta T \tag{7}$$

$$+ 86.4 \frac{P}{E_2} \left[\frac{(1-\alpha)\, Q_1 - Q_2}{B_f Q_3} + \alpha \frac{Q_1}{Q_3} \right]$$

If we call the proportion of the time for which the fan runs γ

$$\left[\frac{(1-\alpha)\, Q_1 - Q_2}{B_f Q_3} + \alpha \frac{Q_1}{Q_3} \right] = \gamma \tag{8}$$

then PSV energy > fan energy if

$$\frac{\rho_{air}}{E_1} \Delta T \,(B_{psv} Q_4 - \gamma B_f Q_3) > 86.4 \frac{P}{E_2} \gamma \tag{9}$$

If equation 9 is made into an equality it can be solved for γ and hence from equations (8) and (1), G_o can be obtained.

2.4 <u>The effect of varying the proportion of moisture removed in the kitchen</u>

From equations (1) and (8) it can be shown that where $Q_2 < (1 - \alpha)\, Q_1$

$$\frac{1}{\frac{1}{B_f Q_3} - \alpha \left(\frac{1}{B_f Q_3} - \frac{1}{Q_3}\right)} \tag{10}$$

G is proportional to Q_1 which is proportional to

Thus the smaller α, the lower the value of G_o . Where however, $Q_2 > (1 - \alpha) Q_1$, G_o is proportional to α^{-1} and the smaller α the larger G_o.

This is illustrated in the table below for T_{inside} = 16° C, $T_{outside}$ = 7° C, outside moisture = 5.5 g/kg and a house of 7AC/hour at 50pa.

α	0	0.25	0.5	0.75	1.0
G_o	6.66	7.40	7.03	4.18	3.13

Table 1 - The effect of changing the proportion of moisture removed in the kitchen

There is little data which might be used as a guide to the appropriate value of α. It might be reasonable to assume that half of the moisture generated in the kitchen or bathroom is removed in the kitchen or bathroom. A value nearer to 0.25 might be more appropriate. However, as can be seen from the table, it will not make a large difference to G_o. The effect of closing the kitchen door is of course equivalent to $\alpha = 1$.

2.5 Humidity Controlled PSV

For controlled PSV the flow rate has been calculated from the relative humidity obtained by linear interpretation of the psychometric chart and the consequent open area of the humidity control device using the following equation:

$$g_{inside} = g_{70} - (Q_2\ \rho_{air}\ (g_{70} - g_{outside}) - G\ )\ /Q_2\ \rho_{air} \tag{11}$$

From linear extrapolation on the psychometric chart

$$RH = RH_0 + A\ (T_o - T_{ins}) + D\ (g_{inside} - g_o\ ) \tag{12}$$

where T_{in} is the temperature in the house in °C

A and D are constants.

The open area of the humidity control device is given by:

$$A_1 = A_0 + (RH - RH_0)\ \frac{\Delta A}{\Delta RH} \tag{13}$$

where Δ A is the range of area over the RH range Δ RH.

The PSV flow rate is given by:

$$Q_4 = B_{psv}\sqrt{\frac{2\Delta p}{\rho_{air}\left(\frac{k_1}{A_1^2}+\frac{k_2}{A_2^2}\right)}} \qquad \textbf{(14)}$$

Where k_1 is the loss coefficient of the humidity control device and k_2 is the loss coefficient of the rest of the system. Δp is the pressure generated by the stack effect. These coefficients were obtained from experimental data. A 4m/s wind speed was assumed throughout but flow was adjusted for changes in outside temperature. The calculation was iterated after the initial calculation of PSV flow rate using the initially calculated PSV flow rate adding this to the background ventilation rate and substituting in equation (11). A couple of iterations were carried out until the calculated PSV flow rate stabilised. The results are also shown in figures 1 to 3. The effect is to lower the cross over point between PSV and fans in terms of moisture production rate but perhaps more important to reduce significantly the difference between PSV and extract fans at low moisture production rates.

3. DISCUSSION

All the eight factors described in the introduction have a significant effect on the relative energy use of PSV and extract fans. Whilst calculations have not been shown for building volume, the effect is similar to that for airtightness by lowering or raising the background ventilation level. Moisture absorbtion has not been illustrated, but the effect is implicit in the use of a steady state analysis. Overall PSV is relatively more energy efficient:

the higher the moisture production rate
the higher the moisture content of outside air
the lower the temperature of the dwelling
the more airtight the dwelling
the lower the moisture absorbtion
the smaller the dwelling
with trickle vents closed
the more moisture spreads around the dwelling
if the kitchen door is closed.

and the opposite for extract fans.

The effect of humidity controlled PSV is to significantly reduce the energy penalty of PSV systems at low moisture production rates and to shift the crossover point to lower moisture levels. Thus it is clear that in some conditions extract fans have the lower energy use and in other conditions PSV has the lower energy use. It should be made clear however, that this is a steady state analysis and based on a single cell ventilation model and it would be unwise to be too precise about the cross over points. However the conditions shown are all within the range which will occur in practice.

4. ENERGY USE IN PRACTICE

There is little systematic evidence of the relative energy use of PSV and extract fans in practice. What little evidence there is (references 7 and 8) suggests that they are broadly comparable but as can be seen from figures 1 - 3 the differences in terms of overall dwelling energy use are relatively small. One is generally talking about a gigajoule or so per annum compared with about 50 gigajoule/annum for total energy use for a 200m^3 house, even with modern UK insulation levels. It would take either very precise experimental work with a lot of detailed measurements or a very large field study to discriminate to this level. In practice in the UK, PSV has been used almost exclusively in public sector housing and in very small owner occupied dwellings such as starter homes. Thus the analysis given above ties in with practical experience.

5. CONCLUSIONS

There is a need for a more precise dynamic, multicell analysis but within the limitations of the analysis presented above, it has been shown that the issue of energy use of PSV and extract fans is a complex one affected by at least eight independent variables. Within the range of these variables which occur in practice, sometimes extract fans have the lower energy use and sometimes PSV. Typically PSV both humidity controlled and uncontrolled is better from an energy point of view in a relatively cold, airtight small dwelling with high moisture production. At the other extreme fans will be more energy efficient in the warm leaky large dwelling with low moisture production.

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fig1 house 200 cu m,7ac/hr at 50pa

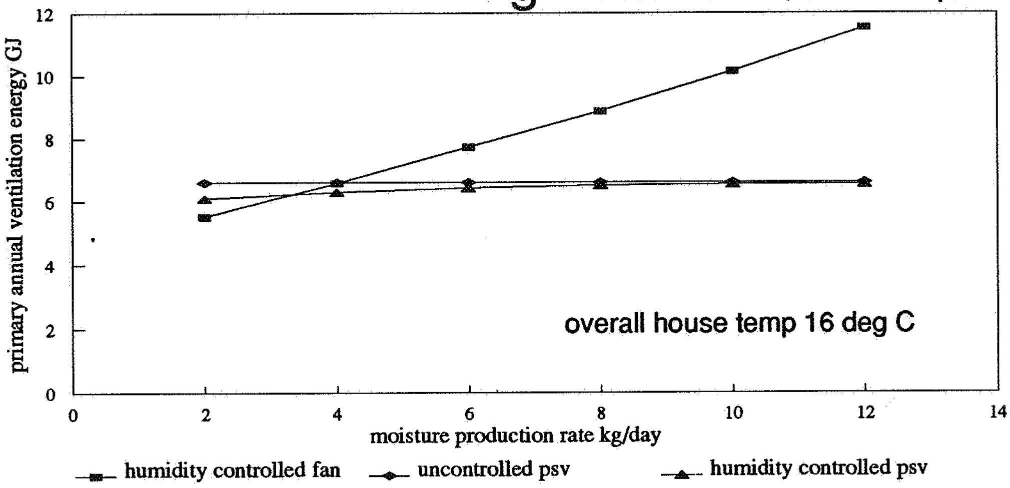


fig2 house 200 cu m,7ac/hr at 50pa

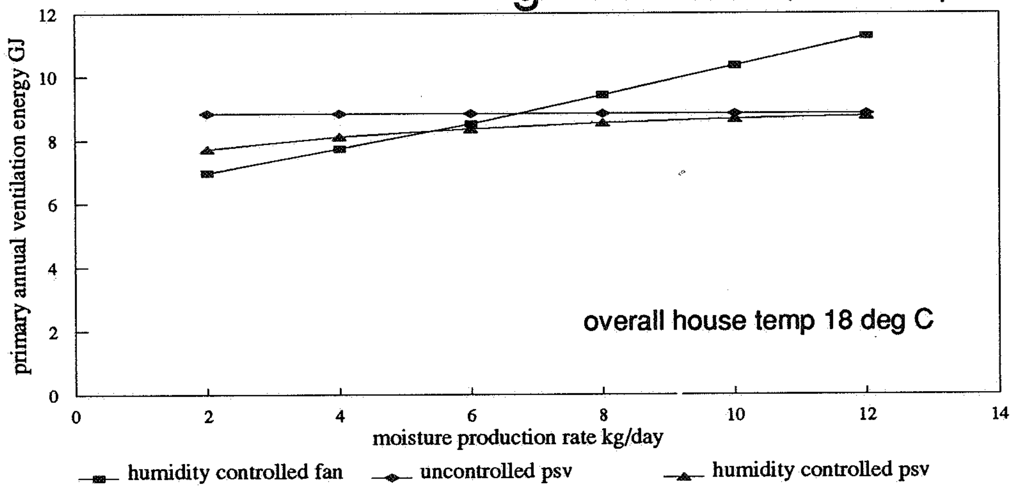
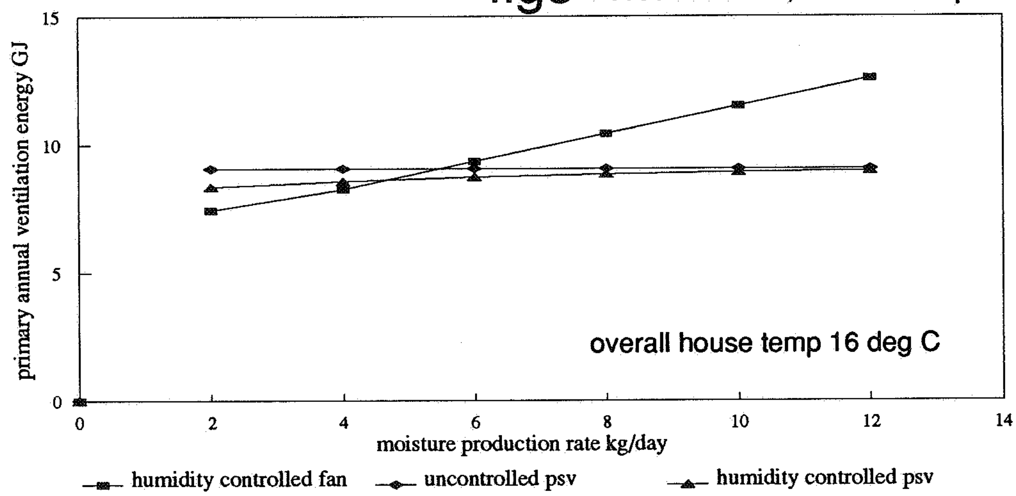


fig3 house 200 cu m,10ac/hr at 50pa



The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Volume Control of Fans to Reduce the Energy Demand of Ventilation Systems

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Synopsis

The fan and the ductnetwork ist designed for 100 % ventilation rate. Because the fan energy is the main important energy consumption in systems all over the year it is worthwile to control the systems correctly. By reducing the air volume rate the pressure drop in the ductnetwork drops nearly with the second power.

The energy demand at the fans is:

$$P = \frac{\dot{V} \cdot \Delta p}{\eta}$$

As the pressure drop ist nearly the volumerate to the second power, the energy consumption goes nearly with the third power of the volumerate. This means to find a very effective control strategy of the fans. There are four possibilities: 1. Damper control, 2. Bypass control, 3. Vortex damper control, 4. Speed control.

The comparison of the four strategies show clearly that the speed control can be the best of the fan design ist good on the optimum efficiency line in the $\Delta p / \dot{V}$-diagram follows about the equation:

$$\Delta p / \dot{V}^{1,9}$$

An other important point is the high efficiency at the electrical drive in part load conditions at low power.

1. Introduction

The main tasks in ventilation systems is the transport of air in order to be able to decrease the air contamination in the rooms. This needs transportation energy which can be calculated:

$$P = \frac{\dot{V} \cdot \Delta p}{\eta}$$

The air systems also have great differences in this value because of the high range of pressure drop.

2. System characteristics

The conservative plants have duct-velocities up to about 7 m/s and give the possibility to put the air-inlets direct to the ducts without any throttle.

To be able to reduce the size of the ducts, high velocity plants have been introduced about 20 years ago. The velocities in the ducts rise up to about 25 m/s.

In these systems between the main duct and the air inlet there must be a throttling unit to reduce the pressure. The main effect is the equilbration of the air distribution in large duct systems.

Caused by the high velocity the energy consumption to transport the air is higher than in conservative plants.

Regarding the above mentioned equation for calculating the energy consumption of the fan we have a simple proportionality between pressure drop Δp and power P, because the air flow rate $\dot{V}$ is constant.

In air ducts we almost always have turbulent flow, therefore the influence of diameter or reference diameter at constant air flow rate is given at smooth ducts by:

$$\Delta p \approx d^{-4,75}$$

and in extremly rough ducts:

$$\Delta p \approx d^{-5}$$

The partial energy amount for transport in the ducts is therfore:

$$P \approx d^{-4,75} or \;\; P \approx d^{-5}$$

if $\dot{V}$ = const. and η = const.

The pressure drop in the central units does not change.

3. Dimensionless Numbers for fans

the energetical behaviour of a fan can be shown by dimensionsless numbers:

$\phi_N = \frac{c_{ax}}{u_N}$ .....velocity ratio

$\psi_N = \frac{\Delta p}{\frac{\rho}{2} u_N^2}$ .....pressure ratio

$\lambda - \frac{\phi_N \cdot \psi_N}{\eta_i}$ .....power input coefficient

$\mathrm{Re}_u = \frac{u_N \cdot d_H}{\nu}$ .....Reynolds Number for tangential velocity

$\mathrm{Re}_c = \frac{c_{ax} \cdot d_H}{\nu}$ .....Reynolds Number for axial velocity

$with \;\; d_H = d_s - d_N$ .....hydraulic diameter

4. Part load characteristic

The load of a ventilation system is changing sometimes over a large range and therefore it is necessary to vary either the specific enthalpy difference between supply air and return air or the air flow rate.

Using the varying volume systems there is a decrease of the transportation energy depending on the air flow rate.

As we saw this part of energy amount is fairly large and therefore it is very interesting to discuss all possibilities from the point of view of the whole energy amount. Reducing the air flow rate the velocity in the ducts decreases proportionally. The pressure drop and the power decreases also but with a higher power if the efficiency η of the fan is constant:

Smooth tube: $\Delta p \approx w^{1,75}$ *and* $P \approx \dot{V}^{2,75}$

Very rough tube: $\Delta p \approx w^2$ *and* $P \approx \dot{V}^3$

In the middle range of practical use the tubes and the whole system will follow an equation of:

Therefore it is necessary to discuss the possibilities of the variation of the air flow rate with the fans.

5. Part flow characteristic of fans

The energy consumption of fans can be calculated from the equation:

$$P = \lambda \cdot A \cdot \frac{\rho}{2} \cdot u^3$$

with A as total flow area in m^2

ρ as air density in $\frac{kg}{m^3}$

u as periphal speed in $\frac{m}{s}$

λ as the power amount coefficient

P as as energy amount in W

With constant rpm P is proportional to λ. The air volume $\dot{V}$ moved by this fan is:

$$\dot{V} = A \cdot w_{ax}$$

with w_{ax} as the axial velocity in m/s.

To describe this air flow rate by a dimensionless number we state:

$$\varphi = \frac{w_{ax}}{u}$$

A diagram $\lambda = f(\varphi)$ shows therefore the power consumption as function of air flow rate.

Figure 1 shows this function for air control by air flow dampers. We see an increase of λ over a wide range of decreasing φ. A variation of air flow rate by this method of fan control therefore does not reduce the energy consumption for air transportation.

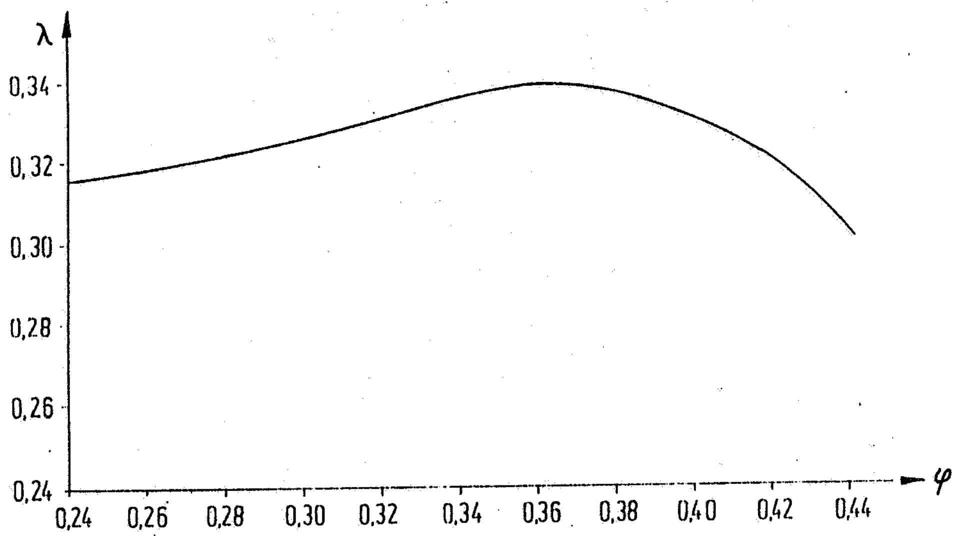

Figure 1: Control-Characteristic using Air-Dampers

Much better we find the control characteristic of the fans using spin dampers at the entrance to the fan (figure 2). We see a very high decrease of energy consumption by changing the angle of the blades.

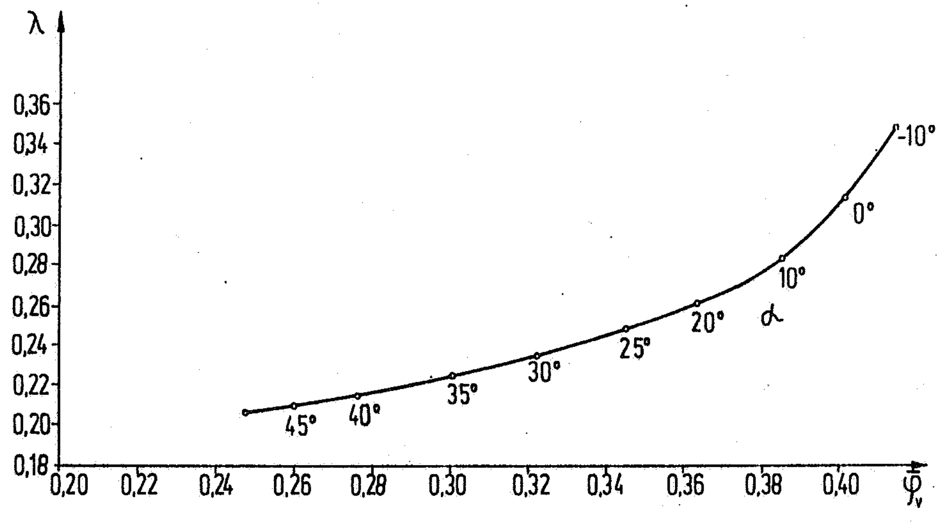

Figure 2: Control-Characteristic using Spin-Dampers

Very good characteristics can be shown by using a speed variation of the fan to control air volume. Figure 3 shows a characteristic diagram of a fan, where we can see that for instance the line of the highest efficiency η_i is a parabolic curve.

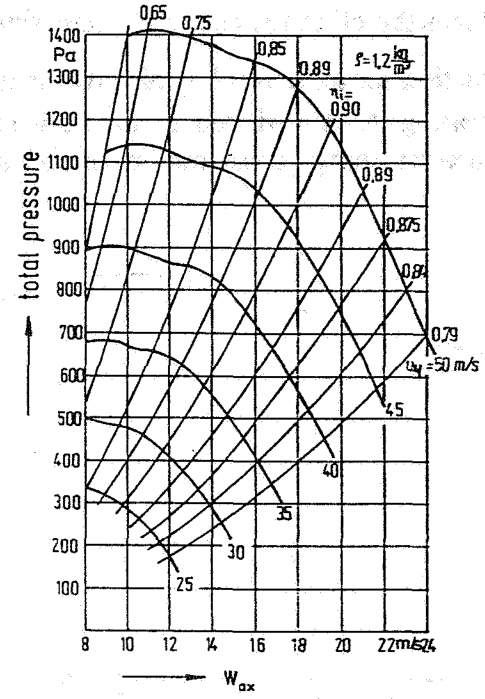

Figure 3: Characteristic of Fan

As we showed in chapter 4 the pressure drop depends nearly on $w^{1,9}$. This means ist would be the best to design the axial fans with a characteristic line for the optimal efficiency following the equation:

$$\Delta p \approx w^{1,9}$$

There it is possible to vary air flow rate by changing the draft speed of the fan nearly with constant efficiency.

to be able to decrease the energy consumption it is necessary therefore to install a fairly simple variation of shaft speed by electrical circuit.

Conclusion

The energy demand of the fan is very important for ventilation systems. The most effective way to vary the air volume rate is the control of shaft speed. Using a good fan design it is possible to remain always at the highest fan efficiency, which can really save energy.

The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

A Design Guide for Thermally Induced Ventilation

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Synopsis

A design guide for displacement ventilation (thermally induced ventilation) has been prepared. It is based on quasi stationary experiments carried out in the Sulzer Infra laboratory in Winterthur. The significant design parameters identified by factorial analysis are the air flow rate, the internal load, the convective part of the internal load and to a lesser extent the room height. Using a linearized polynom representation for the temperature increase near the floor as well as for the vertical temperature gradient in the occupied zone a design nomogram has been obtained. Within its range of application the design nomogram also applies for displacement ventilation systems combined with cooled ceilings. The design guide is published in german and french language and is one of the major outputs of the Swiss research program on «Energy Relevant Air Movements in Buildings (ERL)».

Background

Detailed knowledge on displacement ventilation has tremendously increased in the early nineties. International cooperation has encouraged the refinement of the technique of Computational Fluid Dynamics and numerous full scale experiments and field tests have produced data on room air flow. These activities mainly took place in the scientific world.

In the consulting branch we observe a strong market penetration of low velocity inlets for displacement ventilation. Manufacturers have produced video films to demonstrate room air movements and have made their air supply units more attractive for architects.

It is only very recently, that P.V. Nielsen (Displacement ventilation - theory and design, 1993) and H. Skistad (Displacement ventilation, 1994) have published comprehensive books in english language. Until then, the system designer only had the choice to either carefully study the limited literature or to directly discuss the design with manufacturers. In the latter case one has no tools to verify the design parameters proposed by the manufacturers and becomes very quickly product-dependent . It is in the view of giving better support to system designers, that the elaboration of a design guide has been started 1992 in Switzerland.

We here give a brief overview of the experimental procedure, show some aspects of the results obtained and finally demonstrate the nomogram developped for system design. For more details the design guide itself as well as the various scientific reports must be consulted.

In the following we will use the term «Thermally Induced Ventilation» to describe a ventilation system where the air is driven by heat sources (apparatus, persons) or sinks (cold windows). In the scandinavian literature such a system is often referred as displacement ventilation. We feel that the term displacement ventilation is to closely related to piston flow type systems and therefore is not the appropriate term to describe the physical processes which are driving the room air movement.

Planning of the Experiments

From our own practical experience in designing ventilation systems and from the experimental data available we identified a set of parameters likely to be relevant in the design of thermally induced ventilation systems (Table 1).

Design parameter	Influenced quantity
Room height	Temperature profile, range of applications
Air supply rate	Thermal comfort, air flow pattern, cooling capacity
Supply air temperature	Thermal comfort, air flow pattern, cooling capacity
Internal load (source strength, location of the source, convective part)	Air flow pattern, temperature profile
Radiative cooling	Cooling capacity, range of applications

Table 1: Relevant parameters in the design of thermally induced ventilation systems

The range of value of the design parameters to be chosen depends on the specific application. Here we limit ourselves to single cell office rooms and meeting rooms. The range of the above parameters for such rooms was therefore chosen as following (Table 2):

Design parameter	"low value"	"high value"
Specific air flow rate	10 $m^3/h/m^2$	20 $m^3/h/m^2$
Internal load	10 W/m^2	30 W/m^2
Convective part of internal load	20 %	80 %
Room height	2.4 m	3.6 m

Table 2: Range of value of design parameters investigated

The following comments have to be made: The supply air temperature was not considered as an independent parameter in the experimental set-up, since it is connected to the supply air velocity through the Archimedes number. It was kept at a constant temperature (18° C) for both summer and winter conditions. The "low value" for the specific air flow rate has been chosen relatively high compared to the swiss recommendations for mininum air flow rates in non-smoking areas of office buldings. The values for internal load apply to thermally induced ventilation without cooled ceilings.

The factorial design method was used to plan the number of trials (experiments) in the laboratory. This method has the advantage to significantly reduce the number of trials and identifies those variables (design parameters) which have the largest influence on a control quantity. Control quantities are e.g. the temperature gradient in the occupied zone or the room air temperature at floor level. They can be represented in the following form:

$$Y=b_0+b_1x_1+b_2x_2+b_3x_3+b_4x_4+....$$

were x_1, x_2, etc are the variables (design parameters). With four variables the number of trials is $2^4=16$. All variables were either set to the "low value" or to the "high value" of table 2. Three "Zero-trials" with intermediate values were added in order to test the linearity.

Experimental Set-up

All experiments were performed under quasi stationary conditions at the Sulzer Infra Laboratory in Winterthur. The measurements took place in a full scale furnished room of 7.25 x 7.25 m. A thermally induced ventilation system with wall mounted diffusers, through one entire sidelength, and two exhaust openings under the ceiling were installed. Air temperatures and air velocities were measured at three different positions, at two working places and free placed, in different heights. In addition surface temperatures were registered. The velocity measurements were carried out using a Dantec low velocity multiflow analyzer with temperature compensated sphere probes. The temperatures were measured with thermocouples connected with a HP precision Voltmeter.

Example Results

The measured air temperature distribution (Figure 1) corresponds to the pattern known from measurements in other laboratories. At higher (30 W/m^2) thermal load the room air temperature is shifted towards higher temperatures and its distribution is much more inhomogeneous than with a relatively low thermal load (10 W/m^2). In addition a larger convective part of the thermal load increases significantly the temperature gradient in the occupied zone.

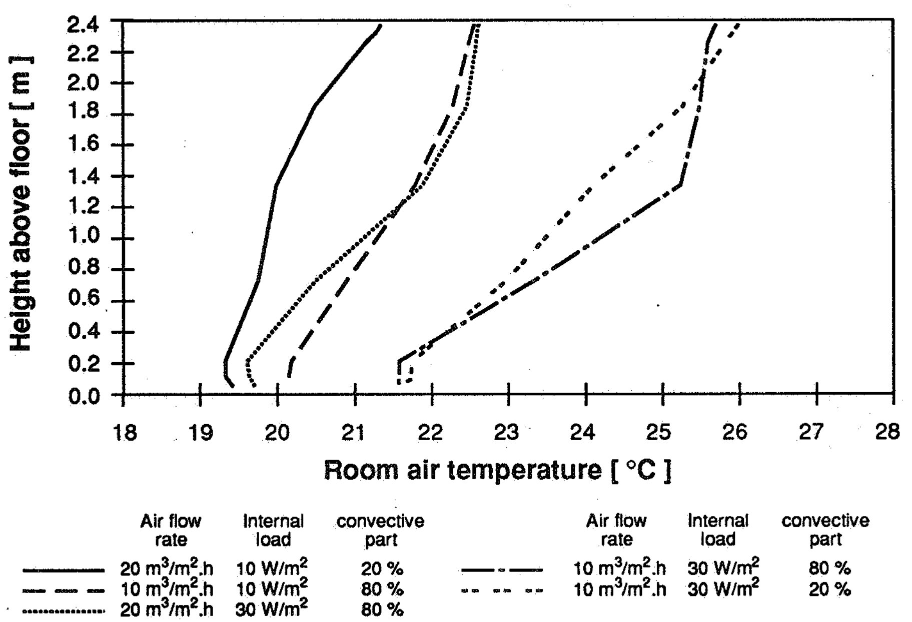

Figure 1: Air temperature distribution for thermally induced ventilation

The air velocity profile has also been measured but is not shown here. In the occupied zone, outside the rising thermal air flows, typical values for air velocities are 0.03 to 0.07 m/s, turbulence indices range from 5 to 50%. The corresponding PPD (pecentage of persons dissatisfied) with 5% is at it's minimum. The highest velocities are found between 5 and 10 centimeters above the floor.

Model assumptions

In many cases the temperature distribution in a room is nearly homogeneous and the temperature measured close to the floor appears to be approximately half way between the supply air temperature and the extract air temperature. Based on these findings the 50/50 model was stipulated several years ago in scandinavian countries (see figure 2). This model assumes that 50% of the temperature difference in the room is taken up in the floor. This applies to rooms of conventional height (2.4m to 3.6m) and to"normal" heat loads.

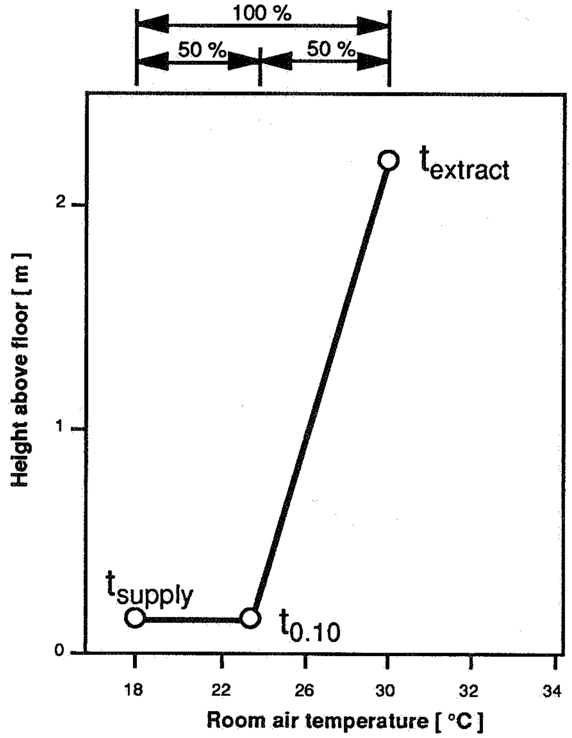

Figure 2: The 50/50 model

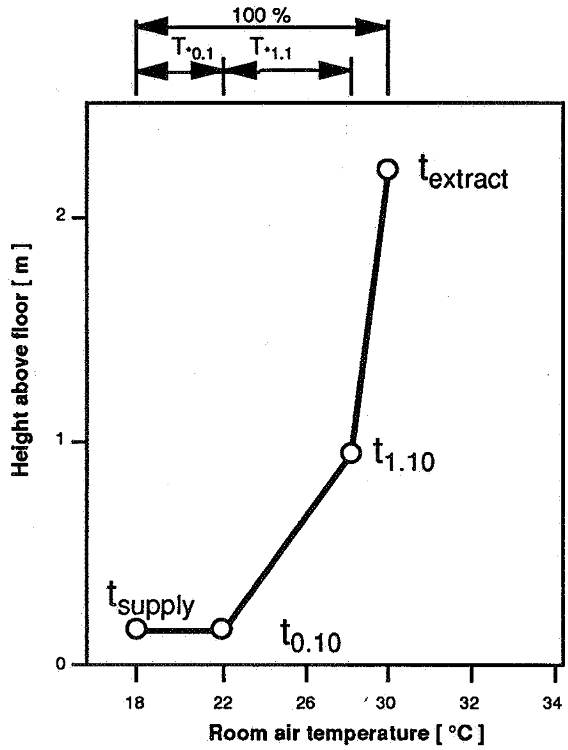

Figure 3: The ERL model

Our own observations show, that a homogeneous distribution is only observed under conditions of very low heat load (10W/m2) and for supply air flow rates in the range of 10 to 20 m3/h/m2 (see figure 1). When the total heat load reaches higher values (which is normally the case) and/or when solar radiation hits the floor, the vertical temperature distribution is no more homogenous. It shows a clear bend at a height between 0.6m and 1.4m depending on the heat load. This occurs because the convection currents become larger than the supply air flow and a mixing of the upper room air is created. As a consequence the measured temperature gradient in the occupied zone is often observed to be larger than the one predicted by the simple 50/50 model. Therefore we looked for a model which better fits the measured data (see figure 3). Since in Switzerland the comfort conditions must be fulfilled in a zone reaching from 0.1 to 1.1m above floor, these marks were taken to calculate the temperature rise near the floor and the vertical temperature gradient in the occupied zone. Note that the difference in height is exactly 1m, so that the temperature rise equals the vertical gradient in the occupied zone.

Data reduction

The analysis of the measured data according to the factorial design method gave the following equations for the dimensionless control quantities:

Temperature rise near floor: $T^*_{0.1}=\{45.5-2.2x_1-2.4x_2-2.1x_4\}\cdot 10^{-2}$ [eqn.1]

Temperature gradient in occupied zone: $T^*_{1.1}=\{57.3-6.6x_1+10.5x_3\}\cdot 10^{-2}$ [eqn.2]

In the dimensionless form the variables $x_1,...x_4$ take either the value 1 or -1 (see appendix for the case of the temperature gradient). For numerical interpretation the dimensionless variables can be transformed into the design parameters using the following relations:

$$x_1=0.2v-3; \quad x_2=0.1q-2; \quad x_3=0.33q_c-1.67 \text{ and } x_4=1.67r_h-5$$

where	v	=specific air flow rate	[$m^3/h/m^2$]
	q	=total heat load	[W/m^2]
	q_c	=convective part of load	[%]
	r_h	=room height	[m]

Graphical representation of the temperature rise near the floor and the temperature gradient can be obtainied from equation 1 and equation 2 respectively using the above transformations. Finally, both representations can be combined in a nomogram. The following simplification were made:

- The relation between temperature gradient in the occupied zone (1.1m-0.1m) and the load has been linearized, and
- the nomogram applies for all room heights between 2.4m and 3.6m.

Design procedure

When designing a ventilation system, the designer must answer the questions "What is the purpose of the ventilation system ?" and "Is the priority to remove heat or to provide good air quality ?". In many cases there will be no clear answer to the second question, and consequently no "best" system. It is however important to have in mind, that displacement ventilation is a system for the best possible air quality, but without cooled ceilings is not appropriate if one deals with a large heat load.

The various steps used in the design procedure are as usual. The design nomogram (see figure 4) is a useful tool to determine the supply air flow rate and to check the temperature gradient in the occupied zone as well as the temperature rise near the floor in one single step. It also offers the possibility to vary the convective part of the heat load.

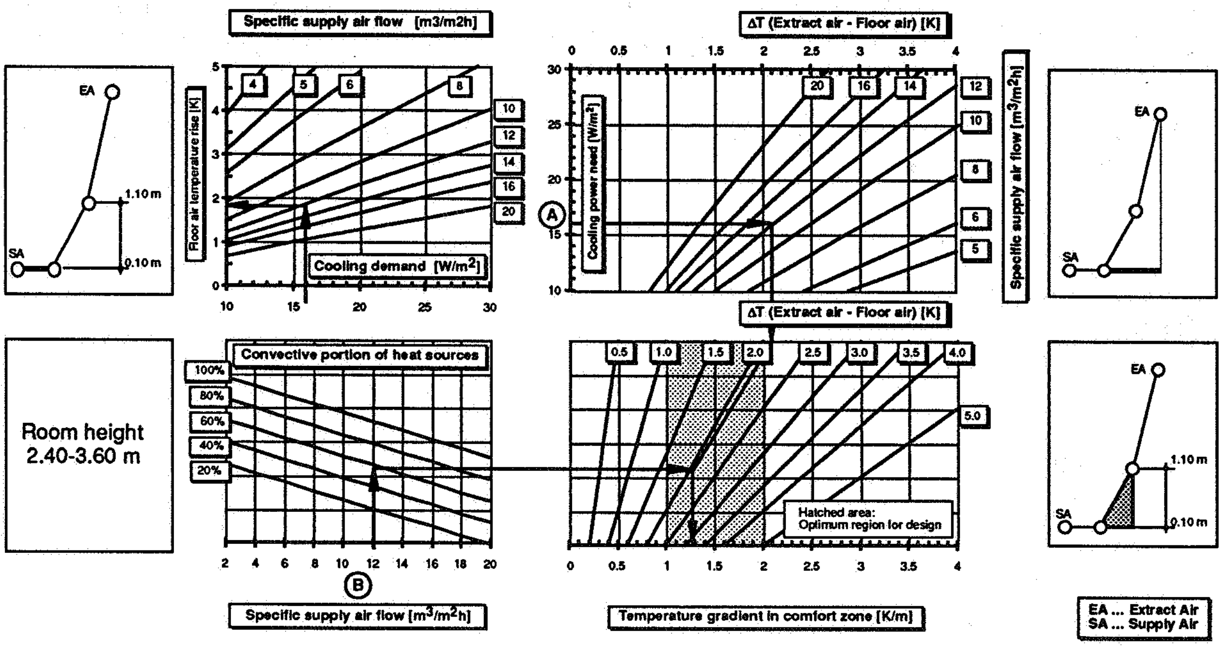

Figure 4: Design nomogram for thermally induced ventilation (displacement ventilation)

In order to illustrate the use of the design nomogram, an example calculation for a single cell office is shown in table 3. The results are compatible with those obtained from other design procedures (see e.g. H. Skistad).

Dimensions and data			*Determine air flow rate*	
Single cell office	Floor area	25 m2		
	Room height	2.8 m	Cooling load*	$q = 16\ W/m^2$
Internal load	2 Persons	240 W	Convective part of internal load	$q_c = 57\%$
	2 PC	200 W		
	1 Laser-Printer	100 W	Specific air flow rate	$v = 12\ m^3/m^2/h$
	Lighting	250 W		
			Tempcrure gradient	$T_{1.1} = 1.3$ K/m
External load		0 W		
			Temperature rise near floor	$T_{0.1} = 1.8$ K
	TOTAL	790 W		

* The cooling load is smaller than the total of internal and external load due to heat storage in the building fabric and non-simultaneity of partial loads

Table 3: Example calculation for a single cell office room

Acknowledgements

The authors wish to thank all scientist in research and industry who have participated in the task. They are to numerous to be listed here.

Our special thanks go to our scandinavian friends Professor Eystein Rodahl and Hans Martin Mathisen from Trondheim as well as Professor Peter V. Nielsen from Aalborg who encouraged the work and gave us advice in setting up the experiments.

The work has been supported by the Swiss Federal Office of Energy (BEW) and by the National Energy Foundation (NEFF).

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Appendix

Auswertung 2^4 Versuche

Versuch	T* 1.10-0.10 [-]	0	1	2	12	3	13	23	123	4	14	24	124	34	134	234	1234
A	0.536	1	-1	-1	1	-1	1	1	-1	-1	1	1	-1	1	-1	-1	1
B	0.549	1	1	-1	-1	-1	-1	1	1	-1	-1	1	1	1	1	-1	-1
C	0.576	1	-1	1	-1	-1	1	-1	1	-1	1	-1	1	1	-1	1	-1
D	0.385	1	1	1	1	-1	-1	-1	-1	-1	-1	-1	-1	1	1	1	1
E	0.689	1	-1	-1	1	1	-1	-1	1	-1	1	1	-1	-1	1	1	-1
F	0.537	1	1	-1	-1	1	1	-1	-1	-1	-1	1	1	-1	-1	1	1
G	0.816	1	-1	1	-1	1	-1	1	-1	-1	1	-1	1	-1	1	-1	1
H	0.731	1	1	1	1	1	1	1	1	-1	-1	-1	-1	-1	-1	-1	-1
I	0.462	1	-1	-1	1	-1	1	1	-1	1	-1	-1	1	-1	1	1	-1
K	0.327	1	1	-1	-1	-1	-1	1	1	1	1	-1	-1	-1	-1	1	1
L	0.528	1	-1	1	-1	-1	1	-1	1	1	-1	1	-1	-1	1	-1	1
M	0.363	1	1	1	1	-1	-1	-1	-1	1	1	1	1	-1	-1	-1	-1
N	0.771	1	-1	-1	1	1	-1	-1	1	1	-1	-1	1	1	-1	-1	1
O	0.514	1	1	-1	-1	1	1	-1	-1	1	1	-1	-1	1	1	-1	-1
P	0.740	1	-1	1	-1	1	-1	1	-1	1	-1	1	-1	1	-1	1	-1
Q	0.630	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
Wirkung		b0	b1	b2	b12	b3	b13	b23	b123	b4	b14	b24	b124	b34	b134	b234	b1234
Summe		9.1748	-1.063	0.4037	0.0007	1.6825	-0.148	0.4091	0.4256	-0.464	-0.231	0.011	0.2744	0.2267	-0.028	-0.482	-0.114
Mittel		$\boxed{0.573}$	$\boxed{-0.066}$	0.025	0.000	$\boxed{0.105}$	-0.009	0.026	0.027	-0.029	-0.014	0.001	0.017	0.014	-0.002	-0.030	-0.007

Table A: Factorial analysis for the temperature gradient

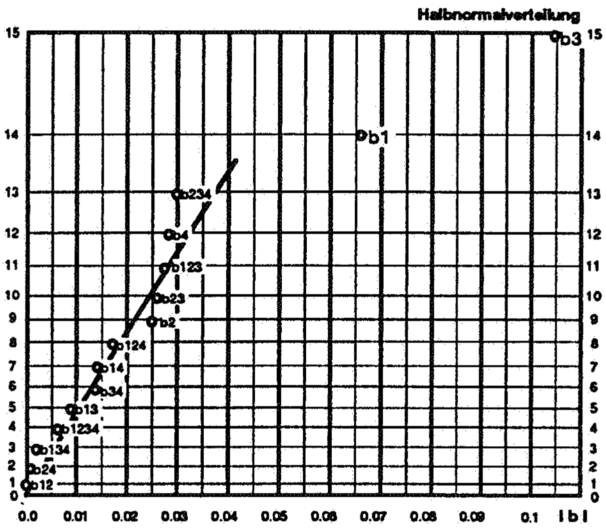

Figure A: Half-normal distribution showing significant coefficients for the temperature gradient

The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Air Movement in a Re-clad Medium Rise Building and its Effect on Energy Usage

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ABSTRACT

This paper presents the results of a monitoring programme on a medium sized educational building which has had the external walls re-clad. The objective behind the re-cladding was to improve the durability of the building and to improve the thermal performance.

The objectives of this work were to establish the viability of the calculation techniques used to simulate the ventilation, thermal and moisture performance of the re-cladding system.

The results have shown that there is a good agreement between the methods currently being used and the actual performance.

1. BACKGROUND

As part of the upgrading policy of the University of Sheffield the external facade of a nine story office building has been re-clad with a ventilated cavity wall structure which includes thermal insulation and weep hole devices to remove any moisture build-up.
As this type of re-cladding system is not common in such large buildings it was felt necessary to investigate its performance with a view to establishing the viability of the current techniques for estimating the performance of structures. In order to do so an extensive programme of both physical and computer modelling was carried out.

1.1 The Wall Construction

The external wall of the building originally consisted of exposed concrete floor slabs with a GRP infill panels. In order to minimise the cold bridging effect of such a construction the re-cladding consisted of attaching to each slab an angled beam onto which was built a brick outer layer. Because of the exposure of the building it was felt that weep holes should be left in the wall to allow any moisture which may build up to escape. However these holes will also allow air to enter and therefore there may be a reduction in the insulating properties of the structure. Figure 1 shows a cross section of the re-clad wall.

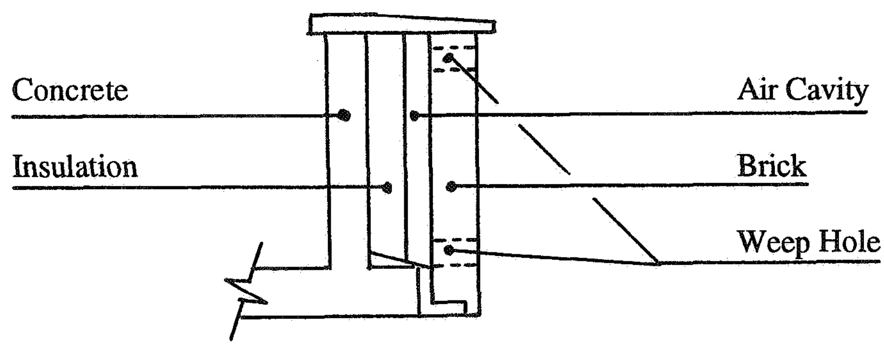

Figure 1: Construction of External Wall

2. LABORATORY MEASUREMENTS

2.1 Ventilation performance of the Wall Construction

This series of experiments concentrated on establishing the air flows within the cavity between the outer brickwork and the thermal insulation. Most of the results of this programme were presented at the an earlier AIVC Conference[1] and indicated that it was possible to measure the air flow into an air cavity 1.2 metres high and 46 meters long. The measurements presented were then used in the COMIS simulation programme to carry out a study into the cavity performance under various wind conditions. The wind conditions simulated were based on measurements taken at the local Meteorological Station. In order to carry out the simulations it was necessary to obtain reliable values of the wind pressure coefficients acting on the building.

2.1.1 Wind Tunnel Testing

The building including the immediate surroundings were simulated in a boundary layer wind tunnel to establish the Cp values acting on the facades of the building. A full report on this work is available from the School of Architectural Studies[2].
Table 1 shows for the four main facades the difference in Cp values obtained by simulation with and without surrounding buildings.

Table 1: Cp Values for the Office Simulated		
Orientation	Isolated	Within Surroundings
North	-0.117	-0.031
North East	-0.203	-0.137
East	-0.142	-0.138
South East	-0.193	-0.129
South	0.14	0.02
South West	0.196	0.161
West	0.199	0.144
North West	0.065	0.152

2.2 Insulation material and humidity

2.2.1 Introduction

Previous work by Burch et al. (1989)[3] and McLean et al. (1990) [4] suggested that the vapour permeability of building materials is not affected significantly by the prevailing ambient temperature, and for effectively non-hygroscopic materials, such as plasterboard and polystyrene, the permeability is essentially independent of humidity. For calculations of these materials a constant value can be assumed, causing little error. However, for hygroscopic materials, such as wood, plywood and brick the permeability depends substantially on the ambient humidity and may vary significantly. To minimise errors in calculations for these materials the vapour permeability should be taken into account.

2.2.2 Moisture absorption

Table 2 shows the results of the laboratory investigations. Very little vapour was absorbed by the insulation material.

Table 2: Various test conditions during the experiment

day, time		box		laboratory	
-	weight	temperature	humidity	temperature	humidity
-	[g]	[°C]	[%]	[°C]	[%]
dry sample	35.464	-	-	-	-
1, 10:40	35.555	22.1	76.9	22.3	27.1
1, 11:40	35.555	21.8	92.8	21.6	28.5
2, 11:20	35.568	22.0	94.5	22.1	31.4
3, 13:30	35.561	19.9	98.5	21.6	39.3
4, 11:00	35.563	19.6	99.3	20.4	49.4

The test has shown that the insulation material is non-hygroscopic, a negligibly small amount of moisture was accumulated in the insulation material.

2.2 3 Tests for Hygroscopy

A specimen of 0.2 x 0.1 m was submerged in a water bath. After 44 hours the specimen was cut up into four parts and no capillary diffusion could be detected.

3. FIELD MEASUREMENTS

A full monitoring programme of measurements in one office in the building was set up and continued for one year. Figures 2 and 3 indicate the type of results obtained.

The long-term data collected was used to establish the following:

- the time constant of the wall structure
- the dynamic u-value of the ventilated wall structure
- the influence of the wind on the temperature stratification in the cavity
- the ventilation simulation with COMIS, and
- the thermal simulation with SPARK.

3.1 Time constant of the wall structure

With the measured weather data, the temperatures in the wall and the room temperature, it was possible to determine the time constant of the wall structure. The average time lag of this wall structure was calculated to be 6 hours and 15 minutes. The analysed data is presented in Table 3. The value for the time constant for this type of structure taken from Table A3.17 in the CIBSE Guide (A3) ranges between 8 and 9 hours.

Table 3: Time lag of the wall structure during 23.3 - 26.3.1993

t_{out} max °C	time of day hrs:min	t_{in} max °C	time of day hrs:min	time lag hrs:min
9.4	10:10	19.7	16:20	6:10
10.6	10:44	20.3	18:15	7:31
15.3	11:38	19.2	17:38	6:00
13.8	10:31	21.6	15:51	5:20
			average time lag	6:15

The problems encountered in the determination of the time constant of this structure using the measured data were the following:

1. Heat sources in the room,
2. The unknown occupancy of the room,
3. Workmanship

3.2 The dynamic u-value of a ventilated cavity wall

To estimate the dynamic u-value the model developed by Anderson(5) has been used. Solving Anderson's model for a given time period results in the average heat transmission rate incorporating the dynamic behaviour of the wall being estimated. Tests carried out by Anderson et al. (1985) show that the u-value can be estimated within an accuracy of 10 % provided the following conditions are met,

(a) the indoor outdoor temperature difference is at least 20 K,
(b) the daily outdoor temperature swing (high to low) is not larger than half the average indoor outdoor temperature difference,
(c) the average indoor and average outdoor temperatures do not vary significantly over the course of the test, and

The results of the calculations for several months are presented in Table 4 along with the calculated values from recognised design manuals.

Table 4: Comparison of the calculated and standard u-values			
	measured (Anderson) $W/m^2 K$	calculated u-value $W/m^2 K$	CIBSE / measured %
February	0.46	--	0
March	0.39	--	- 15.2
April	0.48	--	+ 4.3
May	0.43	--	- 6.5
June	0.54	--	+ 17.4
average	**0.46**		
CIBSE Guide	--	0.43 - **0.46** - 0.49	0
DIN 4701	--	0.494	
ASHRAE	--	0.51 - 0.52	

The results of the integrated u-value calculations show, with the exception of the June and March u-values, good agreement with the u-values calculated using the various standards. The difference between the calculated u-value using the measured data and the averaged CIBSE Guide value is also stated.

3.3 Effect of Wind Speed on Cavity Air Temperature.

A hypothesis that high wind speeds would destroy air stratification in the ventilated cavity was investigated. Although initial results indicated that there was some evidence for this a detailed study of the cavity air temperatures, wind speed/ direction, solar radiation and outside air temperature did not substantiate this hypothesis.

3.4 Analysis of the COMIS simulation results

Considering that the computer simulations were carried out using constant "set" temperatures for the cavity air, the outside air and the crack temperature, as well as the wind direction and wind speed the results from the simulation with the COMIS programme and the measurements on the cavity structure in the Building compare reasonably well. Table 5 summarises the results of the simulations and measurements carried out.

Table 5: Simulated and measured results				
test	wind direction flow direction	wind velocity m/s	COMIS [m^3/s]	measured [m^3/s]
1	340 north - south	2.5 measured	0.001645	0.00165
2	200 - 210 south - north	4.1 measured	0.00262	0.0029
3	280 north - south	5.6 measured	0.00473	0.0048
4	240 south - north	5.6 - 12.3 MET Office	0.00532 - 0.01167	0.00771
5	270 south - north	5.7 - 12.4 MET Office	0.00127 - 0.00379	0.00145
6	260 south - north	6.2 - 11.8 MET Office	0.00165 - 0.00351	0.00233

3.4 Analysis of the SPARK results

For the SPARK simulation five chosen data sets of representative weather were considered giving a large number of results to be analysed. The approach adopted was to compare the heat flux through the inside of the wall, the insulation temperature and the measured temperature between the insulation material and the concrete. Figures 4 and 5 show typical output from the analysis.
The comparison of the simulated and the measured data for the thermal simulation with SPARK showed good agreement for the insulation temperature. This result could not be produced for the simulation of the inside wall heat flux. Although the correlation between the measured and the simulated heat flux can be described as satisfactory the absolute values for peak heat flow through the wall, especially during high solar radiation, is very poor. The overall conclusion drawn for the SPARK simulation is that the program as it stands cannot handle the inside heat flux. The main limitation of the SPARK program was that it calculated the one-dimensional heat flow through the opaque part of the wall only, disregarding the complex three-dimensional heat flow processes taking place not only through the opaque part of the wall, but also through the window, which was not simulated.

4. CONCLUSIONS

The thesis from which this paper has been written was concerned with experimental and theoretical studies of the thermal and ventilation performance of a retrofitted ventilated cavity wall.

The main objectives were:

i) **Moisture Performance**
- To test if the insulation would absorb moisture.

ii) **Thermal performance**
- To determine how the wall structure performs, thermally, and to derive the dynamic u-value of the structure.

iii) **Ventilation performance**
- To test if the ventilation of the cavity works.
- To see if the air flow in the cavity can be measured.

iv) **Computer modelling**
- To compare the measured and the modelled values for the thermal performance of the wall structure, in order to assess the wall performance in practice.

i) Moisture performance

Concluding it can be stated that the experiments carried out to determine the effect of moisture on the insulation material in the cavity have shown that the effects were insignificant

ii) Thermal performance

By setting up a long-term monitoring system collecting temperature data throughout the wall structure, measuring the inside heat flux and collecting relevant weather data, it was possible to calculate the dynamic u-value of the wall. Data for five test periods, chosen for their distinctive features, were analysed. The calculated averaged dynamic u-value agreed well with the steady state calculation according to three design guidelines, CIBSE A3, DIN 4701 and ASHRAE, which do not take account of weep holes. The agreement of the averaged dynamic u-value for the five test periods with the CIBSE A3 Guide was within ± 7%.

ii) Ventilation performance

The results of the measurements indicate that horizontal air flow in the cavity occurs, and is mainly dependent on the pressure forces around the building due to the wind. For high wind speed, with wind direction being windward for the cavity under investigation, flows of up to 0.0071 m^3/s were measured. For a lower wind speed of up to 2.5 m/s the flow was determined to be 0.00165 m^3/s.

iii) Computer modelling

a) COMIS

For this part of the investigation two approaches were adopted - the tests carried out under laboratory conditions, and the in-situ measurements. The laboratory test were mainly carried out to test and validate the tracer gas technique developed to measure the in-situ air flow in the cavity. It was found that the uncertainty involved in the technique was within ±7%.

The overall conclusion for the ventilation simulation using the COMIS program package was that the results agreed very well with the measured data.

b) SPARK

The comparison of the SPARK simulation results and the measured performance of the wall achieved unsatisfactory agreement. The correlation between the two sets of data ranges from 0.599 to 0.933. However, the comparison of the simulated and the measured data for the insulation temperature showed good agreement. Although the correlation between the measured and the simulated heat flux on the inside wall can be described as satisfactory the absolute values for peak heat flow through the wall, especially during high solar radiation, is very poor.

An explanation for sometimes low correlation value lies in the different situations measured and modelled. The main limitation for the modelling was that the in-situ measurements take the complex three-dimensional heat flow into account, whereas the simulation was carried out for one-dimensional heat flow only.

From the results for the thermal simulation with SPARK it is concluded that the program does not model the situation very well.

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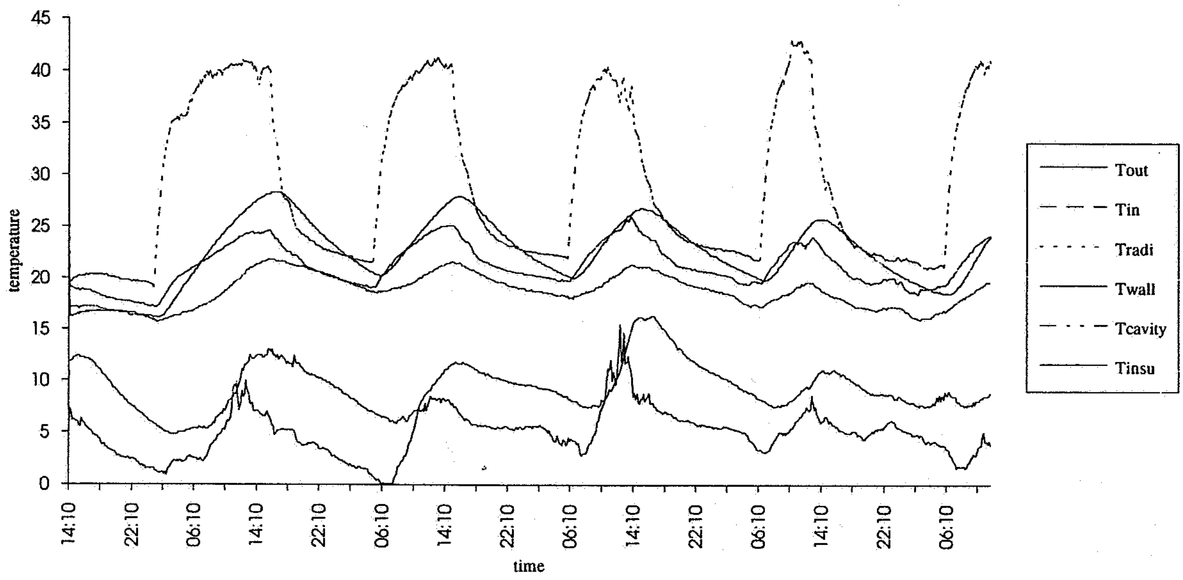

Figure 2: Measured Inside Wall Heat Flux for February

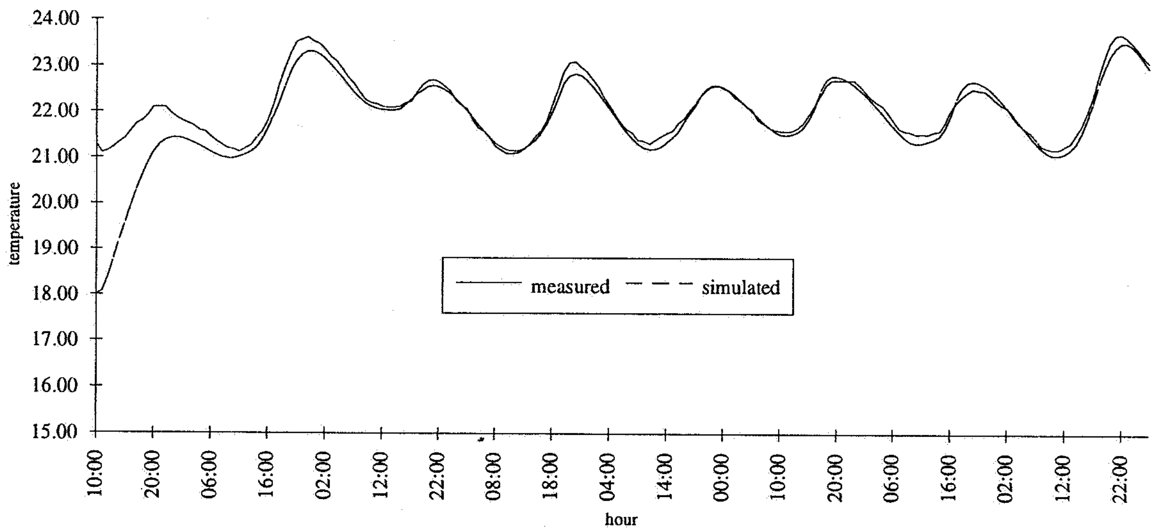

Figure 3: Comparison of the Measured and Simulated Insulation Temperature in June

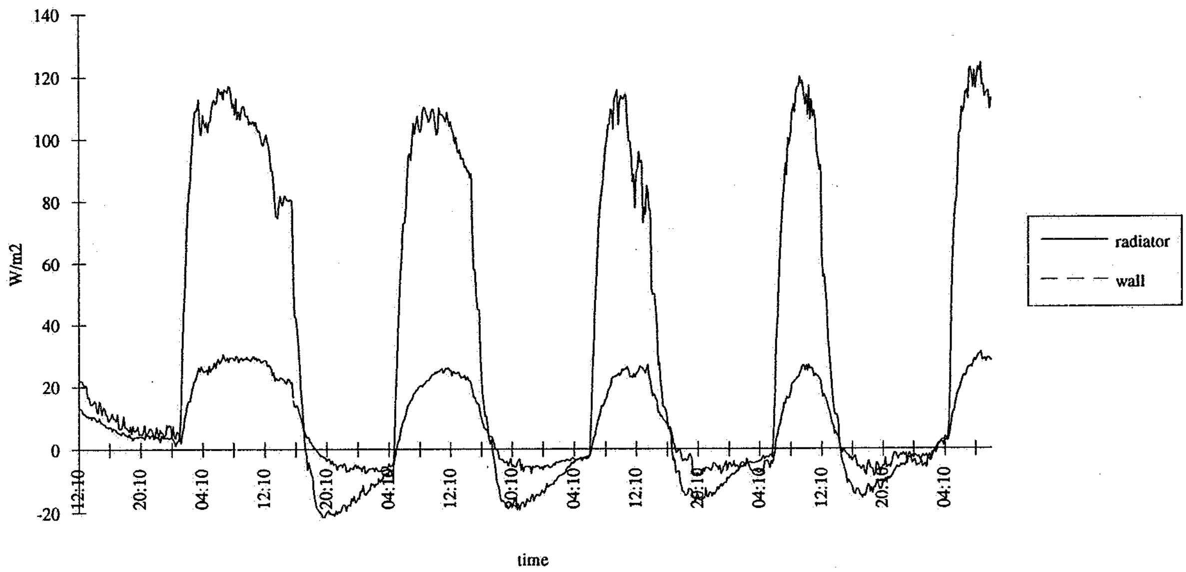

Figure 4: Measured Inside Wall Heat flux for February

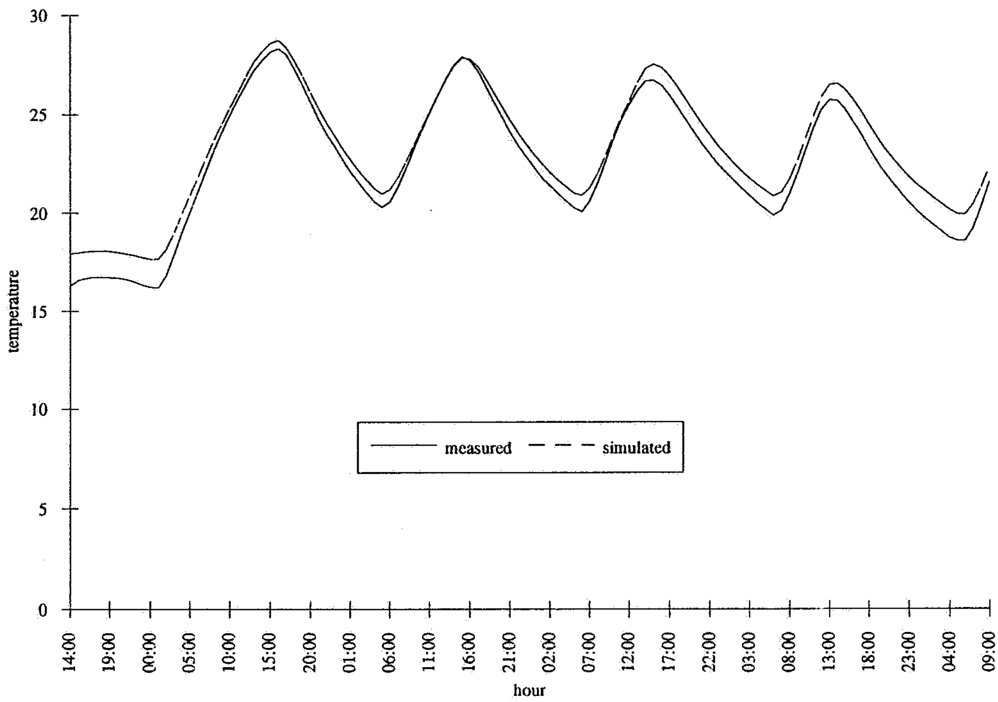

Figure 5: Comparison of Measured and Simulated Insulation Temperature in February

The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

The Performance of Dynamic Insulation in Two Residential Buildings

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ABSTRACT FOR THE AIVC CONFERENCE 1994

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THE PERFORMANCE OF DYNAMIC INSULATION IN TWO RESIDENTIAL BUILDINGS

In order to reduce the heat loss from buildings it is common to increase the thickness of insulation in the building envelope. The consequence of this action is more expensive buildings. Building regulations in countries with cold climate require U-values for the envelope which results in thicker and therefore often stronger constructions than needed for structural capacity.

Another strategy to save energy has been to reduce the ventilation rates in buildings. The consequence of this has been a lot of complaints from the users, and this action has certainly contributed to the diagnosis "sick building syndrome".

Dynamic insulation is an alternative to these actions. Dynamic insulation means a construction where the air is being forced through the insulation, usually from the colder outside air into the heated building. The theoretical U-value can be reduced to zero. In addition the incoming air is preheated.

The Norwegian Building Research Institute has been engaged to evaluate 12 row houses. with dynamic insulation used in the roof, which has been built in the Oslo area. Two of the houses are monitored with temperature sensors, pressure transducers and wind direction and velocity meter. In addition tracer gas measurements inside the houses and on the attic are carried out. Energy consumption is registered every week. The paper presents the main results from these measurements and describes how pressure differences, wind and temperature differences affect the performance of the dynamic insulation. The measured temperature profile through the dynamic insulation is compared with calculations.

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Ventilation and Energy Flow Through Large Vertical Openings in Buildings

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SYNOPSIS

After a short description of the physical phenomena involved, unified expressions are worked out describing net airflow and net heat flow through large vertical openings between stratified zones. These formulae are based on those of Cockroft for bidirectional flow, but are more general in the sense that they apply to situations of unidirectional flow as well. The expressions are compatible with a pressure network description for multizone modelling of airflow in buildings. The technique has been incorporated in the flows solver of the ESP-r building and plant energy simulation environment.
The relative importance of the governing variables (pressure difference, temperature difference and vertical air temperature gradients) is demonstrated by parametric analysis of energy performance in a typical building context and by comparison with experimental data in the literature. It is concluded that vertical air temperature gradients have a major influence on the heat transferred through large openings in buildings and should be included in building energy simulation models. Finally, it is discussed how the air temperature gradient, an input parameter which depends strongly on the heating and cooling mode, could be predicted.

Symbols

a_i	temperature profile coefficient for zone i (K)	z	height coordinate (m)
b_i	temperature gradient in zone i (K/m)	z_n	height of neutral level (m)
C_d	discharge coefficient (–)	z_0	height of reference level (m)
c_p	specific heat of air (J/kgK)	z_b	height of bottom of aperture (m)
g	acceleration of gravity (m/s^2)	Φ_{ij}	heat flow from zone i to zone j (W)
h	aperture height (m)	ρ	air density (kg/m^3)
M	molecular mass of air $(kg/kgmole)$	ξ	integration variable (m)
$\dot{m}_{ij}$	air mass flow from zone i to zone j (kg/s)	α	$\equiv (z_0 - z_b)/h$ (–)
P	pressure (Pa)	C_a	$\equiv \Delta P(z_b + h)$ (Pa)
$\dot{q}_{ij}$	air volume flow from zone i to zone j (m^3/s)	C_b	$\equiv \Delta P(z_b)$ (Pa)
P_{ref}	reference pressure (Pa)	C_t	$\equiv hgK\left(\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)}\right) = C_a - C_b$ (Pa)
R	universal gas constant $(J/kgmole\,K)$	K	$\equiv P_{ref}M/R$ $(Pa\,kgK/J)$
T	temperature (K)	Z_a	$\equiv \frac{2\sqrt{2}}{3}\frac{C_d hW}{C_t} C_a^{3/2}$ $(m^2Pa^{1/2})$
u	horizontal air velocity (m/s)	Z_b	$\equiv \frac{2\sqrt{2}}{3}\frac{C_d hW}{C_t} C_b^{3/2}$ $(m^2Pa^{1/2})$
W	aperture width (m)		

1 INTRODUCTION

Airflow through doorways, windows and other large openings are important paths via which air (including moisture and pollutants) and thermal energy are transferred from one zone of a building to another. In case of large openings, the airflow at the top usually differs from the flow at the bottom of the opening. Under certain conditions this may even result in bidirectional flow through the opening.
In recent times there has been an increased interest in modelling airflow through large openings in buildings (eg Allard et al. 1992). The current publication seeks to be a basic contribution in this area by presenting and demonstrating a general approach for predicting airflow and heat flow through large vertical openings between stratified zones.

2.1 Net Heat Flow when Zero Volume Flow

For the mass and heat transfer through large vertical openings, Balcomb et al. (1984) and others like White et al. (1985) and Boardman et al. (1989) used the so-called *isothermal zone* Bernoulli model.

According to Bernoulli, the maximum velocity $u(z)$ in a large vertical opening between two zones resulting from a static pressure difference (thereby excluding any frictional losses) is given by:

$$u(z) = \sqrt{\frac{2\Delta P}{\rho}} = \sqrt{\frac{2\Delta\rho}{\rho} g(z-z_n)} = \sqrt{\frac{2g}{T}\Delta T(z-z_n)} \quad (m/s) \qquad [1]$$

where z_n indicates the height of the neutral level (ie the level at which the pressure difference $\Delta P \equiv P_1 - P_2 = 0\ Pa$), $\Delta\rho \equiv \rho_1 - \rho_2$, and ΔT is the temperature difference between zone 1 and zone 2 ie $\Delta T \equiv T_2 - T_1$.

In this expression, it is implicitly assumed that ΔT is independent of the height coordinate z, ie that temperature gradients are equal and not too large. When the top-to-bottom temperature difference over the opening is small compared to the absolute temperature, this approximation is highly accurate.

The heat flow Φ_{21} from the warmer zone (2) to the colder zone (1) is carried by air flowing from 2 to 1 *above* the *neutral level.* The heat flow Φ_{12} from the colder zone (1) to the warmer zone (2) takes place *below* the *neutral level.* These contributions are given by:

$$\Phi_{21} = c_p C_d W \int_{z_n}^{z_b+h} \rho_2(z) u(z) T_2(z) dz \quad (W) \qquad [2a]$$

$$\Phi_{12} = c_p C_d W \int_{z_b}^{z_n} \rho_1(z) u(z) T_1(z) dz \quad (W) \qquad [2b]$$

Balcomb's expression for the *net heat flow* through the aperture is obtained by inserting the expression for $u(z)$ into the expressions for Φ_{21} and Φ_{12}, thereby assuming that the temperature profiles in both zones are linear, ie $T_i(z) = a_i + b_i z$, and assuming that the *net volume flow* is zero, ie that the neutral level is located in the middle of the aperture. The expression reads:

$$\Phi_{12} + \Phi_{21} = \frac{C_d \rho\, c_p W}{3} \sqrt{\frac{g}{T}}\, h^{3/2} \Delta T^{1/2} * \left[\Delta T + 0.3h(b_1 + b_2)\right] \quad (W) \qquad [3]$$

and is good approximation when the thermal gradients in both zones are equal, and not too large.

From Eq. [3] it is seen that by including the temperature gradients b_1 and b_2 the heat flow is increased by the factor

$$\left[1 + \frac{0.3\, h(b_1 + b_2)}{\Delta T}\right] \qquad [3b]$$

In practice b_1 and b_2 are not well known, but the importance of this correction factor for small ΔT shows the need to include the effect of stratification in building energy simulation environments like ESP-r.

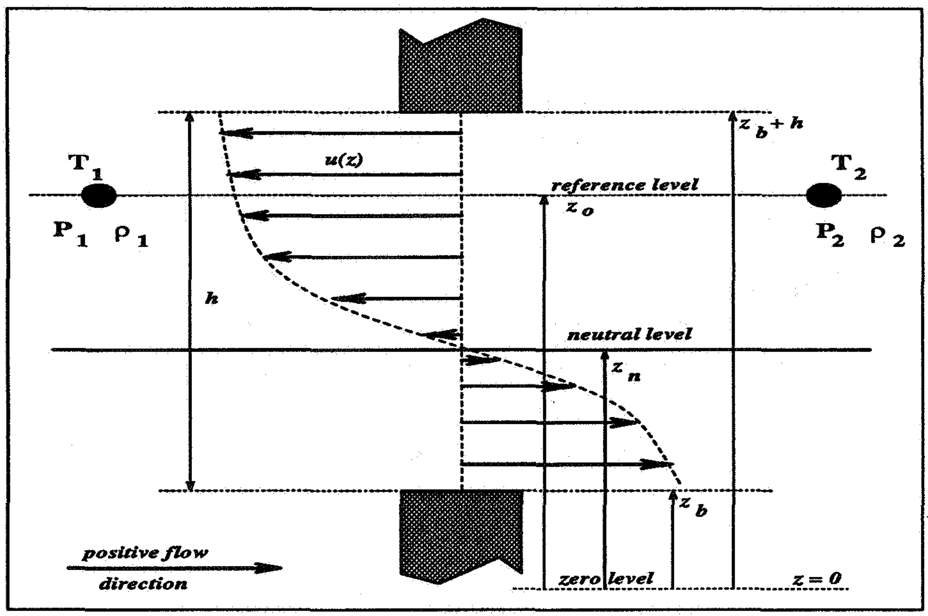

Figure 1 Definition of various parameters

2.2 Mass Flow between Stratified Zones

In the following, an expression for the mass flow through a large opening separating two zones of different temperature and pressure is derived. The general case is considered, ie there is a static pressure difference at reference level z_0 between zone 1 and 2, and different vertical temperature profiles occur in the two zones. These temperature profiles are assumed linear, though the temperature gradients can be different. The situation is depicted in Figure 1.

First we assume that conditions are such that the neutral level, ie the level at which the pressure is equal in zones 1 and 2, is located in the opening, so that *bidirectional* air flow occurs.

The *stack pressure difference* between a point at height z and a point at reference height z_0 is calculated from:

$$\Delta P(z) - \Delta P(z_0) = \int_{z_0}^{z} g\left[\rho_2(\xi) - \rho_1(\xi)\right] d\xi \quad (Pa) \qquad [4]$$

Although density variations due to pressure variations are negligible small, those resulting from temperature differences should be taken into account, especially when temperature gradients are large. The air density in zone i is inversely proportional to the temperature, namely:

$$\rho_i = \frac{P_{ref} M}{R T_i} \quad (kg/m^3)$$

where P_{ref} is some reference pressure, eg the atmospheric pressure, M is the molecular weight of air, and R is the universal gas constant. To a very good approximation one may write:

$$\rho_i = \frac{K}{T_i} \quad (kg/m^3)$$

with K constant. The expression for $\Delta P(z)$ now reads:

$$\Delta P(z) - \Delta P(z_0) = \int_{z_0}^{z} gK\left[\frac{1}{T_2(\xi)} - \frac{1}{T_1(\xi)}\right]d\xi \quad (Pa) \qquad [5]$$

Assuming a linear temperature profile $T_i(z) = a_i + b_i z$ one obtains:

$$\Delta P(z) - \Delta P(z_0) = gK\int_{z_0}^{z}\left[\frac{1}{a_2 + b_2\xi} - \frac{1}{a_1 + b_1\xi}\right]d\xi$$

$$= \ldots\ldots\ldots\ldots$$

$$= gK\left[\frac{1}{b_2}\ln\frac{T_2(z)}{T_2(z_0)} - \frac{1}{b_1}\ln\frac{T_1(z)}{T_1(z_0)}\right] \quad (Pa) \qquad [6]$$

If the temperature gradient in both zones is not too large, we have to a very good approximation:

$$\ln\frac{T_i(z)}{T_i(z_0)} \approx \frac{T_i(z) - T_i(z_0)}{T_i(z_0)} \quad (-),$$

ie the *first order approximation* is highly accurate. Inserting the linear temperature profile gives:

$$\ln\frac{T_i(z)}{T_i(z_0)} \approx b_i\frac{z - z_0}{T_i(z_0)} \quad (-)$$

so that in first order approximation:

$$\Delta P(z) - \Delta P(z_0) \approx gK\left(\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)}\right)(z - z_0) \quad (Pa) \qquad [7]$$

This means that $\Delta P(z)$ changes linearly with the height coordinate z when temperatures in both zones differ at the reference height z_0. Note that the first order approximation results in Eq. [7] which is independent of the temperature gradients in both zones, b_1 and b_2. Inserting the *second order approximation*, ie

$$\ln\frac{T_i(z)}{T_i(z_0)} \approx \frac{T_i(z) - T_i(z_0)}{T_i(z_0)} - \frac{1}{2}\left(\frac{T_i(z) - T_i(z_0)}{T_i(z_0)}\right)^2 \quad (-)$$

in Eq. [6] gives:

$$\Delta P(z) - \Delta P(z_0) \approx gK\left(\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)}\right)(z - z_0) - \frac{1}{2}gK\left(\frac{b_2}{T_2^2(z_0)} - \frac{b_1}{T_1^2(z_0)}\right)(z - z_0)^2 \quad (Pa),$$

showing that the temperature gradients give only a second order contribution to $\Delta P(z)$. The *relative error* made by assuming that

$$\ln\frac{T_i(z)}{T_i(z_0)} \approx \frac{T_i(z) - T_i(z_0)}{T_i(z_0)} \quad (-)$$

is of the order of $(T_i(z) - T_i(z_0))/2T_i(z_0)$. Even for a ceiling-to-floor temperature difference of 6 K, this relative error will be $\approx 1\%$ at most. As in the mass flow calculation the square root of $\Delta P(z)$ is integrated over the height of the opening, the resulting error will even be smaller. In the following, this error will therefore be neglected.

If the opening through which the air flows extends from z_b to $z_b + h$, the pressure difference between the two zones at bottom level z_b will be equal to:

$$\Delta P(z_b) = \Delta P(z_0) + gK\left(\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)}\right)(z_b - z_0) \quad (Pa) \qquad [7a]$$

and at the top of the opening:

$$\Delta P(z_b + h) = \Delta P(z_0) + gK\left(\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)}\right)(z_b + h - z_0) \quad (Pa) \qquad [7b]$$

Now, if EITHER $\Delta P(z_b) > 0$ *and* $\Delta P(z_b + h) < 0$ OR $\Delta P(z_b) < 0$ *and* $\Delta P(z_b + h) > 0$ then the neutral level z_n is located inside the opening and *bidirectional* airflow occurs. If $\Delta P(z_b)$ and $\Delta P(z_b + h)$ have the same sign, or if one of them is zero, only *unidirectional* flow takes place.

According to Bernoulli's Law, a pressure difference $\Delta P(z)$ results in a local air velocity $u(z)$ proportional to the square root of $\Delta P(z)$. Therefore, an infinitesimal volume flow $d\dot{q}$ through an element of height dz in the opening can be written as:

$$d\dot{q} = W\,u(z)\,dz \quad (m^3/s) \qquad [8]$$

If we consider the case where $T_2 > T_1$ and where the pressures at reference level z_0 in both zones are such that the neutral level is located inside the opening (so bidirectional airflow will occur), then the mass flow from 2 to 1 is equal to:

$$\dot{m}_{21} = \int_{z_n}^{z_b+h} \rho_2 d\dot{q} = C_d W\sqrt{2\rho_2} \int_{z_n}^{z_b+h} \Delta P(z)^{1/2} dz \quad (kg/s) \qquad [8a]$$

and the mass flow from 1 to 2 is equal to:

$$\dot{m}_{12} = \int_{z_b}^{z_n} \rho_1 d\dot{q} = C_d W\sqrt{2\rho_1} \int_{z_b}^{z_n} \Delta P(z)^{1/2} dz \quad (kg/s) \qquad [8b]$$

where C_d is an empirical constant.

In these expressions, the error made by placing $\sqrt{2\rho_i}$ in front of the integral sign is negligible because density variations are very small over the integration interval when compared to variations in $\Delta P(z)$. Inserting the linear expression for $\Delta P(z)$ into the integrals gives for $\dot{m}_{21}$ and $\dot{m}_{12}$ the following expressions:

$$\dot{m}_{21} = \frac{2}{3} C_d W\sqrt{2\rho_2}\,\frac{h}{C_t}\,C_a^{3/2} \quad (kg/m^3) \qquad [9a]$$

$$\dot{m}_{12} = \frac{2}{3} C_d W\sqrt{2\rho_1}\,\frac{h}{C_t}\,(-C_b^{3/2}) \quad (kg/m^3) \qquad [9b]$$

where:

$$C_t \equiv hgK\left(\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)}\right) = C_a - C_b \quad (Pa)$$

$$C_a \equiv \Delta P(z_b + h) \quad (Pa) \quad \text{and} \quad C_b \equiv \Delta P(z_b) \quad (Pa)$$

Note that in the situation in figure 1, the pressure difference at the top level of the opening, $C_a \equiv \Delta P(z_b + h)$ is *negative*, so that $C_a^{3/2}$ is an *imaginary* number. To keep the value of $\dot{m}_{21}$ real, the absolute value of C_a should be taken.

It is convenient, however, to write the *net mass flow* of air through the opening as a complex quantity, ie:

$$\dot{m}_{net} = \dot{m}_{21} + \dot{m}_{12} = \frac{2\sqrt{2}}{3} C_d W \frac{h}{C_t} * \left(\sqrt{\rho_2} C_a^{3/2} - \sqrt{\rho_1} C_b^{3/2}\right) \quad (kg/s) \qquad [10]$$

This expression was first derived by Cockroft (1979). The net mass flow is a *complex number*, of which the *real* part gives the flow from 1 to 2 and the *imaginary* part gives the flow from 2 to 1.

It must be emphasized that the Cockroft formula for $\dot{m}_{net}$ in the form given above only holds for the special case depicted in Figure 1! There are two reasons why it is necessary to modify the expression:

i If zone 1 on the left were the *warmer zone* instead of the cooler one, $\dot{m}_{12}$ would take place *above* the neutral level, and $\dot{m}_{21}$ *below* it. The integration interval for both contributions would be interchanged, so that in Cockroft's expression, the term containing C_a is now $\dot{m}_{12}$ and the term containing C_b is now $\dot{m}_{21}$. The formula now reads:

$$\dot{m}_{net} = \dot{m}_{12} + \dot{m}_{21} = \frac{2\sqrt{2}}{3} C_d W \frac{h}{C_t} * \left(\sqrt{\rho_1} C_a^{3/2} - \sqrt{\rho_2} C_b^{3/2}\right) \quad (kg/s) \qquad [10a]$$

However, the *real* part still gives the flow from 1 to 2 and the *imaginary* part still gives the flow from 2 to 1.

ii If the external pressures in both zones differ considerably, the neutral level will shift to a height *below or above* (ie outside) the opening, so that the airflow becomes *unidirectional.* In this situation, one of the flow terms results from an integration over the *entire* opening, ie from z_b to $z_b + h$, while the other term is canceled. In the situation of unidirectional flow, the pressure differences at the bottom and top of the opening, C_b and C_a, have the *same sign* (unless one of them vanishes), so that $C_a^{3/2} - C_b^{3/2}$ is either a *real* or a *pure imaginary* number.

By carefully comparing the expressions for $\dot{m}_{net}$ which can be established for the different cases of unidirectional and bidirectional flow, ie by "tuning" the temperature difference and the pressure difference between zone 1 (left) and zone 2 (right), the following very convenient formula for $\dot{m}_{net}$ which holds in *all cases* can be obtained:

$$\dot{m}_{net} = \dot{m}_{12} + \dot{m}_{21} \quad (kg/s) \qquad [11]$$

$$\dot{m}_{12} = \sqrt{\rho_1}\,\mathrm{Re}(Z_a - Z_b) \quad \geq 0 \quad (kg/s) \qquad [11a]$$

$$\dot{m}_{21} = -\sqrt{\rho_2}\,\mathrm{Im}(Z_a - Z_b) \quad \leq 0 \quad (kg/s) \qquad [11b]$$

where:

$$Z_a \equiv \frac{2\sqrt{2}}{3} \frac{C_d h W}{C_t} C_a^{3/2} \quad (m^2 Pa^{1/2}) \qquad Z_b \equiv \frac{2\sqrt{2}}{3} \frac{C_d h W}{C_t} C_b^{3/2} \quad (m^2 Pa^{1/2})$$

As the direction $1 \rightarrow 2$ is, by definition, the *positive* direction, the contribution $\dot{m}_{21}$ should be *non-positive,* which explains the minus sign appearing in it. The artificial complex quantities Z_a and Z_b are introduced for convenience and have no physical meaning. In the complex plane, $Z_a - Z_b$ is located either on the positive real axis (when there is a unidirectional flow $1 \rightarrow 2$), on the positive imaginary axis (when there is a unidirectional flow $2 \rightarrow 1$), or in the first quadrant of the complex plane (when the flow is bidirectional). When for a given temperature difference between zone 1 and 2 the external pressure difference $\Delta P(z_0)$ is continuously increased from highly negative to highly positive, $Z_a - Z_b$ describes a smooth continuous curve.

2.3 Heat Flow between Stratified Zones

Just as for the mass flow, a convenient expression for the bidirectional heat flow through a large opening between stratified zones can be derived, giving Φ_{12} and Φ_{21} as real and imaginary parts of complex quantities.

Whereas mass flows are calculated by evaluating integrals of the type:

$$\int \rho_i(z)d\dot{q} \quad (kg/s)$$

heat flows are analogously calculated by evaluating integrals of the type:

$$\int c_p T_i(z)\rho_i(z)d\dot{q} = c_p C_d W \int \sqrt{2\rho_i(z)}T_i(z)\sqrt{\Delta P(z)}dz \quad (W)$$

To be able to evaluate these integrals analytically for linear temperature profiles $T_i(z) = T_i(z_0) + b_i(z - z_0)$, the integrand $\sqrt{2\rho_i(z)}T_i(z)\sqrt{\Delta P(z)}$ above, should be of the form $[polynomial] \cdot \sqrt{\Delta P(z)}$, which means that $\sqrt{2\rho_i(z)}T_i(z)$ should be approximated by its "best linear fit", which is (as can be checked easily):

$$\sqrt{2\rho_i(z_0)} \cdot [T_i(z_0) + 1/2 b_i(z - z_0)] \quad (\sqrt{kg/m^3}K)$$

Evaluation of the integrals is a rather laborious task, which will not be documented here due to space constraints. However, when these integrals are worked out in the same way as was done for the mass flows, we obtain convenient expressions for the heat flows Φ_{12} and Φ_{21}, namely:

$$\Phi_{net} = \Phi_{12} + \Phi_{21} \quad (W) \qquad [12]$$

$$\Phi_{12} = c_p\sqrt{\rho_1} * \mathrm{Re}\,(\tilde{T}_{1a}(z_0)Z_a - \tilde{T}_{1b}(z_0)Z_b) \quad \geq 0 \quad (W) \qquad [12a]$$

$$\Phi_{21} = -c_p\sqrt{\rho_2} * \mathrm{Im}\,(\tilde{T}_{2a}(z_0)Z_a - \tilde{T}_{2b}(z_0)Z_b) \quad \leq 0 \quad (W) \qquad [12b]$$

where:

$$\tilde{T}_{ia}(z_0) \equiv T_i(z_0) - b_i h\left[\frac{C_a}{5C_t} + \frac{\alpha - 1}{2}\right] \quad (K) \text{ for } i = 1,2$$

$$\tilde{T}_{ib}(z_0) \equiv T_i(z_0) - b_i h\left[\frac{C_b}{5C_t} + \frac{\alpha}{2}\right] \quad (K) \text{ for } i = 1,2$$

in which α is a dimensionless reference height ($\alpha \equiv (z_0 - z_b)/h$), and the densities ρ_1, respectively ρ_2, are evaluated at the reference level z_0.

3 APPLICATION

Eq. [11] and Eq. [12] have been incorporated into the *large vertical openings* component of the flows solver (Hensen 1991) of the ESP-r building and plant energy simulation environment (Aasem et al. 1993). This particular solver is based on a nodal network mass balance approach, and can be used - amongst others - for multizone modelling of airflow in buildings.
In the following some calculation results are given, which demonstrate the relative importance of the flow governing variables by means of parametric analysis.
For this we started from a base-case involving two building zones connected by a door opening with width $W = 1.0\ m$, height $h = 2.0\ m$, and reference height $\alpha = 0.5$. The discharge coefficient C_d was assumed to be 0.50. Various combinations of pressure difference, temperature difference, and vertical air temperature gradients were considered.

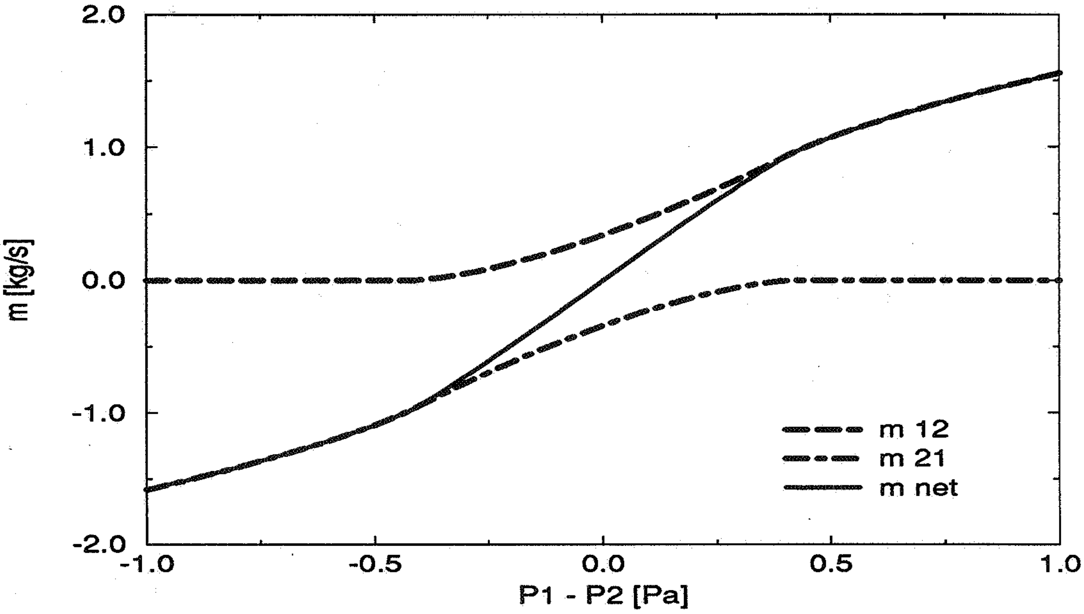

Figure 2 Mass flow rate $\dot{m}$ (kg/s) vs pressure difference $P_1 - P_2$ (Pa) for $|T_1(z_0) - T_2(z_0)| = 10\ K$

Figure 2. shows the mass flow results as a function of the pressure difference between zone 1 and zone 2, for an absolute temperature difference of 10 K. From Eq. [11] follows that the temperature gradients do not influence the mass flows. Flow $\dot{m}_{12}$ will be above the neutral level when zone 1 is the warmer zone, otherwise it will be below neutral level. From the results it is clear that there is only a small band in ΔP for which bidirectional flow occurs. It should be noted however that the corresponding airflows are quite large; eg for $\Delta P = 0.25\ Pa$ $\dot{m}_{12}$ is $\approx 0.75\ kg/s$ or $\approx 2250\ m^3/h$. This implies that there will also be a large heat flow associated with that. If we would make graphs for the heat flows Φ (and assuming that there are no vertical temperature gradients), then the shapes would be quite similar to the ones in Figure 2. Obviously the y-axis values will be different and would range from -600 kW to 600 kW for the range of pressure and temperature differences in Figure 2.

Figure 3. shows the net mass flow results for various absolute temperature differences. At very low or zero temperature difference there will only be uni-directional flow and the airflow will be similar to the flow through a large orifice. Figure 3. indicates that an

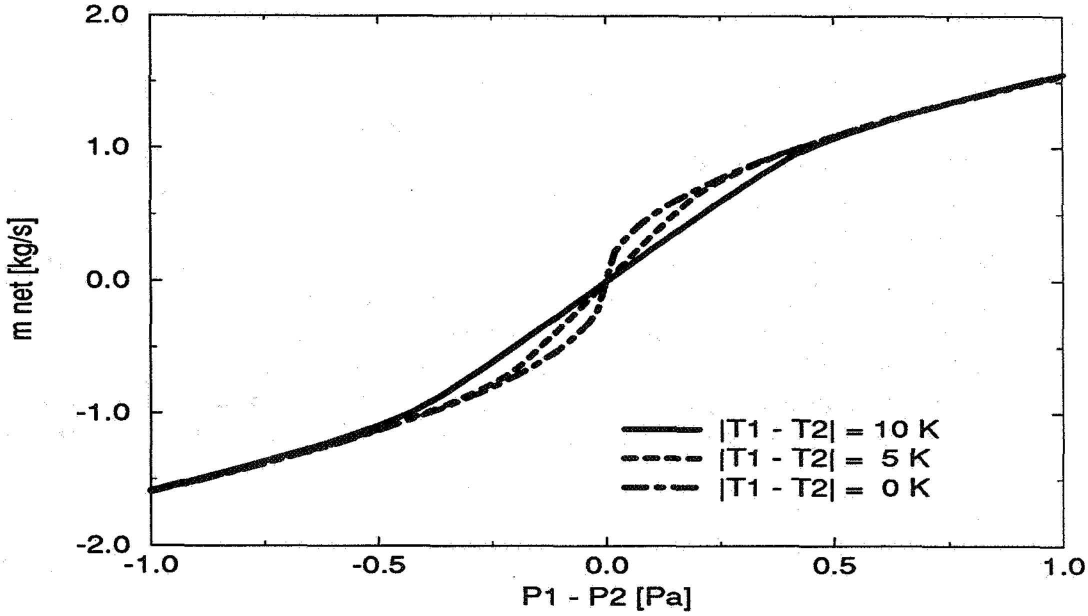

Figure 3 Net mass flow rate $\dot{m}_{net}$ (kg/s) vs pressure difference $P_1 - P_2$ (Pa) as a function of absolute temperature difference $|T_1(z_0) - T_2(z_0)|$ (K)

increase in temperature difference "smoothes" the transition from flow in the direction of $1 \rightarrow 2$ to the direction of $2 \rightarrow 1$ when ΔP changes from positive to negative.

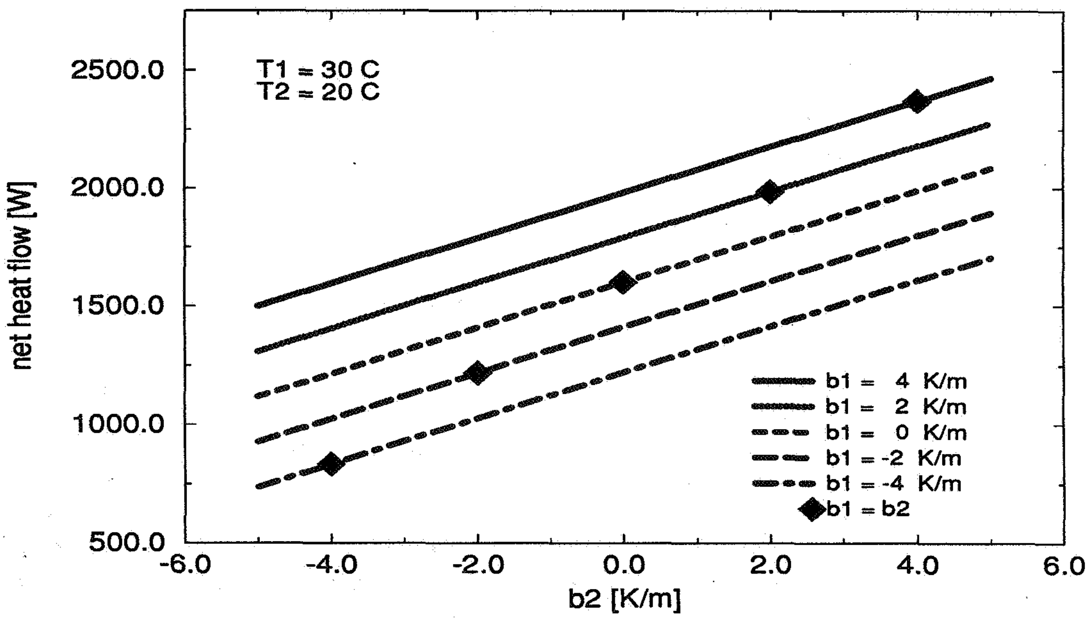

Figure 4 Net heat flow Φ_{net} (W) vs vertical temperature gradient b_2 (K/m) for zone 2 as a function of gradient b_1 (K/m) for zone 1; $\Delta P(z_0) = 0$ (Pa)

As indicated above, the mass flows are not influenced by the vertical temperature gradients. This is clearly not the case for the heat flows as can be seen in Figure 4. This figure shows the net heat flow (ie $\Phi_{net} = \Phi_{12} + \Phi_{21}$) between zone 1 and zone 2, assuming that the reference temperature in zone 1 is 30 °C and is 20 °C in zone 2. For this case there is no pressure difference at reference height; ie $\Delta P(z_0) = 0$ Pa. From Figure 4 follows that net heat flow for this case would be $\approx 1600\ W$ when the temperature gradients would not be taken into account (ie $b_1 = b_2 = 0\ K/m$). If there would only be a gradient in one of the zones (eg $b_1 = 0$ and $b_2 \neq 0$) then (for this particular case) the change in net heat flow is about $100\ W$ for each unit change in vertical temperature gradient. If both gradients are non-zero then the changes can even be bigger as can be seen in Figure 4. For instance for a common case where there are vertical temperature gradients of $\approx 1\ K/m$ in each zone, then the net heat flow would be $\approx 1800\ W$ instead of $1600\ W$, which is a difference of 12%.

5 DISCUSSION

In the literature a number of natural convection configurations are presented where air temperature stratification was observed to be important (Allard et al. 1992, Ch2 and 3). To handle these situations it is important to know when temperature stratification in buildings occurs and how its magnitude could be estimated or predicted. Some examples are therefore briefly discussed. However, all these situations concern zero net flow because other configurations have not been studied in detail in the literature.

Balcomb (1984 and White et al. 1985) studied the heat distribution in more than ten full scale passive solar buildings. He recognized that the heat flow through internal doors (for example to a sunspace) is proportional to the difference in the mixed mean temperatures of the upper and the lower air streams in the opening, T_d, rather then to T, which is the difference between the mean temperatures of the connected zones. He found invariably that in these buildings the ratio T_d/T was large and typically 1.3 (and as high as 1.6) during the day, for a T ranging between 3 and 12 K. Assuming linear air temperature gradients in the zones (typically 1 to 2 K/m) Eq. [3] was used (White et al. 1985) to calculate the heat flow from the measured temperatures and velocities. It was suggested that the degree of stratification will depend on the rate of heat exchange but no method was proposed to predict stratification.

Boardman (1989, Scott et al. 1988, Neymark et al. 1989) studied the influence of aperture dimensions on interzonal natural convection in experiments on both an air filled full scale enclosure and a waterscale model. During each run the temperature difference between the hot wall in the hot zone and the cold wall in the cold zone, ΔT_{hc}, was kept constant. In this way heat flow through a door in a solar house was simulated. To describe the experimental results an isothermality factor is defined as $q = \Delta T / \Delta T_{hc}$ where ΔT is the difference between the mean air temperatures in each zone. While the mean zonal temperature difference ΔT was initially close to ΔT_{hc} ($q = 1$), increasing the opening height caused both the temperature gradients b_1 and b_2 to increase and q to approach zero. The temperature drop ΔT_{hc} was finally concentrated in the boundary layers near the hot and cold walls. Using Eq. [3] for the heat flow gave consistent results. From the assumption that stratification scales linearly with the overall length scales and temperature differences, the gradients are written as (Scott et al. 1988):
$b_1 + b_2 = F(\Delta T_{hc}/h)(1 + h/h_r)$ (h is door height and h_r is the room height). Comparing the thermal resistances of the door and the wall boundary layer, the data analysis yielded $F = 0.3$, in other words $(b_1 + b_2)$ was between one and two thirds of $\Delta T_{hc}/h$, the theoretically maximum of the gradient in the opening. Although, the result seems only strictly valid for the configuration of the experiment, it helps to understand how temperature gradients are set up in buildings and what their order of magnitude can be.

Allard et al. (1992) report on test-cell experiments to study mass and heat transfer through open doors under different heating and cooling configurations.

In the first test-cell (Allard et al. 1992, Ch.3.2), the interzone temperature difference is created using heating and cooling plates and the door height is 2 m. The temperature gradients b_1 and b_2 were typically 3 K/m, for a ΔT of about 2 K, and the heat flow increasing factor (Eq. 3b), becomes as high as 2.8. However from the measured velocity profile, it was found that the discharge coefficient is $C_d = 0.3$ rather than 0.6.

In the second test-cell (Allard et al. 1992, Ch.3.3), the interzone temperature difference over the 2 m high door is created using a hot and cold wall, a configuration similar to Boardman's (1989, Scott et al. 1988, Neymark et al. 1989). Varying ΔT_{hc} over the range -5 to 30 $°C$, the isothermality factor q stayed close to 0.1. The gradient was roughly linear ranging between 0.5 and 4 K/m. Using Boardman's model $b_1 + b_2 = F(\Delta T_{hc}/h)(1 + h/h_r)$ it is noted that in all 5 experiments the gradient scaled with ΔT_{hc}. Forcing over the height of the doorway, a straight line through the data, a factor $F = (0.23 \pm 0.3)$ is obtained with an uncertainty on the fitting procedure of an additional ± 0.5 K/m. However from the measured velocity profiles, discharge coefficients between 0.27 and 0.54 were derived without an apparent correlation with the experimental configuration.

These two cases show clearly that for the configuration with hot and cold walls the temperature stratification can be estimated, but that the use of Eq. [3] without detailed knowledge on both b and C_d (Pelletret et al. 1991) imply large uncertainties in the heat flow calculation.

Studies of the temperature stratification in closed rooms and for various heater configurations are quite numerous in the literature (see References in (Inard 1988) and (Inard and Buty 1991). The stratification varies strongly. For example for the case of floor heating it is weakest and for ceiling heating it is strongest. For the case of the convective heater, the gradient in test cells was found to be correlated with the convective heating power (Inard 1988). In addition to this, Allard et al. (1992, Ch.4.5) pointed out that there is an analogy between a convective heater and an open door or window in the sense that the stratification caused by open windows appeared to vary with power as in (Inard 1988) when the ventilative cooling power is used). This idea has not been worked out however. In particular the gradient must be correlated with the floor area and depend on the power density rather than on power.

To cope with these uncertainties and to better understand the dependence of both the discharge coefficient and the stratification on the heating and cooling configuration, Allard et al. (1992, Ch.3.5 and 3.7) proposed that future work should include a systematic comparison between numerical computations (CFD) and simplified models.

Finally, this means that although the use of the new algorithm in ESP-r requires expertise at present in the form of input parameters b and C_d, it can be expected that models will be developed that are able to predict stratification for each particular zonal configuration.

5 CONCLUSION

A general solution is presented for predicting (net) airflow and (net) heat flow through large vertical openings between stratified building zones. The solution proved to be compatible with a nodal network description of leakages for multizone modelling of airflow in buildings. By parametric analyses, the relative importance of the flow governing variables is demonstrated. From the results it is clear that - apart from the other governing variables like pressure and temperature difference - vertical air temperature gradients have a major influence on the heat exchange by inter-zonal airflow.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Survey of Mechanical Ventilation Systems in 30 Low Energy Dwellings in Germany

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1 Synopsis

This paper shows preliminary results of 18 out of 30 inspected ventilation systems in low rise, low energy residential buildings. We propose a method for the assessment of energy efficiency of ventilation systems.

The majority of the inspected exhaust systems fulfills the conditions for the demanded air flow rates and energy efficient operation. However, typically the distribution of airflows to the rooms of the supply zone is rather weather dependent due to insufficient airtightness of the buildings and large stack heights.

2 of 5 exhaust supply systems with heat recovery mismatch energy efficient operation due to high pressure drops. The airtightness of the buildings is insufficient.

Generally, there is a lack of operation and maintenance instructions. By optimized ductwork, fans, motors, and controller the electricity consumption could be reduced by more than 50%.

2. Introduction

In Hesse, a state of the Federal Republik of Germany, since 1987, a increasing number of low energy houses were constructed. This development was mainly due to political measures and sponsoring by the Hessian governement as well as the work of the Institut Wohnen und Umwelt (IWU). One prerequisite for support from the sponsoring program was a mechanical ventilation system, which was demanded mainly by air quality reasons..

In 1993, a program was set up to investigate the performance of the supported ventilation systems. This work was done by the consulting office ebök under contract of the IWU, financed by the "Hessisches Ministerium für Umwelt, Energie und Bundesangelegenheiten". This paper covers preliminary results of 18 of the 30 tested ventilation systems. A final report will be available at the end of 1994.

3. Research Planning

3.1 Types of Buildings and Systems

All systems were installed in 2 to 3 storey 1 or 2 family houses or terraced houses.

The following system types are included in the study

- Exhaust air with manual fan speed control (6 systems) (E_S)
- Exhaust air with humidity control (7 systems) (E_H)
- Exhaust and supply air system with heat recovery by heat exchanger (5 systems) (ESX)

3.2 Measurement Techniques

Since the ductwork contained no designed measurement planes, measurement of air flow rates, pressure drops etc. were often difficult to perform. Therefore a number of different measurement devices and techniques in accordance with /VDI 2079/ and VDI /2080/ were used. Measuring equipment and typical resulting errors are as follows:

- Power demand by digital wattmeter (typical error 5% of reading).

- Pressure levels by Pitot tube and digital micromanometer (typical error 6%, up to 20% o.r. at very low pressure differences).
- Air flow rate calculated by air velocity measurement by heated wire anemometer (typical error about 14% to 22% o.r.).
- Air flow rate by dynamic air speed indicator (System Halton) (typical error 7% to 12%).
- Air flow rates at air terminals by anemometer-hood (typical error 15% o.r.).
- Relative air flow distribution at terminals by pressure drop factors (typical errors 20% o.r.).

3.3 Design Conditions and Assessment Standards

The design conditions were defined as follows:

- 30 m^3/h outside air flow rate per person, at least 0.3 ac/h, and no more than 0.8 ac/h.
- All rooms with increased humidity or odour emissions are to be equipped with exhaust vents. According to /DIN 1946/ part 6 (draft) minimum air flow rates are established: kitchen 60 m^3/h, bathroom 40 m^3/h, toilet 20 m^3/h, at minimum air exchange rate of 2 ac/h. Minimum air flow rate for integrated cooker hoods 120 m^3/h.
- Living rooms with supply vents (ESX) or outside air supply vents (E) and openable windows.
- Openings in interior walls or doors to allow air flow from supply rooms to exhaust rooms.
- Demand controllable total air flow rates, at least 2 levels, 100% and 50%, of the design condition. An adjustable distribution of supplied air is desirable.
- No disturbing noise levels or draughts produced by the ventilation system.
- Good conditions for inspection and maintenance.
- Energy efficient operation of the ventilation system.

It is assumed that all living rooms and the kitchen have openable windows to allow additional natural ventilation on demand and during summertime. No severe indoor production rates of contaminants, for example radon or formaldehyde, should be present.

3.3.1 Energy Efficiency

Today German building code /WSVO 1993/ is revised for environmental reasons. There is a statement for ventilation systems with recovery by air-to-air-heat-exchanger that the ratio of recovered useful heat to electricity consumption (COP) should exceed a factor 5. This is motivated by different emission levels into the atmosphere by generation of heating energy and electricity.

Using the heating degree day method /HMWT 1990/ specific ventilation energy losses Q_{ex} by 1 m^3/h air flow rate are calculated under typical German weather conditions for low energy houses over one heating period (degree day limits 20°/12°C, heating degree days dd=3400 Kd, specific heat capacity of air $c_{p,air}$=0.34 Wh/(m^3K)). They amount to

$$Q_{ex} = c_{p,air} * dd * 24 = 0.34 * 3400 * 24 = 27744 \, [Wh^2/m^3] \quad (1).$$

In order to reach the COP of 5, the maximum allowable air-flow-specific electric power consumption, is calculated by

$P_{spez,max,ESX} = Q_{ex} * eta_X / (t_{op} * COP)$ (2).

Assuming a mean recovery effectiveness eta_X of 70% and an operation period from 1 Sep. to 31 May (operating time t_{op}=6552 h/year), yields the limiting value of air-flow-specific power

$P_{spez,max,ESX} = 0,61\ Wh/m^3$.

This number is used as a threshold condition for the energy efficiency of ESX systems.

Assuming an exhaust only system to be one half of an ESX-system the limit for energy efficient exhaust systems amounts to

$P_{spez,max,E} = P_{spez,max,ESX} * 0.5 = 0.3\ Wh/m^3$.

4. Results

4.1 System design

Tab. 1: Basic data of buildings and design values of ventilation systems (E_S: exhaust system fan speed controlled, E_H: exhaust system humidity controlled, ESX exhaust supply system with air to air heat exchanger

system name and type	ventil. volume [m^3]	liv. area [m^2]	actual occup. [person]	design air flow rate [m^3/h]	area spec. rate [$m^3/(hm^2)$]	air exch. rate [ac/h]
FR, E_S	422	179	5	180	1,01	0,43
BU, E_S	300	131	4	160	1,22	0,53
BE, E_S	477	205	3	150	0,73	0,29
HA, E_S	366	140	4	140	1,00	0,38
WH, E_S	513	201	4	180	0,90	0,35
MU, E_S	341	131	5	180	1,37	0,53
WU, E_H	501	188	5	180	0,96	0,36
LA, E_H	476	181	2	180	0,99	0,38
HG, E_H	515	194	3	180	0,93	0,35
HL, E_H	347	118	2	100	0,85	0,29
GS, E_H	359	144	4	140	0,97	0,39
FL, E_H	327	116	3	120	1,03	0,37
KU, E_H	768	307	7	300	0,98	0,39
PR, ESX	427	178	4	180	1,01	0,42
OT, ESX	450	201	5	180	0,90	0,40
RK, ESX	910	441	6	300	0,68	0,33
SC, ESX	389	168	4	160	0,95	0,41
WL, ESX	465	150	3	180	1,20	0,39
mean val.	464	187	4	177	0,98	0,39

Tab. 1 shows the basic data of the systems and buildings. The system name is an internal code.

In most buildings, assignment of rooms to supply or exhaust zones was correct.

4.2 Air Flow and Air Exchange Rates

Of the inspected 18 systems 14 almost met the design values. 3 systems had too low air flow rates. The reason was mainly due to high pressure drops caused by poor design or installation of ductwork. In 1 exhaust system the distribution of exhaust air to the rooms was totally wrong, because some vents were taped or not installed.

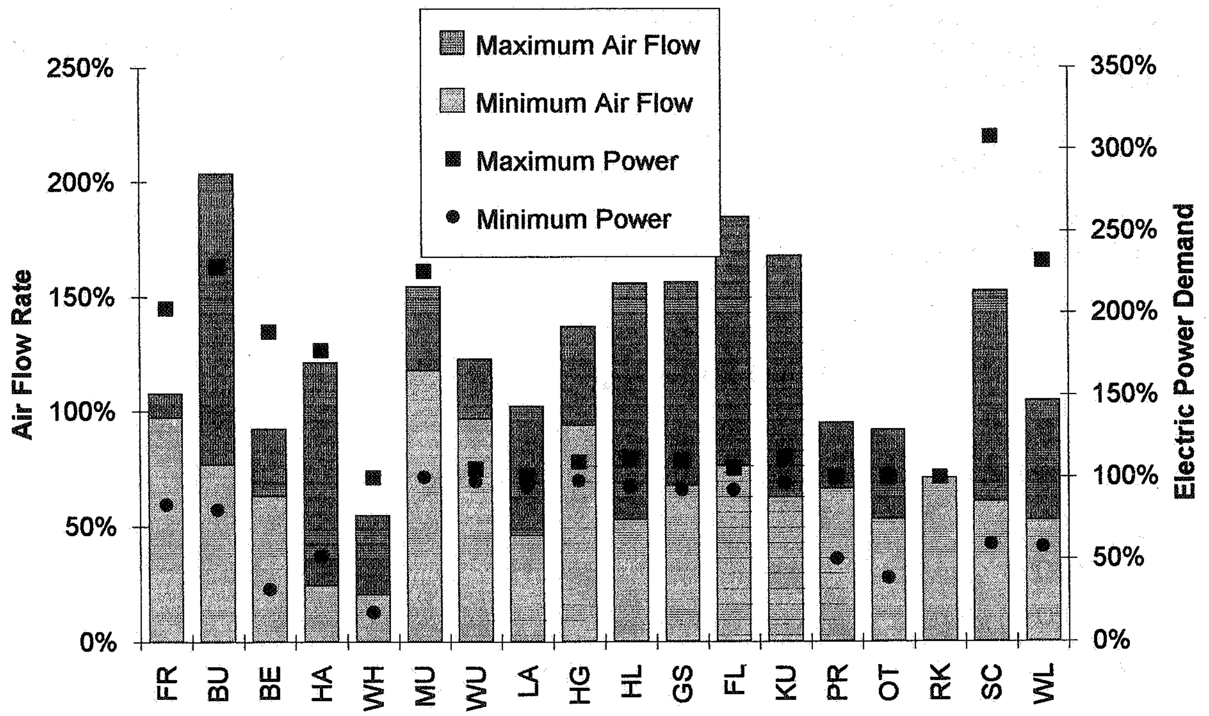

Fig. 1: Control range of air flow rates and electric power demand applied to values at design condition level

Moreover, most systems exhibited more or less severe weaknesses:

- In 9 out of 13 buildings with exhaust systems airtightness was insufficient or the stack hight to large, so the ventilation rate and especially the ventilation of rooms with outside air supply vents was strongly influenced by stack and wind generated forces. Visible indicator for stack driven exfiltration in some buildings was the dust deposition on the filters of the outside air supply vents: at the ground floor the filter was dirty on the outside, at the middle floor dirt settled on both sides, at the upper floor mainly the inside of the filter was dirty. In some rooms of these houses regularly additional ventilation by windows will be necessary.
- The airtightness in all buildings with ESX systems was insufficient compared to recommendations /SIA 180/. This will result in considerable additional in- and

exfiltration. The ventilation losses of the buildings will be considerable higher than predicted by calculations, assuming an airtight envelope /Werner 1993/.

- Air flow rate of integrated cooker hoods were not sufficient for a high capture capacity.
- For some systems sound pressure levels were to high in the design level position, in some cases this was caused by missing sound attenuators, in some cases by sound generation in ductwork.
- In some systems draughts were found due to wrong placement or wrong type of supply vents.

For about 50% of the systems the range of air flow rate control was not sufficient. For speed controlled systems this was due to wrong balancing between the characteristics of ductwork and fan or oversized fans. For the humidity controlled systems this was due to high pressure losses in the ductwork compared to the pressure drop of the humidity controlled air outlet. One of the systems had only an ON-OFF-switch. Fig. 1 shows the relative variation of air flow rate by control and the electric power demand applied to design condition levels.

4.3 Maintenance and Inspection

Almost no operating and maintenance instructions for the ventilation systems were available, in some cases there were data sheets by component manufacturers.

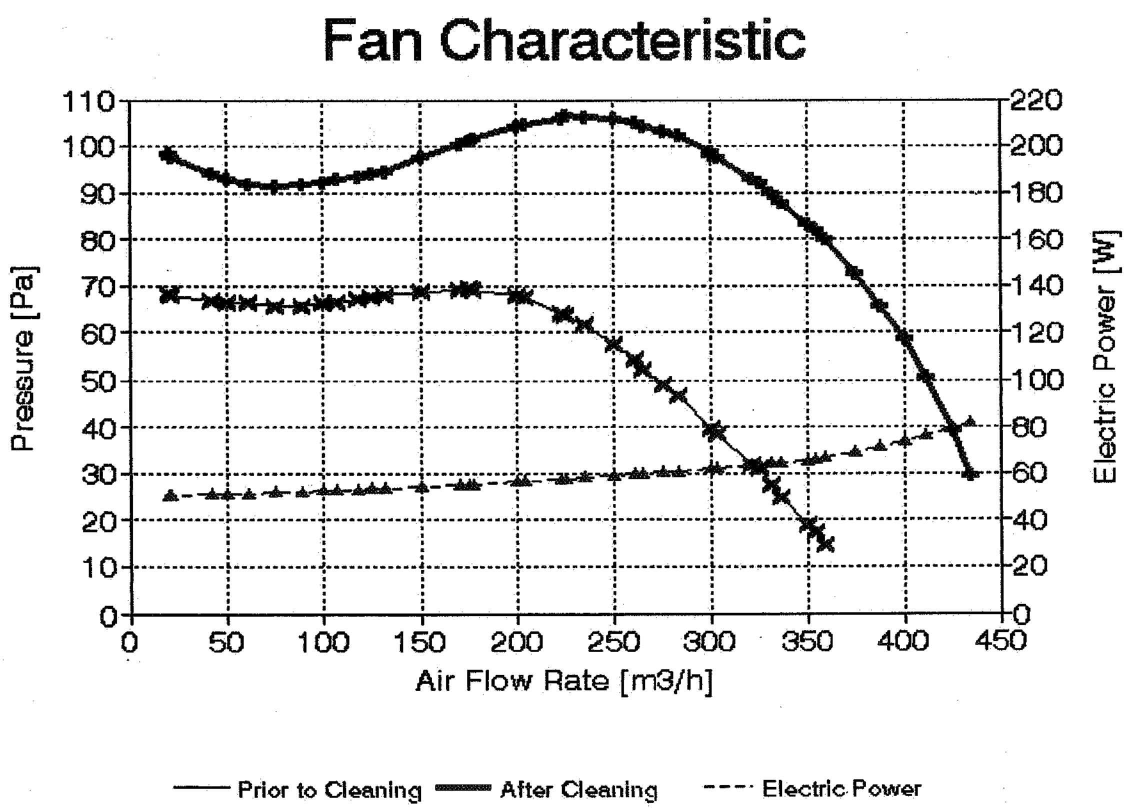

Fig. 2: Characteristic of a fan prior to and after cleaning.

Accessibility of fans and filters for inspection and maintenance purposes was often poor. In many cases this was obviosly due to total lack of planning:

- Turning some fan housings by 180° would improve the accessibility of the maintenance flap and also reduce bends in the ductwork.
- Removal of one filter is impossible because of a lateron installed thermal insulation of a hot water storage tank.

In many cases dirt was found in fans and ductwork dating from the construction period 2 or 3 years ago, besides dust, also pieces of polystyrene insulation were found, wich was a reason for too low air flow rates and noise nuisance.

Frequently fans, filters, or vents were not clean, especially in the systems with integrated cooker hood. Fig. 2 shows the characteristics of pressure and electric power of a fan prior to and after cleaning /Rochard 1994/. Improper maintenance is the most promiment reason for reduced air flow rates and air quality problems.

None of the systems had inspection protocols or tables with adjustment dimensions for vents or other adjustable parts.

Since the occupants of the inspected single family and terraced houses are not experts in ventilation systems, detailed and comprehensible maintenance, instruction, and operation documents are indispensible.

4.4 Ductwork

In all systems ductwork consisted of circular tubes, normally made by metal sheet coated with zink or corrugated flexible metal tubes, in one case plastic tubes of plumbing system type were used.

Typical pressure drops of the inspected exhaust systems amounted to 75 to 100 Pa, for ESX systems a range from 100 to 300 Pa was found (cumulated of exhaust and supply ducts). Recalculations of the pressure drops of the ductwork typically showed possible improvements: avoidable bendings, too narrow diameters, sharply bended or squashed flexible tubes, wrong air outlets and so on.

Improper fixing or jointing of ducts was frequently found, some ducts were found to be completly disjointed (tape got loose and the wrongly fixed tubes slipped away).

4.5 Efficiency

Measurement of the total air flow, the total static pressure in front of and after the fan, and the energy consumption of the motor were used to calculate the systems' overall efficiency (Fig. 3). The efficiency increased with the size of air flow. Except one, all fans were of radial type with foreward leaning fan blades.

The low efficiency of the systems was due to blade and motor type used in small fans, electronic motor controllers, the position of fan blades in the casing, dirty blades, and working conditions outside the range of optimum fan efficiency.

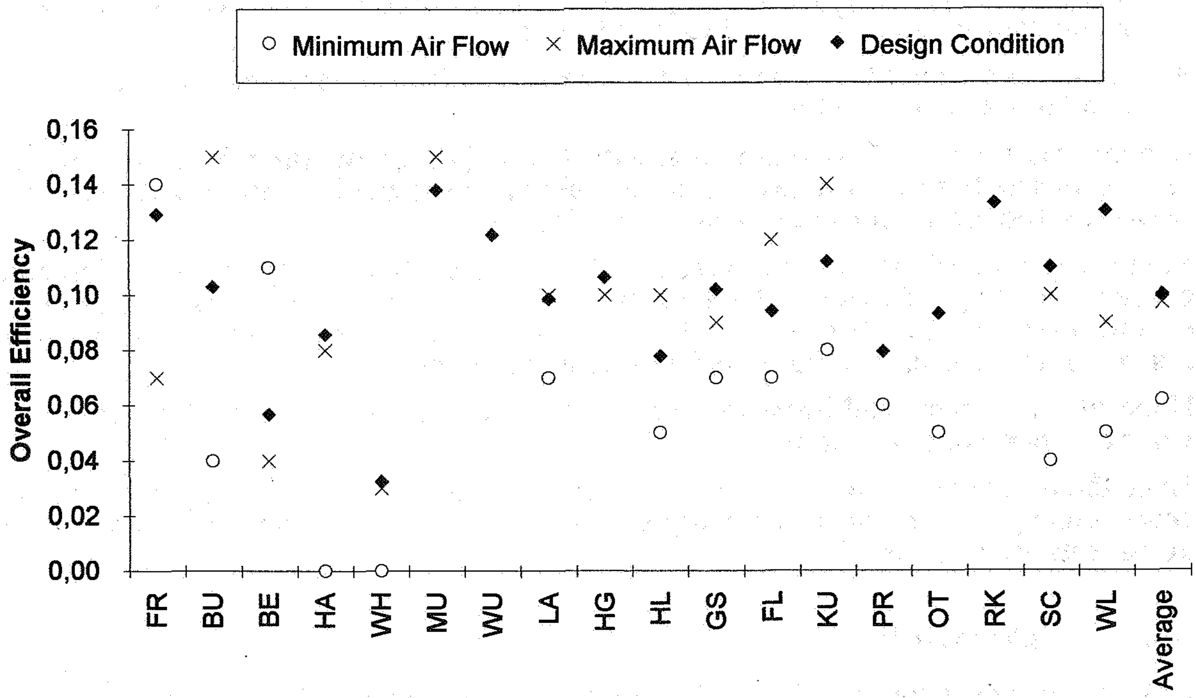

Fig. 3: Measured overall efficiency of ventilation systems

4.6 Electrical energy consumption

4.6.1 Exhaust Systems

The mean electric power of the fans at design conditions ranged from 31 to 76 W, the average air flow specific power amounted to 0,27 Wh/m^3, values spread between 0,17 and 0,57 Wh/m^3, 3 out of 13 exhaust systems exceeded the limit of 0,3 Wh/m^3. Improving the ductwork will lower electricity consumption in speed controlled systems. In some humidity controlled systems the fan capacity was too high for the designed air flow rate.

Under design conditions the electricity consumption was calculated for 6000 operating hours using the measured power demand (Tab. 2).

Tab. 2: Mean, minimum, and maximum values of calculated annual energy consumption per m^2 of living area of exhaust systems

mean [kWh/(m^{2}a)]	minimum [kWh/(m^{2}a)]	maximum [kWh/(m^{2}a)]
1,49	1,21	1,97

4.6.2 Supply Exhaust Systems with Heat Recovery

The mean electric power accumulated of both fans at design conditions ranged from 39 to 151 W, the average air flow specific power amounted to 0,54 Wh/m^3, values spread between 0,22 and 0,91 Wh/m^3. Two sytems have a very low value < 0.3 Wh/m^3, they were found to possess a very good hydraulic construction of the casing of heat exchanger and fans. Two systems show high values > 0.8 Wh/m^3, they exibited relatively high pressure drops inside the casing due to hydraulic construction and additional heat exchangers for electric heat pumps.

Under design conditions the electricity consumption was calculated for 6000 operating hours using the measured power demand (Tab. 3).

Tab. 3: Mean, minimum, and maximum values of calculated annual energy consumption per m^2 of living area of exhaust supply systems

	mean [kWh/(m^{2}a)]	minimum [kWh/(m^{2}a)]	maximum [kWh/(m^{2}a)]
	2,79	1,32	4,85

5. Measures for Better Efficiency

Possible measures for better efficiency of the systems are:

- Correct design of ductwork.
- Correct choice of fans for operation in the optimum range of fan efficiency.
- Correct adjustment of ductwork.

The average specific consumption of the tested exhaust systems could be improved by the above listed measures to 40%. For ESX systems the possible reduction would lower the the specific consumption by about 50%.

Taking into account newly developed technologies now available also for small ventilation systems (more efficient AC motors, improved speed controllers or DC motors), the specific energy consumption could be lowered from the present mean level by about 3/4 for exhaust systems and about 2/3 for ESX systems. More details are given in /Rochard 1994/. Two prototype systems currently under investigation show promising preliminary results.

6. Conclusions

- In general, the tested ventilation systems fulfill the requirement of energy efficiency. Nevertheless there is still a significant potential for improvement.
- Airtightness of buildings is insufficient. This leads to increased ventilation losses in buildings with ESX systems and to weather dependent ventilation of the supply zone in buildings with exhaust systems.
- Most ductwork and fans are far from the optimum performance. This corresponds to the result of almost total lack of design documents.
- Maintenance of the systems is unsatisfactory. This corresponds to complete lack of maintenance instructions and the poor accessibility to filters and fans, found frequently.

- To improve system performance in the future, better knowledge of architects, engineers and craftsmen is inevitable.
- Developed and available technologies with higher efficiencies should be applied for ventilation systems of small buildings too.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Simple and Reliable Systems for Demand Controlled Ventilation in Apartments

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SYNOPSIS

The paper is presenting experience from a several year long time of operation in a group of apartment buildings in the Stockholm area, Sweden, having an extremely low energy usage, less than 110 kWh/(m2.year), electricity supply to the building services included.

The system solution used has a very low pressure drop in the exhaust ducts. Every exhaust point is connected to an individual duct leading to a fan chamber in the attic. The pressure in that chamber is kept constant. The attendant in a flat wanting a higher flow rate starts an individual booster fan situated at the top of his own duct. Supply air is furnished by valves installed in the external walls of the flat. Balancing is made in the fan chamber only. Thus nobody can arrange a higher base flow rate for an individual flat without having access to the fan chamber.

The investment level is comparable to that of a traditional system. Duct dimensions are chosen so as to allow them to be built-in into the walls. The system, which was designed by Mr Henry Willman of HEWAB Engineering, Stockholm, is applicable for offices and the like with or without a mechanical air supply system.

1. DCV IN GENERAL

Demand Controlled Ventilation is a method to achieve the energy conservation wanted and at the same time serve those residing in the treated space with an acceptable air quality. The user's rightful requirements on air treatment systems are in short mainly the following:

a. Clean air in the occupation zone
b. Demand controlled flow rate
c. Individual climate in room where you stay permanently
d. Energy efficient operation
e. Simple and easily understandable design
f. High availability
e. Low life cycle cost for building and installations

In most cases these requirements are also in line with those set by the owner as well as the administrator and the operator, to whom low LCC requirement normally leads to low density of automatic systems.

2. BALANCING - A LIFE CYCLE PROCESS

Most systems are built in the form of stems and branches. Thus, in order to achieve a correct distribution of air, supply or exhaust, in all branches and air terminal devices (ATDs), it is necessary to balance the system. Normally the ATDs are furnished with or themselves act as pressure reducing valves. Here is one of the major problems: The user of the apartment or office room can himself (mostly the responsible are men !) "adjust" the pressure drop of the newly set position of the ATD. Thus the system is once mor out of balance, meaning that some room get too much air and some too little. The way of coping with this problem is to put all types of adhusting valves outside the apartment or office. By doing so you win twofold: No uauthorized tampering with the valves is possible, and no admittance to the apartment or office will be necessary for the purpose of balancing. A further step is to arrange the system so that each ATD is connected to the fan room by a separate duct, in which the adjustiable valve is inserted.

3. DCV IN APARTMENTS

According to IEA Annex 18, "Demand Controlled Ventilation Systems", DCV, the main environmental problem in apartments is moisture. The maximum level of relative humidity of room air, 55% to prevent dust mites, 75% to prevent mold growth in building material, can be met by using DCV. Ventilation systems for appartments can be designed so as to allow for individual flow rate control without a complicated technical equipment. A heat pump system using exhaust air and to some extent outdoor air as a source renders the system a very low energy usage for heating, ventilating and production of heated tap water. The amount of refrigerant in the system can be kept at a low level by using a package heat pump unit connected to a water circulation system with a heat exchanger in the exhaust air, see figure 1.

The physical background to the fact that the so-called "Controlled Ventilation", using high pressure drop in the air terminal devices, cannot fulfill the requirements on a reasonable noise level and energy demand in a system where ventilation flow rate is increased at certain times, is shown by simply using the Bernoulli therorem, see figure 2.

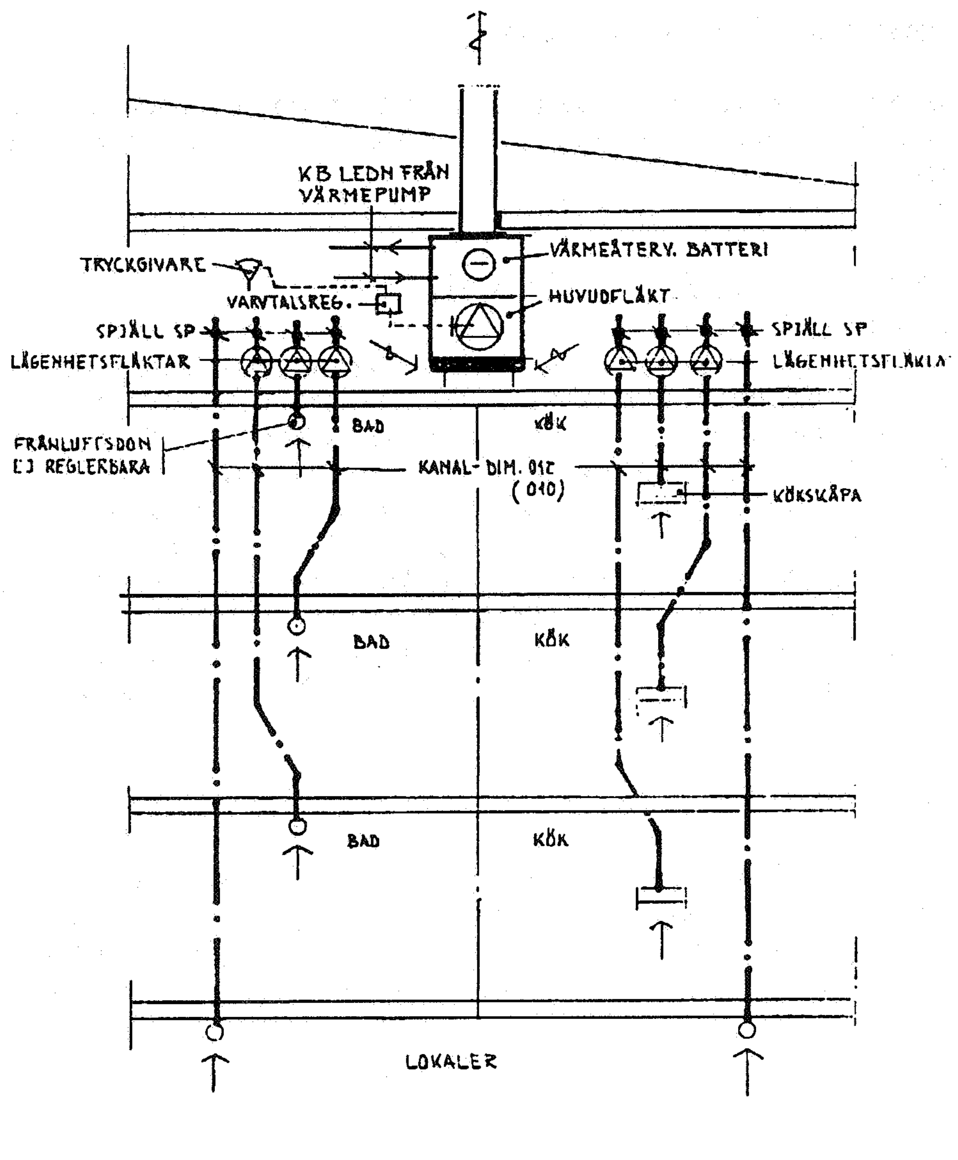

Figure 1. System layout for a domestic building

The pressure drop(δp) per unit length of a duct is approximately proportional to the square of the velocity(v) of the flow:

$$\delta p = \frac{\rho * v^2}{2} * k \quad Pa/m \qquad (1)$$

Here ρ= air density, about1,2 kg/m3
k = "constant", depending of the smotheness of the duct surface

The flow rate(q)is a function of the velocity(v) and the cross area (A) of the duct:

$$q = v*A \quad m3/s \qquad (2)$$

For a duct of circular cross area and diameter (d) we find:

$$A = \frac{\pi * d^2}{4} \quad m2 \qquad (3)$$

The "constant" (k) in eqv(1) can be set to (for given velocity range and surface smotheness):

$$k = \frac{0,02}{d} \qquad (4)$$

The fan power P) is calculated from:

$$P = q*\delta p \quad Watt \qquad (5)$$

The pressure drop, under circumstances given, will become a function of the second power of the flow rate and of the fifth power of the diameter: r

$$\delta p = \frac{\rho * v^2}{2} * \frac{\lambda}{d} = \frac{\rho * \lambda}{2d} * \frac{16q^2}{\pi^2 * d^2 * d^2} = K * \frac{q^2}{d^5} \qquad (6)$$

$$\text{For } q=3q \text{ we get } \delta p_1 = \delta p * \frac{3q)^2}{q^2} = 9 * \delta p \qquad (7)$$

If the original pressure drop is 300 Pa, the new pressure drop will become 9*300 = 2700 Pa

Figure 2:The rules of flow as applied to a duct system and a ventilator

4. DCV IN OFFICES, SCHOOLS AND DAY NURSERIES

The solution described above is also applicable for office buildings and other buildings of the same character, for instance schools and day nurseries, with or without a mechanical supply system and for all types of external energy supply.

The supply unit does not need connection to a heating system, as the unit is furnished with a high efficiency heat exchanger, see figure 3. The characteristics of such a system can be studied in figure 4.

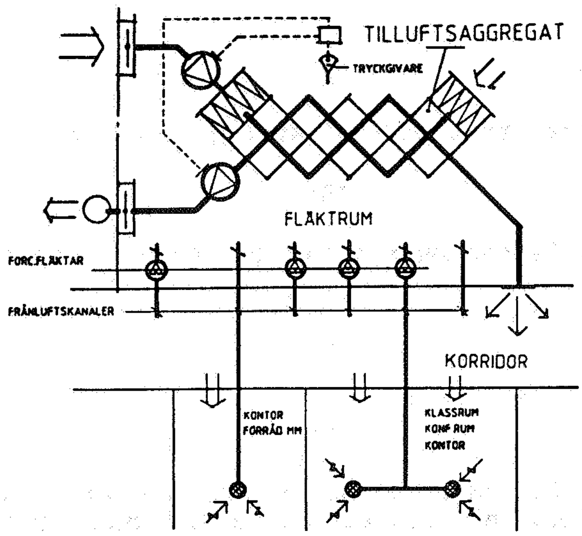

Figure 3. System layout for a DCV system in an office building

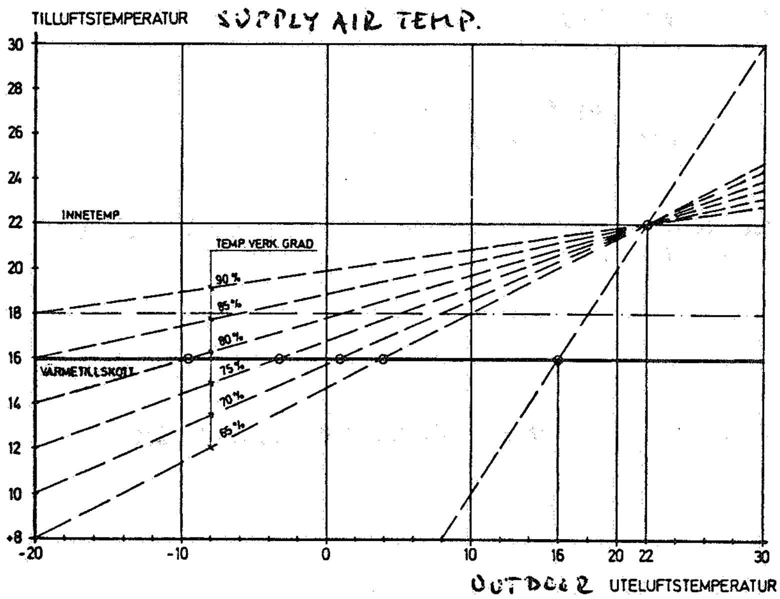

Figure 4. The influance of efficiency of a heat exchanger

The attendant in a room wanting a higher flow rate starts an individual booster fan situated at the top of his own duct. Supply air is furnished either by valves in the external walls or by a supply air unit operated in parallell with the exhaust unit,

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Ventilation Concept, Indoor Air Quality & Measurement Results in the "Passivhaus Kranichstein"

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1 Synopsis

The "Passivhaus Darmstadt-Kranichstein" is a 4 unit terrace house with an extremely low total annual energy consumption of less than 32 kWh/m^2 of living area, thereof about 12 kWh are needed for room heating /Feist 1994/. The determinig factors for the low consumption are the superinsulation, airtightness of the thermal envelope in combination with a highly efficient VAV ventilation system, and an improved window construction.

The "Passivhaus" therfore is a typical example of an improved low energy house. The results of a detailed monitoring program allow decisive statements concerning reduction of energy consumption, relief of environement, indoor air quality, and thermal comfort.

The key results concerning indoor air quality can be stated as follows: The concept of a low energy building with mechanical ventilation system combinated with low emission materials inside the house guarantees good indoor air quality and thermal comfort at relatively small air exchange rates and very low heating demand.

2 Research Project Passivhaus

The "Passivhaus" was planned by the architects Prof. Bott/Ridder/Westemeyer, scientific consulting was done by the "Institut Wohnen und Umwelt" (IWU) and Prof. Bo Adamson (Lund University). The house was built in 1990/91, since October 1991, 4 families have been living in it. A grant was given by the "Hessisisches Ministerium für Umwelt, Energie und Bundesangelegenheiten" (HMUEB).

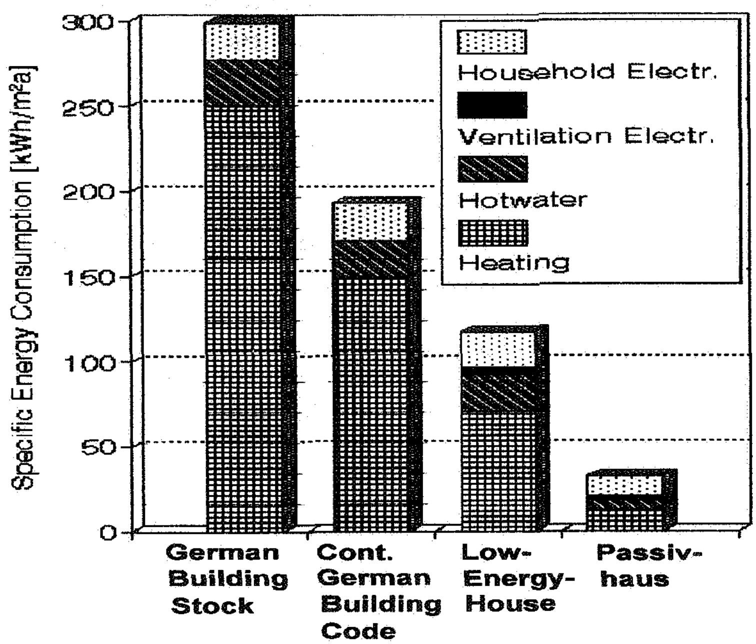

Fig. 1: Spec. energy use: average of German stock, contemporary German building code, low energy house, Passivhaus

The first objective of the reserach project was to check the possible amount of energy savings for heating, hot water and household electricity by passive measures. An analysis of the energy balance over two years demonstrated that the expected reduction of 90% was attained compared to the average contemporary German consumption (Fig. 1).

The monitoring program (Oct 1991 to Sept 1993) was performed by the consulting office "ebök" under contract of the "Wüstenrot Stiftung Deutscher Eigenheim e.V". With grants of HMUEB the program is continued.

3 Passivhaus Darmstadt-Kranichstein

The building is east-west-oriented with an unshaded south facade and has a relatively small surface/volume ratio. The heated floor area of each unit amounts to 156 m^2. The thermal envelope is superinsulated, high effort was put into reduction of thermal bridges and air leakages. Tab. 1 shows the construction of the thermal envelope /Feist 1993/.

Tab. 1: Thermal transmittance of the building envelope

element	construction (warmside to coldside)	U-value [W/m^2K]
roof	water based color, wall paper, gypsum board, spacer battens, PE foil with plastered jointings to the walls and carefully glued and fixed overlaps, I-stud framing system (masonite beam) 445 mm insulated with blown in rockwool, formaldehydefree particle board plastic foil, turf roof.	0.10
wall	water based color, wall paper, gypsum plaster as airtight layer, calcium silicate blocks 175 mm, 275 mm expanded polystyrene foam (EPS), reinforced lime cement plastering.	0.14
ground floor	parquetry with solvent free coating, cement plaster, 40 mm EPS sound insulation, concrete slab, 250 mm PS insulation with reinforced coating.	0.13
window	krypton filled triple glazing with double low-e-coating, additional insulation by CO_2-expanded PU foam moulding of the wooden frames and the outer 25 mm of the glass perimeter.	0.70

The airtightness of the houses is very good, pressurization tests conforming /SS 021551/ yielded air change rates of 0.2 to 0.4 ac/h at 50 Pa pressure difference.

Tab. 2: Measured specific energy consumption

spec. energy use per m^2 heated floor area		period 91/92 [kWh/m^2a]	period 92/93 [kWh/m^2a]
household	electricity	6,27	6,17
ventilation	electricity	2,65	2,93
others	electricity	2,85	2,10
cooking	natural gas	2,43	2,60
hot water	natural gas	8,28	6,12
heating	natural gas	20,81	11,91
total		43,29	31,83

The energy balance is listed in Tab. 2. Period 91/92 covers Oct. 91 to Sep. 92, period 92/93 Oct. 92 to Sep. 93. The differences between the two periods are mainly caused by the fact that the insulation of the ground floor and the window frames were finished in May 1992.

Compared to the average of the German dwelling stock, the heating energy use was reduced to 1/20, the total energy use was reduced to 1/10.

4 Ventilation

4.1 Ventilation Strategy

The ventilation in a house has to guarantee acceptable indoor air quality and thermal comfort.

Uncontrollable in- and exfiltration caused by weatherdependent forces through leakages in the thermal envelope is not suitable for a demand controlled ventilation. Neither placement of the leaks nor the amount or direction of the air flow is controllable. Additionally, there is considerable risk for moisture damage and draughts.

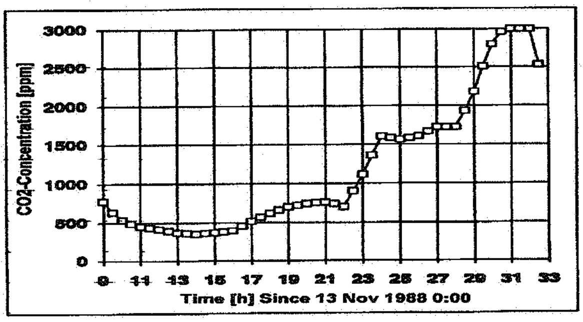

Fig. 2: CO_2 concentration in the bedroom of a modern German building without mechanical ventilation system, windows and doors closed. (Upper limit of range of measuring device 3000 ppm).

Natural ventilation of rooms by opening windows is the traditional and most popular way in Germany. There are no additional investment costs. But often it is difficult to open and close the windows regularly. A typical example for this is the situation in a bedroom with closed windows and doors. During night CO_2 concentration exceeds 3000 ppm (Fig 2), this is a indicator for bad indoor air quality. Continually opened windows result at wintertime in to high air volume flows together with dry air and increased ventilation losses.

A well established solution for a demand controlled ventilation sytem in low energy houses is a exhaust system (ES) /Blom 1990/, used in Sweden since more than 50 years. Air is extracted by fan from kitchens, bathrooms etc. Fresh air enters the house from outside by specially designed vents in living- and bedrooms, but also by other leakages in the building envelope. In an airtight house, a well planned and installed exhaust system this is a good tool to maintain acceptable indoor air quality. There is no heat recovery from exhaust air, but by demand control, it is possible to reduce air flow rates keeping a good air quality. This also results in reduced ventilation losses.

Supply-exhaust-systems (SES) with heat recovery by an air to air heat exchanger can reduce ventilation losses furthermore. There are increased investment costs and a higher electricity consumption compared to exhaust systems. Practical experience in low energy houses shows that installation of SES is only useful, if the following requirements are fulfilled:

- Efficient air exchangers (>75% heat recovery effectiveness) and low electricity consumption, so that the ratio between savings of heating energy and electricity consumption is better than 4.
- Buildings have to be very airtight. Otherwise significant heat losses are caused by additional infiltration, wich are not reduced by heat recovery.

- Proper maintenance of system and filters is undispensible, only clean systems guarantee good indoor air quality.

Nowadays an economical operation of SES in residential buildings in Germany is an exceptional case. Therefore, we usually recommend exhaust systems for normal German low energy houses, wich are simpler and more cost effective.

4.2 Mechanical Ventilation System

For an advanced low energy house like the Passivhaus a highly efficent heat recovery system is undispensible.

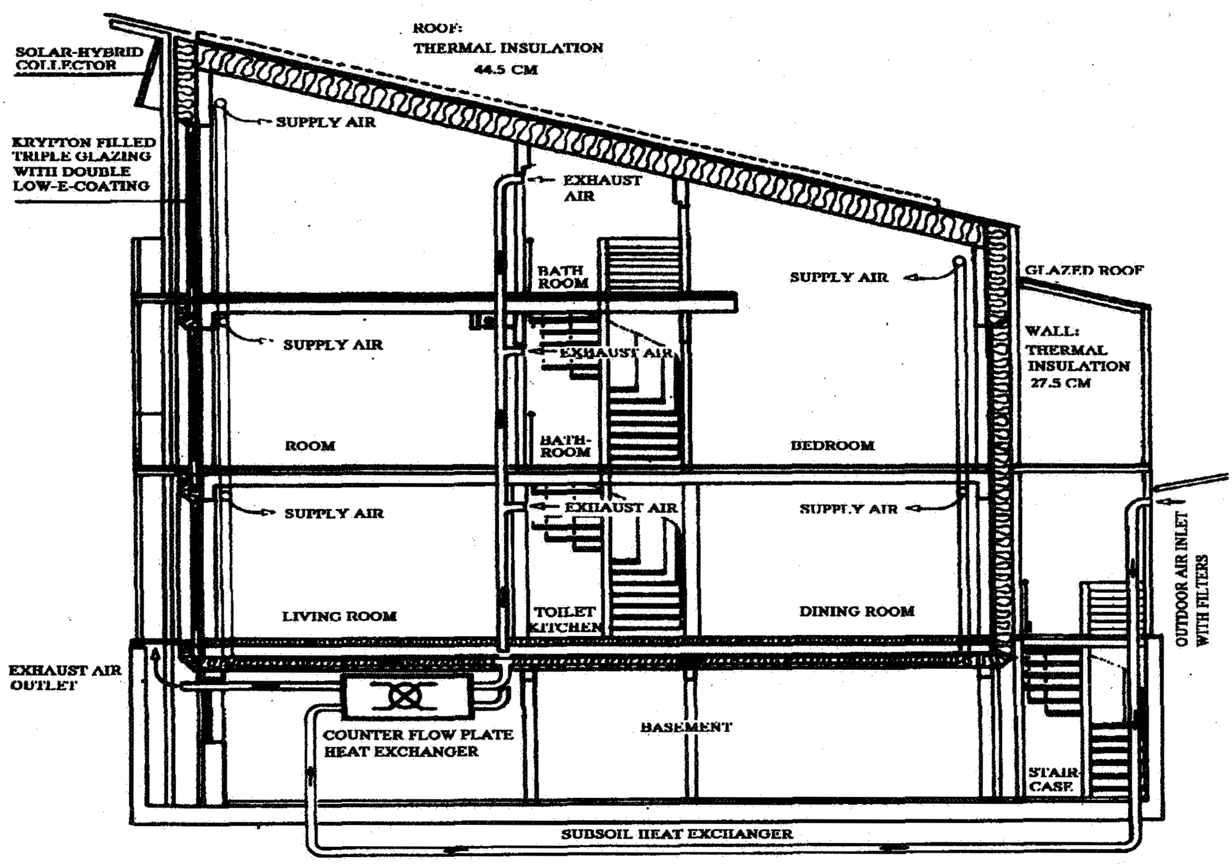

Fig. 3: Integration of the ventilation System in the Passivhaus

Each unit is equipped with a VAV ventilation system. Fig. 3 shows a vertikal sectional view with the arrangement of the ventilation system. Fresh air enters the system by a filter combination (EU3/EU8) at the north front and gets via the subsoil heat exchanger 1 m below the basement floor to a counterflow heat exchanger situated in the basement. At the entrance of the fresh air there is an additional test filter. Supply air ist distributed to the living area, exhaust air is taken from toilets, bathroom, and kitchen and drawn to the heat exchanger. The DC-driven fans are placed in the cabinet of the heat exchanger.

The supply air distribution can be varied (30%/70%) between the living area in the ground floor and the rooms in the upper floors, the exhaust air distribution as well between the rooms in the ground floor and the upper floors. This is done by motor controlled dampers. The air flow rate can be switched to 4 levels (0, 60, 100, 180 m^3/h). Both can be done manually or by a computerized control system. In the automatic mode the air flow rate and the supply air distribution is controlled by CO_2 concentrations in the ground or upper floors, the distribution of exhaust air is controlled by relative humidities.

Both in manual and in automatic mode the inhabitants can change the exhaust air distribution by local demand switches, realeased automatically after a period of time. During cooking, the air flow rate of the cooker hood is boosted by an additional AC-driven fan.

The air flow rates are designed to reduce and control the unavoidable emissions (odour, humidity and CO_2), related to human activity in residential buildings. Other emissions from building materials, furnitures, household products etc. should be minimized by proper selection of materials.

Taking Pettenkofer's number of 1000 ppm CO_2 /Pett 1858/as an upper limit for human caused loads, a minimal air flow rate of 25 m^3/h per person is necessary. An air flow rate of 100 m^3/h is sufficient for 4 persons. The total air exchange rate in the Passivhaus (0.22 ac/h) seems to be rather low. However, using the dampers to adjust air distribution on demand, this results in higher air exchange rates (0.5 ac/h and higher) in single rooms during presence of persons. The controlled interzonal airflow from the living area to bathroom, toilet and kitchen is also sufficient to remove humidity and odours.

Ventilation system and air flow rates were correctly installed and adjusted. Air ducts consist mainly of metal sheet coated with zink. The glasswool in the flexible sound attenuators is covered by perforated metal sheet. Duct joints are fixed by screws and taped thorougly, airtightness of ducts inside the thermal envelope was checked by pressurization.

The subsoil heat exchanger consists of flexible, corrugated synthetic pipes without joints under the ground plate. An inspection by video camera in october 1992, after one year of operation, showed all pipes clean and undamaged. At that time, no condensated water was present, although during some periods in spring and summer the presence of condensated water is suspected.

4.3 Measured Energy Results

Fig. 4 shows the measured air flow rates in one ventilation system of the Passivhaus during a typical day, 23 October 1993. It was a cloudy weekend day and all 4 inhabitants were at home. During nighttime the air flow amounts constant to about 100 m^3/h, during daytime air flows changes according to the activity of the inhabitants.

Fig. 5 shows the measured heat recovery effectiveness of the ventilation system. The mean effectiveness of the complete system at this day amounted to 94 %.

The annual electricity consumption of the complete ventilation system (fans, measuring and control system, motor drives), measured by a separate electricity meter, amounted to 457 kWh. Such a low consumption could only be achieved by changing the fans from AC to DC driven motors.

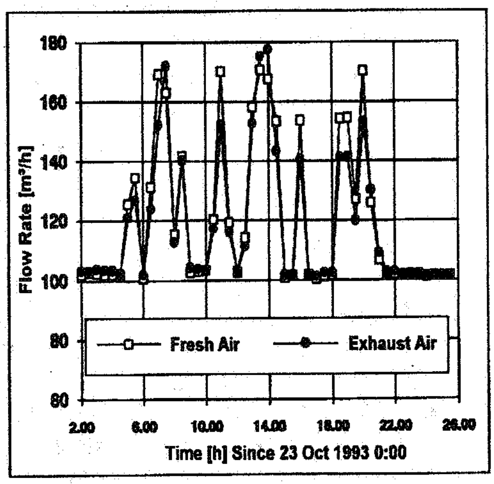

Fig. 4: Measured air flow rates in one ventilation system of the Passivhaus

5. Indoor Air Quality: Results and assessment

The primary purpose of ventilation is good indoor air quality. The Passivhaus is a very good test object to check the correlation between the operation of the system and the indoor air quality: detailed planning, thorough construction and verification by measurements of the physical properties of the existing building and ventilation system amounted in the result of a system complete in working order.

In 1993 the monitoring program therefore was extended to check the indoor air quality too. In addition to the continuos measurement of the energy balance, comprising also CO_2 and relative humidity, the levels of different VOC, dust, spores and radon were tested at different times. The air quality measurements were performed by "eco Umweltlabor Köln" under contract of IWU and supported ba a grant of HMUEB.

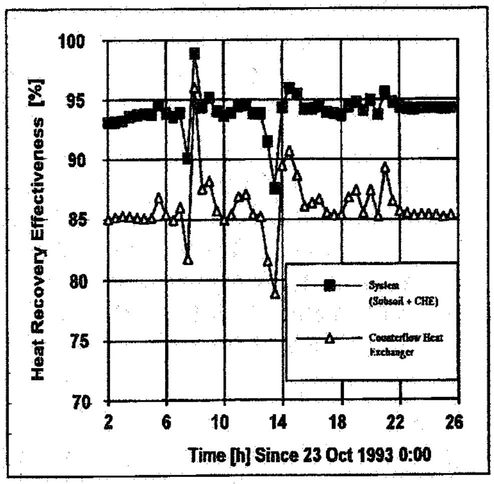

Fig. 5: Measured heat recovery effectiveness of the ventilation system (CHE: Counterflow Heat Exchanger).

5.1 Result: CO_2 Concentration

The level of CO_2 concentration is a good indicator for man made emissions in indoor air of residential buildings. CO_2 levels are continually measured for the living area in the ground floors and the living rooms in the upper floors, respectively. The sensors of infrared absorption type are calibrated regularly by test gas mixtures, the error of results amounts to about $\pm$ 30 ppm.

The amount and the distribution of air flows are sufficent to keep the CO_2 levels below 1000 ppm. A comparison with Fig. 2 shows that the indoor air quality, especially in bedrooms in the upper floors, is much better with continuos ventilation by a mechanical system.

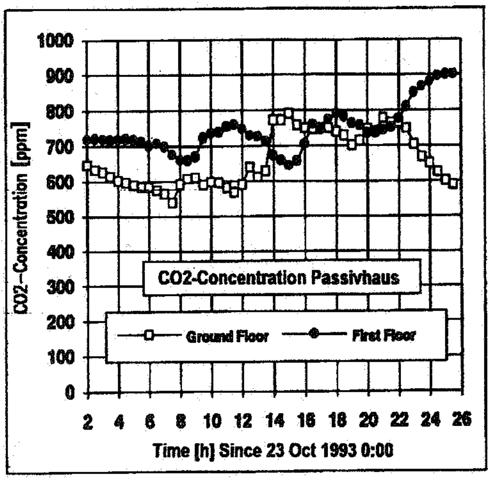

Fig. 6: Measured CO_2 concentrations.

5.2 Results: Relative Humidity

At room air temperatures between 18 and 23°C, wich are typical for the Passivhouse during wintertime, the level of relative humidity is imperceptible for humans. Nevertheless, it is a crucial parameter for indoor air quality.

- In dry air (< 35 %r.h.) it is more likely that dust swirls up and electrostatic charging occurs.

- Relative humidity > 60% improves the living conditions for mites, one of the most freqent occuring riscs for allergic reactions
- Higher humidity levels can lead to growth of mould (>65 %r.h. for some aspergillus species, > 80% for cladosporium herbarum and penicillium species). This is not only a risk for the building structure, it is also an allergenic risk for the inhabitants and may also be the direct caus of infections.

Especially continually high relative humidities are hazardeous: "Keep the houses dry" /Kronvall 1988/ was one of the most important conclusions of the congress "Healthy Buildings" 1988 in Stockholm.

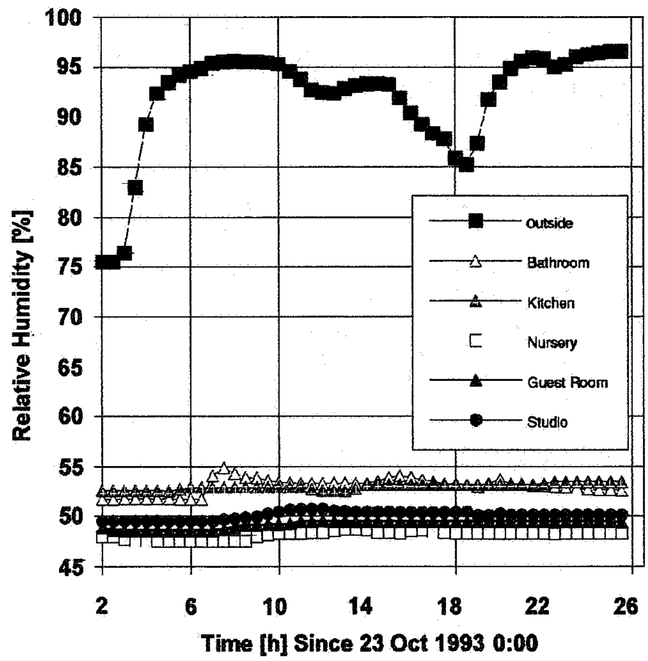

Fig. 7: Measured Levels of Relative Humidity in the Passivhaus".

Fig. 7 shows the course of relative humidities at 23 October 1993 (Saturday), a typical day for the heating season. At relative humidities between 75 and 98% in the outside air the relative humidities in the rooms with supply vents amounted to a nearly constant level of about 50%, in kitchen and bathrooms the levels stayed between 52 and 55%. A small rise, caused by showers, is visible in the bathroom in the morning. The removal of humidity takes a few hours, but the storage capacity by sorption of unsealed surfaces is sufficient to limit the peaks. On average surfaces and room air also in bathroom and kitchen stay dry. Relative humidities above 60% inside the Passivhaus were only noticed during muggy days in summer.

Considering the detected levels of relative humidities there is no condensation on any surface, even the perimeter of the window panes stay dry. Taking into account the diffusion properties of the thermal envelope there is also no condensation in any place of the building structure.

5.3 Results: Radon

The noble gas radon and its daughters are second to tobacco smoke the most frequent reason for indoor caused cancer. The WHO recommends /WHO 1987/ concentrations below 100 Bq/m^3 EER for new buildings. Each of the four units of the Passivhaus were tested 5 times by char coal dosimetry over a period of 3 days: The measured levels were all below 50 Bq/m^3. Even more meaningfull results are delivered by active dosimetry, done from 11. to 18. February 1994. The mean activity levels inside the living rooms were 15 to 25 Bq/m^3 EER. Compared to the average value (median) in the region of Darmstadt (45 Bq/m^3), the radon concentration in the Passivhaus is low. The existing small activity is due to exhalations from building materials, since the floor to the basement is almost airtight (demonstrated by pressurisation), infiltration is minimal (demonstrated by tracer gas measurements) and the supply air has a much lower activity.

5.4 Results: Volatile Organic Compounds

Each of the 4 units and outside air were tested on formaldehyde during 4 sessions.

Fig. 8 shows the course of the accumulated concentrations of chlorinated compounds, polynuclear aromatic hydrcarbons (PAH), alkane, terpene, formaldehyde and other compounds. The average of all 16 measurements of room air concentrations of these compounds, except of terpene, are far below 90% values of the /BGA/ study for german residential buildings.

In the beginning some hydrocarbon levels were found to exceed the BGA 90% values (styrene up to 30 $\mu g/m^3$, ethylbenzol up to 53 $\mu g/m^3$). For both compounds, there is a high probability to escape from the insulation material polystyrene on the outside of the walls. The concentrations never reached serious levels (/WHO/ guidelines believe a level of 800 $\mu g/m^3$ to cause no adverse health effects, the odour detection threshold amounts to 70 $\mu g/m^3$). The levels gradually decreased to the detection threshold of 3 $\mu g/m^3$.

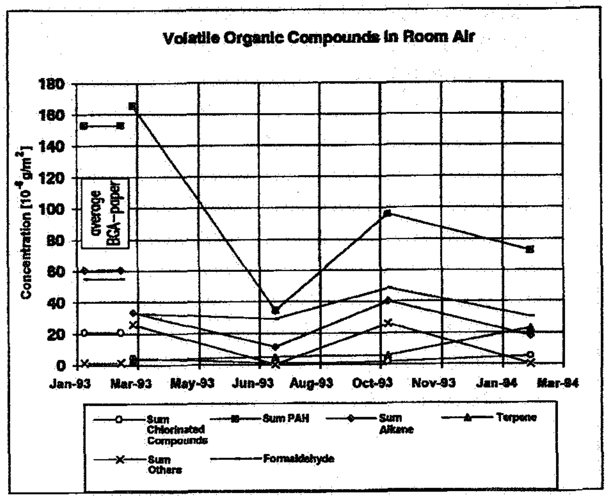

Fig. 8: Summed up Concentrations of Volatile Organic Compounds in Room Air.

The average formaldehyde level amounts to 34 $\mu g/m^3$ (90% BGA value 55 $\mu g/m^3$). Even the highest measured peak value is below the BGA threshold value (120 $\mu g/m^3$), but slightly higher than the WHO recommendation (60 $\mu g/m^3$).

5.5 Results: Suspended Dust

An analysis of mass and number distribution of particles with diameters between 0.5 and 100 μm was performed as well as a qualitative identification.

Concentrations of suspended dust in the rooms were in the range between 18 and 48 $\mu g/m^3$ not far away from the levels of outdoor air (16 to 51 $\mu g/m^3$) wich are all normal concentrations in residential buildings without specific pollution. To some extent there is a deposition in the filters of the ventilation system visible (inside about $4 \cdot 10^6$ particles, outside about $7 \cdot 10^6$ particles). The qualitative identification shows a ratio of organic to anorganic particles of 10 to 100. Manmade mineral fibres were only found at times and occasionally with a indoor maximum of 180 fibres/m^3 at outdoor maximum concentration of 510 fibres/m^3. These are commonly found concentrations, a significant contamination by fibres of the rockwool insulation (roof) is not present. None of the measurements at supply air vents showed mineral fibres.

In the building, no asbestos containig material was used, nevertheless at 2 of 14 tests one asbestos fibre per sample was detected (calculated concentration 50 fibres/m^3). These fibres presumably come from the brakes of the trains at the nearby railway lines.

Conspiciously were pillow shaped particles (2000 to 6000 particles/m^3) with diameters of about 10 μm wich stem from PE-Filters in the heat recovery unit. These filters were replaced by types without particle emission.

5.6 Results: Microbiological Tests

Under everyday conditions a 2 level Andersen-Impaktor was used to test the concentration of airborne germs and spores, the volume of sampled air was 100 to 200 l. The tests were performed 6 times in 2 to 4 of the units. The samples were taken from the living rooms, outside air, and partially also at supply air vents.

5.6.1 Spores

Outdoor concentrations show the expected variation in correlation with the seasons: a maximum during summer (July 1993 2988 KBE/m^3) and lower values in the range of 20 and 630 KBE/m^3 during the other seasons, average concentration amounts to 646 KBE/m^3. 20 of 24 samples of indoor concentrations show very low values of 150 KBE/m^3 or less, higher values occur during summer (with opened windows) at two of the samples (320 and 521 KBE/m^3) and two values from 20 January 1993 (360 and 453 KBE/m^3). The reason for the last two values were afterward identified in mildew covered organic waste and fruits. The average concentration of spores over all 24 samples amounts to 115 KBE/m^3, this is about 82 % below the outdoor concentration. Supply air concentrations normally are below the identification threshold of 5 KBE/m^3 with a maximum of 20 KBE/m^3. Samples of the test filter in the heat recovery unit showed spores only occasionally (maximum concentration 370 KBE/mg mass of the filter); this result agrees with the very low contamination of supply air.

5.6.2 Germs

Germ concentration varied during seasons and over the units between values below identification threshold and a maximum of 190 KBE/m^3, outdoor concentrations varied from 30 to 179 KBE/m^3. The contamination level of indoor air strongly dependends on the behaviour of the inhabitants. The measured concentrations in the Passivhaus can be classified as completly harmless.

5.7 Assessment of Indoor Air Quality

For the assessmment of the measurement results, sketched in chapters 5.1 to 5.6, a team of 5 experts was set up. The results of 5 reports and 3 expert meetings may be summarized as follows (proceedings will be published by IWU later this year):

- Repeated air quality mesurements in the Passivhaus showed comparatively low contamination. The building and ventilation concept of the Passivhaus meets the requirements of good indoor air quality.
- Measured CO_2 concentrations show that the ventilation system is able to control the existing indoor air emissions.
- The humidity level of indoor air stays within the recommended range. Due to the excellent insulation, there is no risk of condensation.
- Radon activity inside the Passsivhaus is very low.
- Concentration level of styrene started at slightly increased values of 30 μg/m^3 and decreased over the time to values about 3 μg/m^3. This can be considered as completly harmless.
- Different assessments were given to the measured formaldehyde levels. The average concentration of 34 μg/m^3 is smaller then the recommended /WHO/ limit of 60 μg/m^3, but one sample is slightly higher (70 μg/m^3).

- Qualitative and quantitative dust measurement results showed no significant contamination, especially of the type of man made mineral or asbestos fibres. The source of the pillow shaped PE particles could be removed by changing the test filter material.
- The measured concentrations of spores of 20 of 24 samples are very low (< 150 KBE/m^3). The four slightly higher values (up to 531 KBE/m^3) are caused by behavior of the inhabitants. The levels of germ concentration is harmless.
- Further microbiological measurements are planned to check, wether or not there is mould grow in the subsoil heat exchanger during warm and humid periods.

6. Conclusions

- All experts agree, that the indoor air contamination is low compared to other investigations. Therefore the ventilation concept proved to be sound.
- The measurements have shown that an average air flow rate of about 25 m^3/h per person fulfills the hygienic requirements for residential buildings, if the following conditions are true: defined interzonal air flow directions, possibilities for user adjustment of supply air distribution, and doubling of the air flow rate on demand; low emission concept for furnishing, building materials, household products etc.
- Efficient heat recovery systems (effectiveness about 80%) with low electricity demand are available but not yet standard on the market.
- The contribution of the subsoil heat exchanger to the reduction of ventilation losses (38%) is better than calculated during planning. Wether or not there is a risk of mould groth during warm periods in that heat exchanger has to be checked by further investigations.
- The automatic air quality control of the ventilation system works well, however, in combination with highly efficient heat recovery systems in residential buildings the reduction of ventilation losses is small compared to the expenditure.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Improvement of Domestic Ventilation Systems

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SYNOPSIS

The aim of the study was to identify methods for the renovation of ventilation systems in domestic buildings which are 3 - 8 storeys high. Three typical buildings were selected and the problems in ventilation were examined. The designers made their proposals for repairs and the research team analyzed the solutions and made improvements. The special problems compared with new buildings included less airtight building envelopes and leakages in existing ventilation ducts.

An analysis was performed, using a multi-zone airflow model, for the whole year and therefore the ventilation heat loss could be found in each case. As anticipated, the airflow rate of passive stack ventilation was too high in winter and too low in summer, but the system can be improved by means of controlled air inlets and outlets. A mechanical extraction system can be improved with demand-controlled ventilation instead of time control. The installation of heat recovery system requires improved sealing of the building envelope to minimize cross ventilation. The proposed systems will be tested and followed up later in experimental buildings.

1. INTRODUCTION

The renovation of existing high-rise residential buildings is becoming a major part of the construction work in Finland. At the same time the reduction of energy consumption is required to conserve the global environment. It has been estimated that 40% of residential heating energy is used for ventilation, which means that the renovation of ventilation systems may play a major role in decreasing the overall energy consumption in buildings.

There are two properties of the Finnish housing stock which have importance in the renovation of ventilation systems. Firstly, almost half (about 45%) of dwellings are in blocks of flats. Secondly, the buildings are fairly new: more than 70% of high-rise domestic buildings have been built since 1960. At that time mechanical ventilation was beginning to exceed natural ventilation in popularity.

The share of mechanical extraction ventilation in apartment blocks is currently more than 70%, leaving some 25% for natural ventilation. A typical mechanical system comprises an extractor fan on the roof of the building and a common vertical air duct shared by kitchens or bathrooms on all floors. The ductwork is much smaller and less expensive than in the natural ventilation system, where there is an individual extraction air duct from every kitchen and bathroom directly to the roof. The outdoor air usually enters through cracks in the windows or other components of the envelope. Purpose-built openings for incoming air, or air inlets, were taken into building practice since 1988 when the latest building code came into force. At the same time exhaust hoods in the kitchen became practically mandatory.

Most of the buildings built before 1980 already need repair or improvement. However, it is not known whether something should be done to the ventilation system at the same time. The main possibilities are to improve an existing passive stack ventilation system, change it for a mechanical extraction system, change it for a mechanical supply and extraction system, or to improve the existing mechanical extraction system. The improved systems should be more energy-efficient and they should satisfy the needs of the occupants: the kitchen ventilation should be better, airing should not be needed too often and draughts should be avoided.

The purpose of this project was to identify methods for the renovation of ventilation systems in domestic buildings which are 3 - 8 storeys high. Three typical buildings were selected and the problems in ventilation were examined. The designers made their proposals for repairs and the research team analyzed the solutions and made improvements.

2. COMPUTER SIMULATIONS

The analysis was performed mainly with using a multi zone airflow model Movecomp /2/, with which the building and the ventilation system could be described in detail. The computations were performed for a 4-storey building that currently has a passive stack ventilation system. The performance of the existing system, as well as possible improvements, were simulated.

The building has a basement and 3 inhabited floors. The length, width and height of the building are 75 m, 12 m and 14 m respectively. Most of the flats have only two walls facing the outside. Therefore, it was considered reasonable to compute only one 63 m^2 flat on each floor, as shown in figure 1. Each flat has three ventilation ducts of its own (flow area of each is about 280 cm^2) directly to the roof, but there are leakages between the ducts. The air leakages are set to measured values in the actual building, see table 1.

Table 1. Air leakages of the investigated building.

Leakage route	Air leakage at 50 Pa
Outer walls	105 L/s = 2.4 1/h
Apartment doors	1 L/s
Floors between apartments	19 L/s
Leakage between neighbouring air ducts	24 L/s
Two air inlets	50 L/s

The building is located in urban surroundings where the wind speed at the building height is assumed to be 72% of the velocity at the weather station. Some additional computations were performed for city and flat surroundings where the relative wind speed is 52% and 105% respectively. These proportions are in close agreement with the values given by the British Standards Institution /4/. The building is assumed to be exposed to the wind. The pressure coefficients for the 12 wind directions were taken from reference /1/.

The simulations were performed for a total of 182 weather conditions. The annual results were obtained using the probability of each weather condition at Helsinki-Vantaa airport, Finland, over 30 years. The annual mean outdoor air temperature 4.4°C, mean wind velocity 3.9 m/s and the design temperature for heating installations -26°C give a general idea of the weather in Helsinki. In each ventilation case the extraction air change rate was adjusted to 0.5 air changes per hour at a temperature of 4.4°C and zero wind speed.

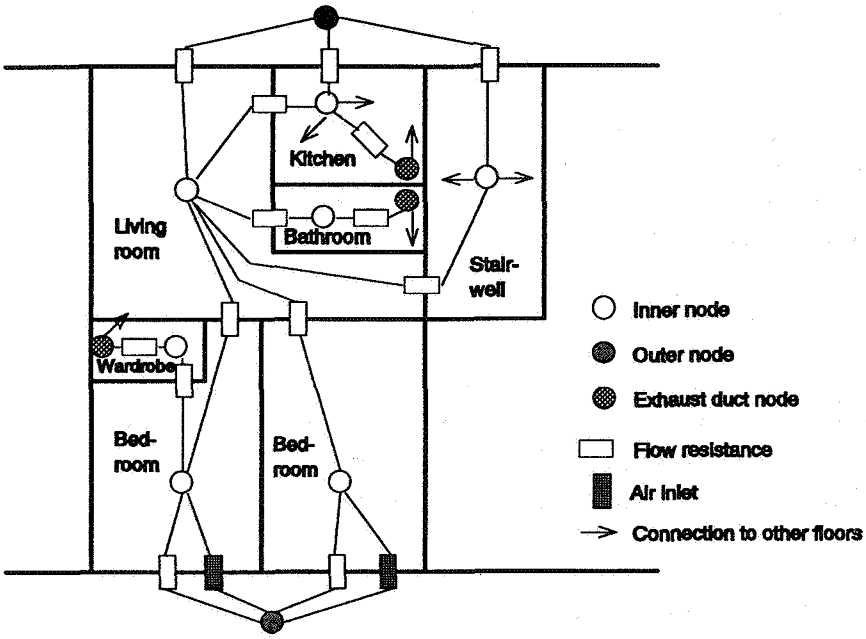


Figure 1. The flow network of one of the four floors. The floors are connected by leakages in floors, leakages between extraction air ducts and the stairwell. The total number of pressure nodes is 75.

3. NATURAL VENTILATION

The effect of control strategies for air inlets and outlets on the performance of natural ventilation is demonstrated in Figures 2, 3 and 4. Figure 2 shows the air change rates under different weather conditions. In Figure 3 the Helsinki weather is used to predict the annual stability of air change and, finally, Figure 4 shows the corresponding ventilation energies.

In the absence of wind the ventilation rate is lower than the target value of 0.5 1/h during warm weather and higher at cold weather. This is shown in Figures 2 and 3 where the upper left graph (no air inlets) represents the current situation in the building. It means that the occupants must increase ventilation during mild weather by opening windows. This will also correctly increase extraction airflow rates because the permeability of the building envelope controls the ventilation. The extraction air outlets are almost fully open and about 90% of the total available pressure difference is lost in the walls. During very cold weather the occupants should be able to control the air outlets to avoid excessive ventilation, draught and increased energy consumption (Figure 4). The need to regulate the extract air outlets is also especially important on lower floors and in more leaky buildings than this one.

The case with air inlets (upper right graphs in Figures 2 and 3) represents a situation in which the power to control the air change has been transferred to air outlets: they have 70% of the total pressure loss. The insensitivity to temperature variation is slightly better but the ventilation is much more wind-dependent because of cross ventilation. The high pressure loss in the extraction air terminals will diminish the risk of back flow through the extract air duct, which is a common problem during the winter months and a very difficult one to repair.

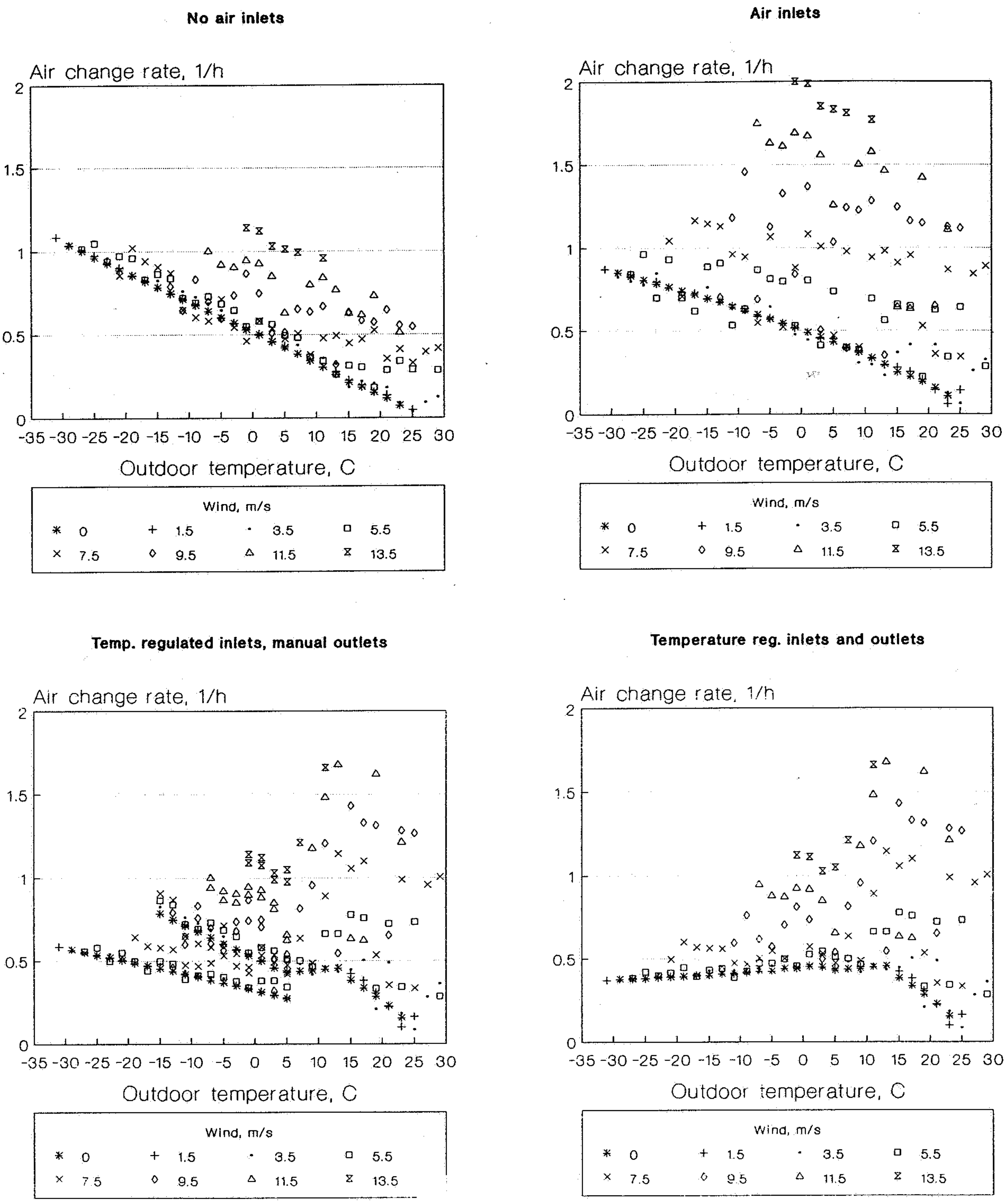


Figure 2. Outdoor air change rate on the second floor of the three inhabited floors in a naturally ventilated building in different weather conditions. The four cases represent different control strategies for air inlets and outlets. The extract airflow rate is adjusted to 0.5 1/h in each case at 4.4°C and zero wind velocity. The wind velocity at the height of the building is 72% of the indicated velocity at the weather station.

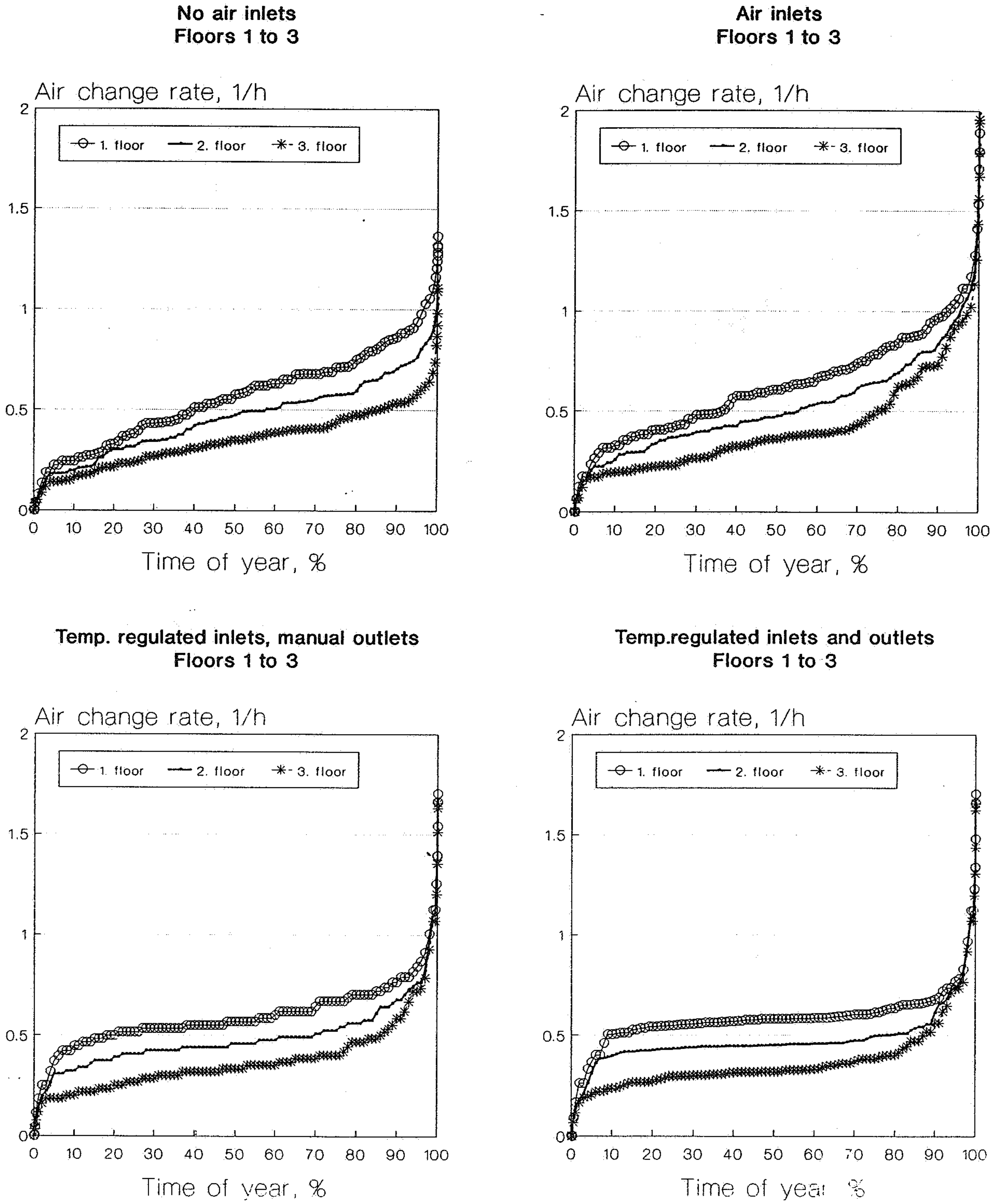
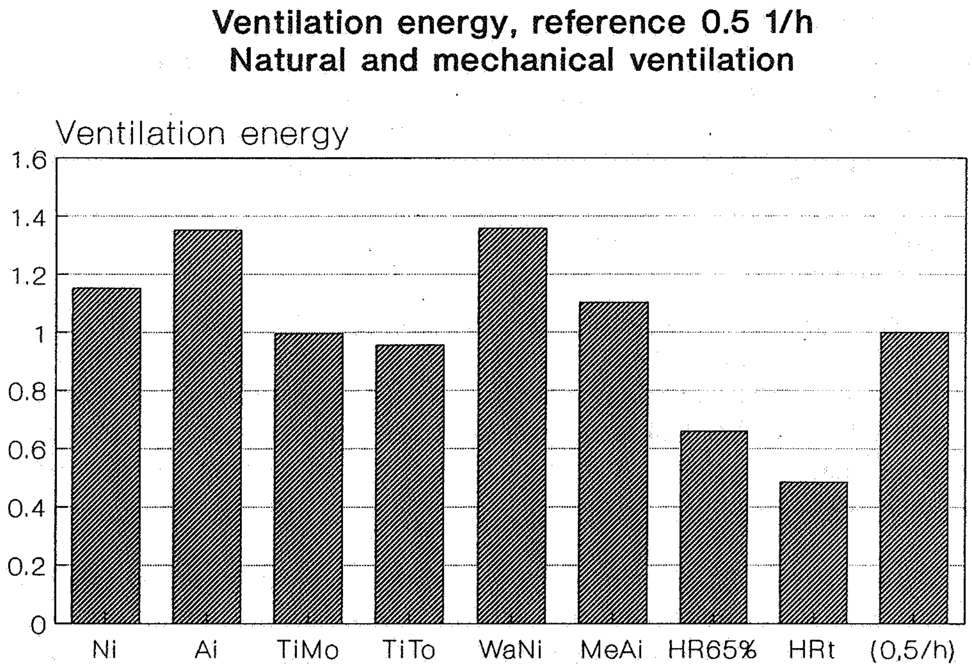


Figure 3. The annual stability of the outdoor air change rate on the three inhabited floors of a naturally ventilated building. The four cases represent different control strategies of air inlets and outlets. The extract airflow rate on the 2nd floor is adjusted to 0.5 1/h in each case at 4.4°C and zero wind velocity. The weather station is Helsinki-Vantaa and the building is in urban surroundings (wind velocity 72% of weather station velocity).



Legend:

Ni	No air inlets
Ai	Air inlets
TiMo	Temperature controlled air inlets, manually controlled air outlets
TiTo	Temperature controlled air inlets and outlets
WaNi	Wind assisted ventilation, no air inlets
MeAi	Mechanical exhaust, air inlets
HR65%	Heat recovery, efficiency 65%, supply air 0.4 1/h, exhaust 0.5 1/h
HRt	As HR65%, but building air leakage only 0.6 1/h at 50 Pa

Figure 4. Ventilation energy of different ventilation systems. The reference value on the right corresponds to a constant airflow rate of 0.5 1/h throughout the year. The heat balance of the building has been simplified by assuming that external energy is needed when the outdoor temperature is lower than 12°C.

A more sophisticated control strategy makes use of temperature controlled air inlets (lower left graphs in Figures 2 and 3). In this case the extract air outlets are switched manually to the winter position during three winter months (December to February). This case provides good ventilation stability (Figure 3) and low energy consumption (Figure 4).

Finally, the case in which the air inlets and outlets are temperature controlled shows the best performance in terms of annual stability (Figure 3, lower right graph) and ventilation energy.

The simulations show that there is sufficient driving force for natural ventilation in multi-storey residential buildings for most of the year. The system can work well if the inhabitants are willing and able to regulate the air inlets as well as the air outlets. But there are also several possibilities of automating the system. The first step in renovating the system could be the installation of two position air outlets which have been preadjusted separately for each floor in the building. Humidity control of airflows from bathrooms is also an interesting possibility. The next step is to install air inlets that can be controlled manually, according to the outdoor air temperature, or that maintain a constant airflow /3/.

4. MECHANICAL EXTRACTION VENTILATION

For better control of extraction airflow rates a fan can be installed on the top of natural ventilation shafts. The large air ducts from each kitchen and bathroom make it possible to have a demand controlled ventilation system. The only problem is the air leakage of old masonry ducts. Due to this leakage it is not possible to use the high pressures which are common in mechanical systems. A study is under way to examine the different possibilities of sealing old ducts.

Demand controlled ventilation should be the aim also in renovating existing mechanical extraction ventilation systems. The problem here is also connected with the existing air ducts that are shared by different floors in this system. Most recent ducts, in particular, are undersized so that the pressure level in the ducts must be very carefully selected.

5. MECHANICAL EXTRACTION AND SUPPLY

A balanced ventilation system has many good properties: the airflow rates to each habitable room are well controlled, noise and dust can be kept outside, and the heat recovery from the extracted air reduces energy consumption.

However, older buildings are often leaky. Balanced ventilation reduces the depressurization of the building and leads to increased infiltration and energy loss. Therefore the energy saving because of heat recovery was only 35% even if the temperature efficiency of the heat exchanger was assumed to be 65% (see Figure 4). The leakage can be reduced by improved sealing on the building. The energy saving will be 50% if the air leakage is reduced by a factor of 4 from the original 2.4 1/h at 50 pascals.

The corresponding annual air leakage can be seen in Figure 5 in which the building air permeability as well as terrain parameters are varied. An important additional parameter is the difference between air extraction and supply which is in this case 0.1 1/h. It will determine the level of building depressurization. It is interesting to note from Figure 5 that the additional air infiltration is directly proportional to the building air leakage at 50 Pa test pressure.

The possibilities of avoiding extensive new ductwork when introducing balanced ventilation will be studied later in the project. An interesting possibility is to use apartment-specific ventilation units and to install the intake and exhaust vents on the outer wall of each apartment. This was shown to be a reasonable solution considering the quality of the intake air /5/.

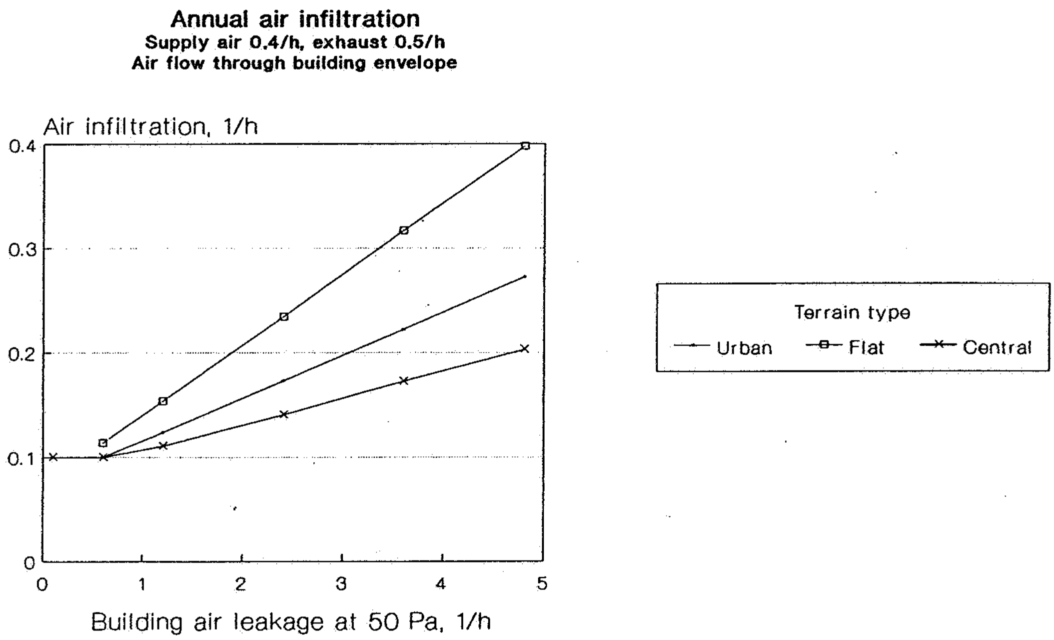


Figure 5. Annual energy based air infiltration in balanced ventilation. "Energy based" means that the infiltration has been computed from the annual infiltration energy loss.

ACKNOWLEDGEMENTS

This study is a part of the technology programme RAKET - Energy Use in Buildings, which is financed by the Finnish Ministry of Trade and Industry.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

The Capenhurst Ventilation Test House

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The Capenhurst Ventilation Test House

1.0 Synopsis

A Test House at EA Technology, Capenhurst, has been refurbished to provide a ventilation test facility. The house was required to meet the following requirements:

- A high standard of air tightness
- Insulation to current Building Regulations or better
- Incorporation of several ventilation systems
- Comprehensive instrumentation

The original timber frame front and rear facades of the house were replaced with brick and block construction. All internal floors, ceilings and partitions were replaced and the external walls replastered. Attention was paid to the sealing of junctions between building elements. The result was a house with the low infiltration of $L_{50} = 2.62$ air changes per hour at 50 Pa pressure difference. Mechanical Ventilation with Heat Recovery, Passive Stack Ventilation and extract fans have been fitted. Temperature and humidity sensors are installed in all rooms and temperature and flow sensors are mounted in ventilation ducts. Sampling tubing is fitted to all rooms for tracer gas measurements.

This paper describes the precautions taken to ensure air tightness and illustrates the range of measurements that have been made.

2.0 The Capenhurst Test Houses

EA Technology operates a number of test houses for studies of space heating and ventilation. A matched pair of detached houses has recently been completed, which will enable sensitive comparisons to be made between the performance of different systems operating under identical conditions. Six further houses are in use on a site about 1 km from the main laboratory. The houses were built some 25 years ago and represented a range of constructions, from nine inch solid brick to a contemporary standard of good insulation. Since then, two houses have had major upgrades and are now well insulated, well appointed habitable test houses. A third house, which is the subject of this report, has been upgraded and re-equipped to provide an experimental facility designed for ventilation measurements. The house is a conventional $90m^2$ three bedroom semi-detached house with a space inserted between the two halves to allow access to the loft; the floor plan is shown in Figure 1. The house is designed as a versatile experimental facility. It is not a habitable house, but meets current Building Regulations except where there is an overriding experimental requirement.

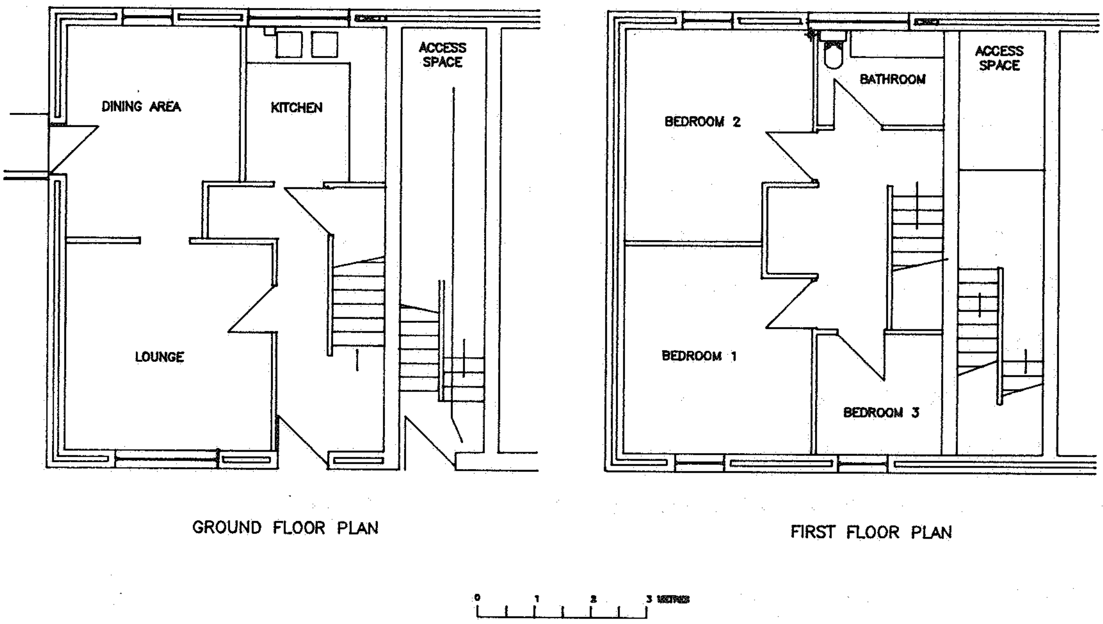


Figure 1. Floor plan of Ventilation Test House

3.0 Construction

3.1 Refurbishment

The pre-existing house was built with brick/cavity/block gable wall. Front and rear walls were lightweight timber frame construction. Renovation of the building involved:

- Removal of all internal fittings and services.
- Removal of internal partitions, internal floorboards and ceiling plasterboard.
- Removal of front and rear walls.
- Removal of plasterboard dry-lining on remaining walls.

The house was then refurbished. The order of construction is important. The aim is to avoid concealing potential leaks before they can be sealed or tested.

1. Build front and rear facades in cavity brick/block insulated with mineral fibre batts.
2. Fit new ground floor in tongue and groove chipboard; seal round edge.
3. Fit new upper storey ceiling plasterboard and seal.
4. Plaster inner blockwork of all walls, including between floor joists. The floor joists are already hung on hangers and do not penetrate the inner leaf.
5. Fit new doors and windows, sealing between frame and wall.
6. It is now possible to carry out a first pressure test. Locate and seal leaks.

7. Install plasterboard ceiling to ground floor.
8. Install internal partitions and skim all plaster work.
9. Fit coving to all wall/ceiling junctions and seal.
10. Seal all wall/floor joints with expanded foam.
11. Carry out a further pressure test and seal leaks as necessary.
12. Fit water and electrical services.
13. Pressure test and seal.
14. Fit ventilation systems.
15. Further pressure test.
16. The house is now ready for final decoration and carpet laying.

The pressure test at Stage 6 is valuable. It takes place with the outer shell of the house complete, all doors and windows fitted and sealed. The interior of the house is empty with virtually complete access to its outer envelope. It is therefore possible to gain access to any leaks and seal with a suitable expanding foam.

3.2 Air tightness

Air tightness is measured using the blower door technique. The pressure test provides a simple way of keeping a check on the air tightness of the house. Table 2 shows that the very low infiltration was maintained during construction and the installation of ventilation equipment. As the house began to dry out during spring and summer, the infiltration increased. The final figure of under 4 air changes per hour is still very much better than the Medallion 2000 standard of $L_{50} = 7$ used by the Electricity Industry as a condition for the installation of full house MVHR. The house will be resealed before commencing a new series of ventilation measurements during the next heating season.

Table 1 Air leakage measurements

Date	L_{50}	Comment
14/09/93	2.35	Outer shell of house complete
13/10/93	2.49	Construction complete
07/01/94	2.62	Ventilation systems installed
28/03/94	3.36	
13/06/94	3.75	After completion of measurements

The U-values of building elements have been calculated using conventional techniques. Table 2 gives a summary of the heat losses.

Table 2 Heat loss summary			
Element	Area	U value	AU
	m^2	$Wm^{-2}K^{-1}$	WK^{-1}
Gable wall	37.49	0.44	16.50
Front & rear walls	43.44	0.46	19.98
Windows	11.48	2.80	32.14
Doors	3.68	2.80	10.30
Ground floor	45.75	0.59	26.99
Roof	45.75	0.28	12.82
Fabric heat loss coefficient			118.7
Internal volume	198 m^3		
Ventilation loss @ 1 ac/h			66.0
Based on thermal dimensions for building elements and internal volume excluding partitions for volume.			

4.0 Equipment

4.1 Heating

The house is heated by direct acting panel heaters, fitted in all rooms except kitchen and bathroom. The integral electronic thermostats can maintain a close temperature control with fluctuations of the order of ± 0.5K. All heaters have are switched via a central time switch operating over a power line carrier system. This controller can be used to switch other equipment, using either hardwired receivers in power outlets or plug in adapters. A total of twelve channels is available, with a minimum switching period of one hour. Under sink water heaters are fitted in kitchen and bathroom. No hot or cold storage tank is fitted and there are no water pipes penetrating into the loft space. No bath is fitted.

4.2 Ventilation

Three independent ventilation systems are fitted in the house. Each may be sealed, to allow independent operation. Transfer grilles have been fitted to all internal doors to give a standard and reproducible leakage between rooms.

4.2.1 Mechanical ventilation

A full house Mechanical Ventilation and Heat Recovery (MVHR) system is fitted. Fresh air is supplied to lounge, dining room and all three bedrooms. Air is extracted from the bathroom and kitchen. The kitchen is fitted with a high level air extract in addition to the cooker hood; in a larger house the second extract would be positioned in a lavatory or utility room. The main unit containing fans and heat exchanger is mounted in the loft space; this allows for replacement or modification if desired. A silencer is fitted in the supply duct to conform with normal practice. The unit was chosen to be capable of providing at least one full air change per hour to the house over a range from 0.5 to 1.0 air changes per hour. All loft duct work is insulated, except for the intake duct, which draws air directly from the loft space. Ductwork inside the house is uninsulated and accessible, to allow for measurement or modifications.

4.2.2 Passive Stack Ventilation

At the time of installing the systems, Passive Stack Ventilation was under consideration for inclusion in the current revision to the Building Regulations for England and Wales. In the absence of the final specification, the PSV system was designed to represent good current practice. Two ducts are installed, extracting from kitchen and bathroom. The kitchen duct travels vertically through the house. In the loft, insulated flexible ducting takes the stacks to ridge mounted terminals for discharge. Trickle ventilators are fitted in all rooms and can be closed when necessary.

4.2.3 Extract fans

Two-speed extract fans are mounted through the wall in kitchen and bathroom and meet the current Building Regulations.

4.3 Instrumentation

Wall mounted temperature and humidity sensors are fitted in each room. Thermocouple sensors are fitted in the ductwork of the ventilation systems. Measuring bends are incorporated in the extract and intake ducts of the MVHR system and hot wire anemometers are installed in the PSV ducts. Measurement values are recorded using a data logger installed in the adjacent cavity.

Four sampling tubes run from the landing to each room in the house, permitting simultaneous sampling and dosing of two tracer gases. Provision is made for the external stowage of gas bottles to avoid any problems of leakage.

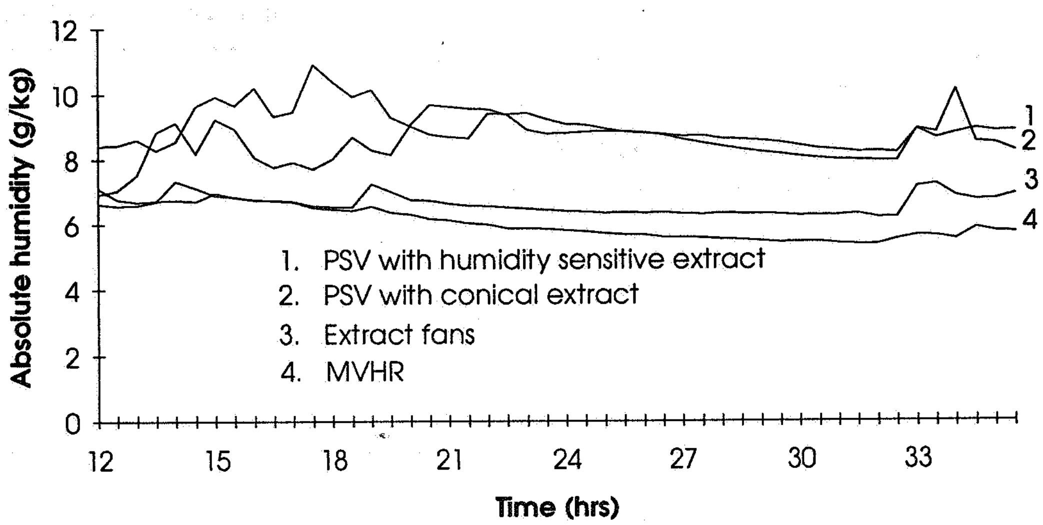
Wind speed and direction, together with insolation on a horizontal surface are measured at roof level on a nearby house. Ambient temperature and humidity are measured in a Stephenson screen in the rear garden of the test house.

5.0 Examples of Measurements

5.1 Humidity Measurements

The Vaisala sensors in each room measure temperature and relative humidity. From these values the logger calculates the absolute humidity in grams per kilogram of dry air. In Figure 2 the results from the kitchen are shown for a twenty-four hour period of operation of each ventilation system, from noon to noon the following day. In each case the kitchen door was closed and the Turmix humidifier was in operation from 12:00 to 14:00, 17:00 to 19:00 and 08:00 to 09:00 to simulate cooking. The moisture generation rate was approximately 0.6 litres per hour of operation in each case. The mean ventilation rates differed between the systems. For the PSV with conical extract the mean extract rate over the day was 12 m³/h - this is an atypically low value due to a low internal/external temperature difference that day (see Figure 3). The flow rate in the stack with humidity sensitive extract was 16 m³/h. The total extract rate of the MVHR system was 99 m³/h, resulting in extraction of approximately 66 m³/h from the kitchen. The extract fan was operated at the same time as the humidifier at its high speed of 225 m³/h and was otherwise off.

Figure 2. Humidity Levels in Kitchen



5.2 Stack flows

Figure 3 shows the average daily stack flows for both stacks with conical extract and kitchen stack with humidity sensitive extract. The flows in kitchen and bathroom stacks with conical extract seem to be about equal; it may therefore be assumed that the flow in the bathroom stack with humidity sensitive extract is about equal to the flow in the kitchen stack with humidity sensitive extract. Stack flows with conical extract increase with temperature difference. The humidity sensitive extract throttles the flow back at higher temperature differences; this would be expected since the ventilation requirement for humidity control reduces with colder drier incoming air.

5.3 Energy

Figure 4 shows the gross energy input to the house each day as a function of the internal/external temperature difference. The energy input comprises all electrical energy including heating and an estimate of solar gain. The results have not been corrected for ventilation rate. Measurements are available of power consumption of individual circuits including power used by ventilation systems and will be subjected to detailed analysis.

Figure 3. Average daily stack flows

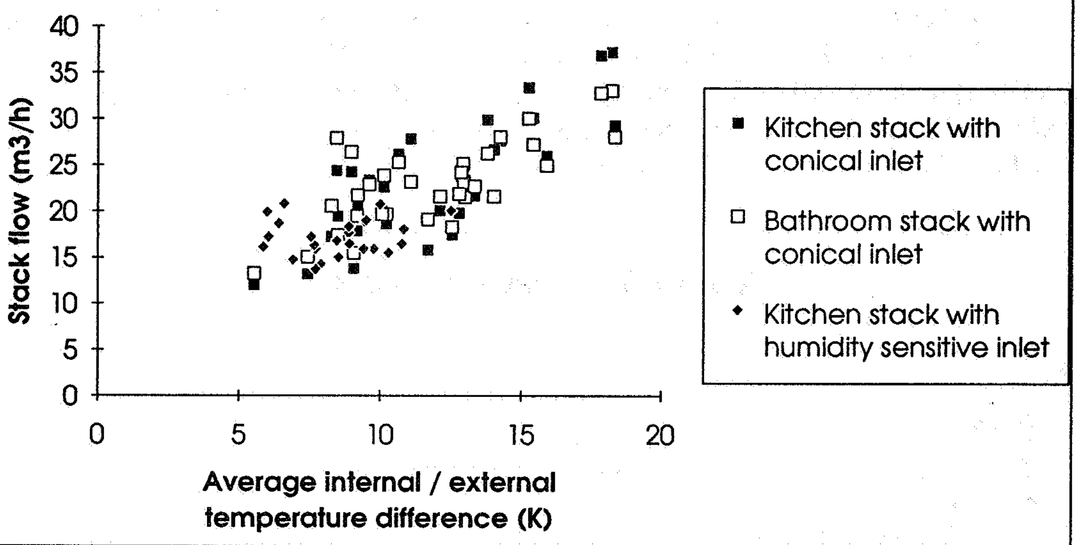
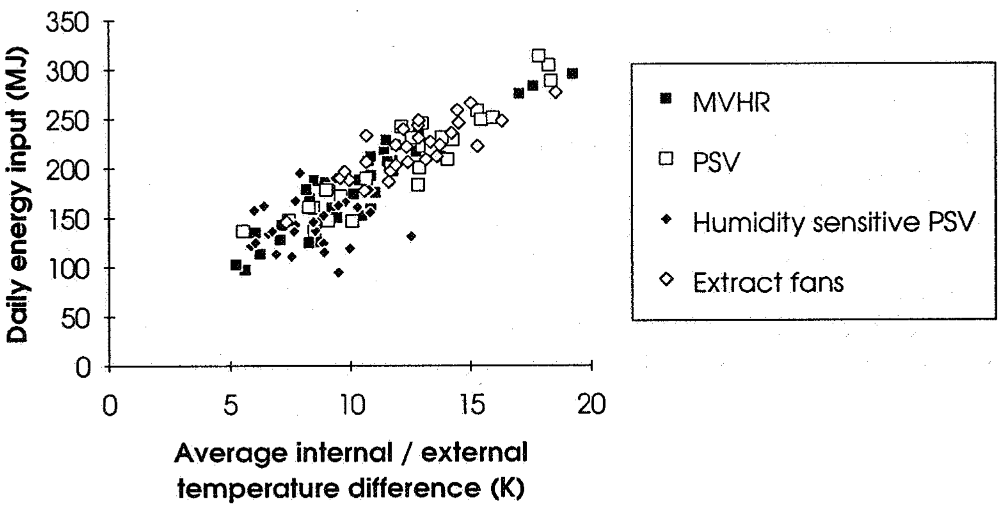


Figure 4. Gross Energy Input



6.0 Discussion

The refurbishment of the test house has produced a test facility enabling practical investigation of the performance of different ventilation systems to be carried out. It is possible to make room by room measurements to investigate the movement of water vapour or other contaminants through the house, as well as measurements of overall ventilation. The house is thus well suited to investigation of Indoor Air Quality as well as the energy performance of ventilation systems.

7.0 Acknowledgements

The measurements carried out in the test house are part of EA Technology's core research programme, which is funded by:

Eastern Electricity plc
East Midlands Electricity plc
London Electricity plc
MANWEB plc
Midlands Electricity plc
National Power plc
Northern Electric plc
NORWEB plc
PowerGen plc
Scottish Hydro-Electric plc
Scottish Power plc
SEEBOARD plc
Southern Electric plc
South Wales Electricity plc
South Western Electricity plc
Yorkshire Electricity Group plc

S L Palin is a member of the Post Graduate Training Partnership, which is funded by the EPSRC and the DTI.

The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Effective Ventilation Strategies Demands Flexible System Design

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Effective ventilation strategies demand flexible system design

Synopsis

User experiences of the workings of a ventilation system have often been pretty disheartening. Draughty, too hot, noisy, too stuffy are some of the verdicts which in many cases have been confirmed by objective measurements.

Often the complaints are due to the air flows not being appropriate to the room. This in turn can be due to adjustment difficulties or to the flow balance in different branches of the system being affected by residents tampering with the supply or exhaust air terminal device settings.

In order to avoid these problems Stifab has developed a product - a self-acting pressure regulator- which makes it possible to design more flexible systems.

The pressure regulator makes it possible for a constant pressure to be maintained in the different branches of the ventilating system. This is practically independent of changes or influences elsewhere in the system.

To guarantee perfect performance throughout the useful life of the ventilating system, the regulators are stationed at appropriate points in the distribution system.

The paper describes examples of different system design in order to get:

1. a flexible demand controlled ventilating system
2. guaranteed air flow balance between supply and exhaust

General aspects of a ventilation system

When designing a ventilation system, care must be taken to ensure that it meets certain fundamental requirements which can never be neglected.

The following aspects are important in this connection:

1. Simple reliable systems
2. Systems which are easy to maintain
3. Systems which are easy to adjust
4. System with in-built inspection facilities
5. Systems with in-built flexibility of capacity (air volumes and cooling and heating capacity)

6. Systems permitting flexibility with regard to flow pattern and air velocity in the occupied zone
7. Systems which are energy-efficient
8. Systems which can maintain a good standard of comfort
9. Systems ensuring that the entire occupied zone will be ventilated, i.e. without any stagnant zones
10. Systems which are stable in relation to both external and internal disturbances
11. Systems which avoid short circuits between supply and exhaust air

Generally speaking, the system has to be designed in such a way as to help simplify the operation, care and maintenance of the installation. Products requiring service must be easy to reach and dismantle.

It is also pretty obvious that if a tenant "sabotages" his own ventilation system, he alone must suffer for it.The other residents in the building must not be affected by this disruption. Unfortunately, disturbances in the form of deliberately blocked exhaust and supply air terminal devices have been only too common.

Fixed measuring points in the system for checking air flows are an obvious requirement.

It is widely testified that an installation will be much more favourably regarded if the user can alter the setting without difficulty. In housing, for example , this flexibility makes it very easy for residents to adapt their air flows to actual requirements.

The ability to control air flows according to the needs of the different rooms is something we have not been over-endowed with where traditional mechanical supply and exhaust ventilation systems are concerned.Instead the aim has been to keep air flows as constant as possible. It is clearly an advantage if residents, within reasonable limits, can adapt the air flow to the actual needs of individual rooms. This must of course be achieved without having to reduce air flows in other rooms. One must always be able to guarantee minimum air flows in the various room units.

Flexibility

We began by mentioning the need for flexibility, and the main focus of development during the nineties is certain to be on questions of this kind.

As individuals making different demands on our climatic system, we will greatly appreciate the installations which we are able to adjust for ourselves. So is there an easy way of making our systems flexible?

One technically simple and good solution is to maintain constant static pressure at strategic points in a distribution network by means of constant pressure regulators CPR, see Fig. 1. In this way there are very good possibilities of varying the air flow downstream of the constant pressure regulator with a simple type of damper. The damper can be governed in various ways. Manual control is preferable to an arrangement where the damper, or the supply and exhaust air terminal devices, are connected to a timer.

At this point perhaps someone will object that this is a way of making systems even more complicated and vulnerable. This is true insofar as one is thrown back on electronics for governing the constant pressure regulators. But simple mechanical regulators are already commercially available. Pressures from a few Pascal up to 200 Pa can be kept constant. The pressure is very easy to set, using a device with a Pa scale.

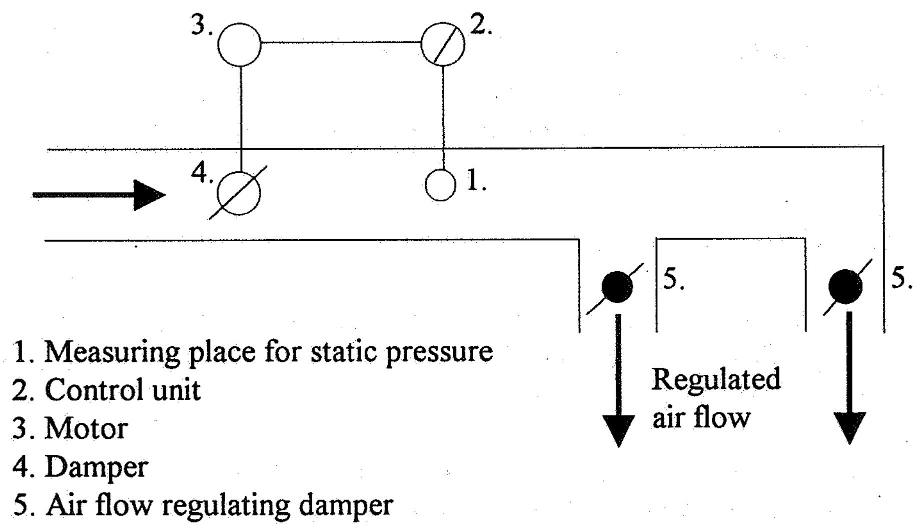


Fig. 1 Pressure regulator and damper system for easy regulation of air flow. Units 1, 2, 3 and 4 together make up the pressure regulator (CPR).

Some applications of the constant pressure system

The constant pressure technique can be used in ventilation installations of every kind. For comfort ventilation, it can be used in:

❐ CAV systems (Constant Air Volume) for obtaining;

- correct and constant air flows
- easier commissioning
- in-built flexibility because, if necessary, the flow can easily be adjusted after commissioning

- VAV systems (Variable Air Volume) for;

    - simple arrangement, since one pressure regulator can be used for all devices on a branch duct
    - easier commissioning
    - lower maintenance costs, thanks to the simpler system

- DCV systems (Demand Controlled Ventilating) to provide;

    - a simple arrangement for demand control of air flows because the constant pressure regulator (CPR) permits manual operation of dampers / terminal devices
    - easier commissioning
    - CPR facilitates future extension of the system

In all systems supply and exhaust air can be controlled in such a way that the two air flows will always be balanced.

Planning

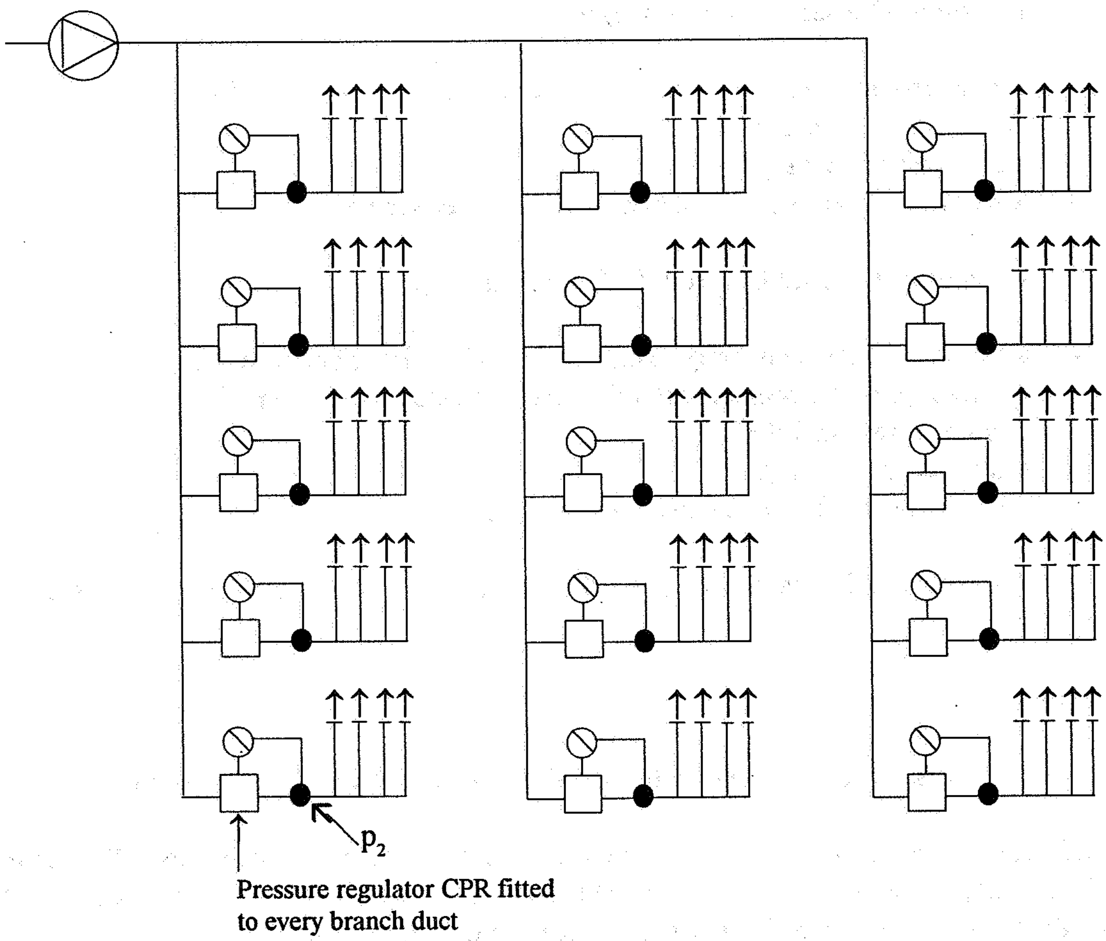
This constant pressure system is easier to plan than traditional ventilation systems.

It is sufficient to check that the static pressure in the branch duct furthest away meets the minimum requirement as per the catalogue data. One is then also guaranteed that the static pressure in other branch ducts will be sufficient for driving the other pressure regulators.

Fig. 2 shows an example in which the pressure regulator is used in a CAV system. One pressure regulator is fitted to every branch duct.

The purpose of the pressure regulator here is to maintain constant air flows in the system. The flows are kept unchanged even if, say, pressure losses across the filters are increased by fouling or if there are thermal driving forces at work in the system.

This means that the system does not have to be adjusted to flows exceeding the nominal ones with a clean filter.



Principle: The pressure p_2 is kept constant in every branch duct. The flow from the terminal devices will thus be kept constant, regardless of any disturbances upstream in the distribution network.

Fig. 2 The pressure regulator used in a CAV system

Constant pressure regulators (CPR) used in a DCV system

All systems in which demand control of air is desirable can be classified as DCV systems.

A system which is simple and therefore very attractive and in which the balance between supply and exhaust air flows does not present any problem is when the pressure difference between outdoors and indoors is used as reference.

Various installation options will be described here.

Once it has been decided to plan a DCV system, one has already decided to give priority to flexibility. This can be imparted to either the supply or the exhaust air side.

The following basic rules should apply:

- The supply air is used for ventilation only. That is, mainly isothermal air is supplied. In these instances, priority can be given to either the supply or the exhaust air side. In domestic installations, for example, it may be appropriate to give priority to the exhaust air side, because certain minimum exhaust air flows are stipulated for various spaces.

- The supply air is used as a heat carrier. That is, the ventilation air is also used for heating and/or cooling. In cases of this kind, the supply air side must be given priority and, accordingly, the requisite flexibility.

Whichever side receives priority, the control unit for the non-priority side must be fitted in such a way that a continuous negative pressure of about 3 Pa can be maintained indoors.

Example: Domestic ventilation
Adjustable exhaust air terminal devices are installed in at least all sanitory cubicles in the dwelling unit. The supply air is delivered to all bedrooms and living rooms through a constant pressure system. See Fig. 3.

All supply air terminal devices must be adjustable. The pressure regulator on the supply air side maintains constant pressure in the dwelling.

Two alternative installations are possible:

- Positioning of the control unit in the dwelling and the pressure nozzle in the outdoor air.
  The desired value of the control unit is set to about + 3 Pa

- Positioning of the control unit where it senses the atmospheric pressure and the pressure nozzle at a represantative point in the dwelling.
  The desired value of the control unit is set about - 3 Pa.

With these solutions the supply air flow is throttled down to a minimum in the event, for example, of a window being opened.

With this principles, the supply air flow can easily be re-allocated with the adjustable supply air terminal devices, without any effect on the total flow.

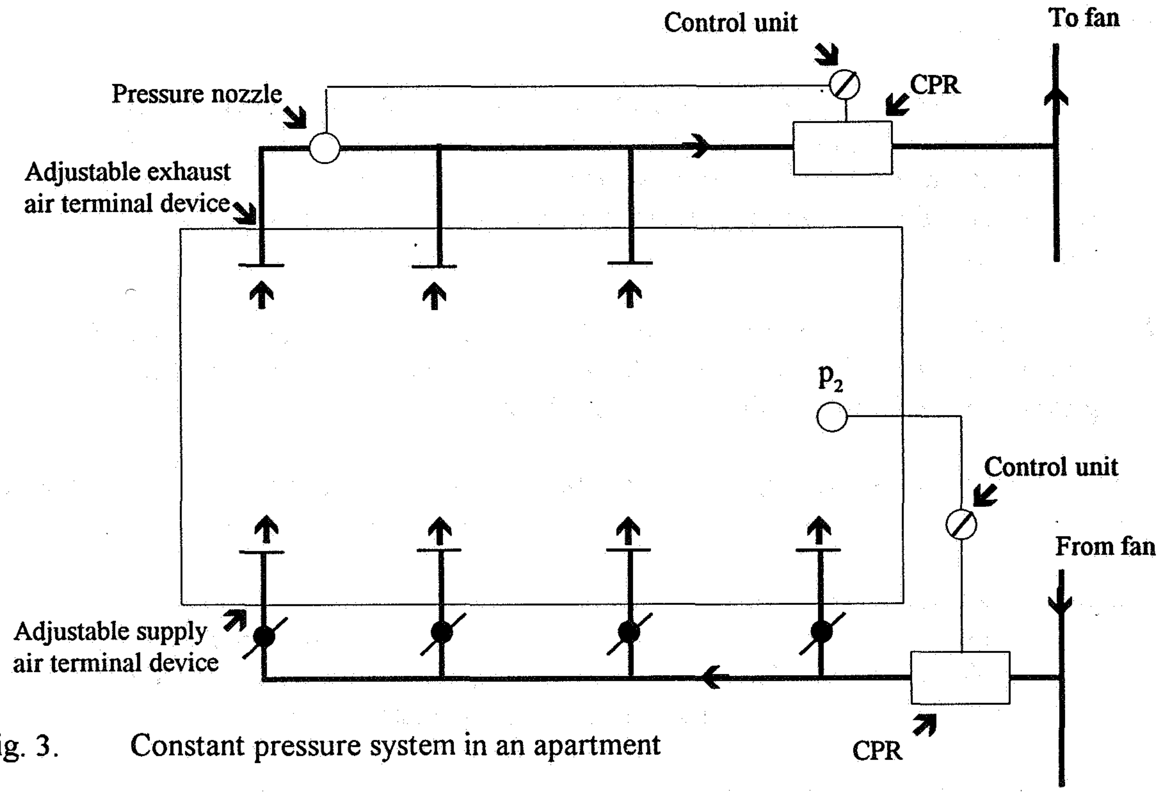


Fig. 3. Constant pressure system in an apartment

Conclusion

It is important that we should develope our ventilation systems. They must be adapted to the requirements made as regards air quality, comfort and the human need of individual regulation.

Flow control flexibility is a neglected field. Although the desire to achieve better systems has existed, progress has been retarded by the increased expenditure so easily resulting from improved solutions.

In this paper we have seen various possible ways, using constant pressure regulators, of achieving simple, flexible solutions. These basic solutions ensure:

1. that the requisite minimum air flows can always be maintained in the individual room units,
2. that the air flows can be forced to a predetermined maximum in the individual room units,
3. that the minimum flow for the dwelling unit as a whole can easily be increased by adjusting the desired value (pressure) setting of the pressure regulator,
4. that air flows can easily be checked, by checking the pressure in the ducts or connecting boxes and knowing the properties of the supply air terminal devices,
5. that the balance between supply and exhaust air flows will always be correct,
6. that there will be no problems with draughts caused by too high a pressure difference between inside and outside.
7. that the noise levels will not exceed the maximum design values, due to the constant pressure in the distributor duct,
8. that the system will be easy to adjust, thanks to the mechanical pressure regulator,
9. that a positive effect will be gained from users or residents being able to control ventilation air flows for themselves.

The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Energy and Environmental Protection Aspects of Desiccant Cooling

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ENERGY AND ENVIRONMENTAL PROTECTION ASPECTS OF DESICCANT COOLING

1. Synopsis

Ventilation and air conditioning systems mainly use fossile primary energies as gas, oil and coal for the heating and cooling processes.

Air conditioning means heating and humidifying the supply air during the winter season and cooling and dehumidifying the supply air in the summer season. For these summer operations the supply air in general is cooled down lower than the dew point in order to dehumidify the air by condensation. Afterwords the supply air is reheated again to reach the required temperature level for room inlet. For this air treatment process cooling and heating energy are used simultaneously. The cooling energy thereby is generated in general in a conventional cooling compressor unit based on CFC-refrigerants.

Due to the threat of an atmospheric ozone depletion leading to a "Greenhouse Effect" mainly caused by CO_2 from burning fossile primary energies and the CFC-based refrigerants (1) used as cooling vapour in compressor chillers new developments of cooling equipment have a realistic chance to enter the HVAC market.

Today it is necessary to operate air conditioning systems with a minimum of primary energy consumption and low pollutant emissions. At the same time the requirements for the indoor air quality are increasing.

Therefore alternative and new air conditioning systems are now introduced using "Desiccative and Evaporative Cooling" (DEC) air treatment processes (2). These units can be operated all over the year and have to be compared under energy and environmental operation aspects with traditional air conditioning systems.

2. Desiccative and Evaporative Cooling (DEC)

In order to separate the summer operations "cooling" and "dehumidifying" in different air treatment steps a rotating desiccant wheel (3) and a rotating heat transfer wheel (4) are combined in a DEC unit. The related processes were proposed already in the early 1940's and granted in the Pennington patent (5). Since that time many contributions have been added to this improvement, but only energy saving activities have brought the breakthrough a few years ago. Today DEC systems are operated with very good results in the USA, Japan and Germany and a broad range of components is available with extremely high efficiencies for adsorption technologies and heat exchangers (6)(7).

According to Fig.1a the DEC unit consists of three well-known air processors: a desiccant air dehumidifier, evaporative coolers and a rotating sensible heat exchanger. The individual function of these principal components is as follows (Fig.1b): the desiccant wheel "A" rotates within the outdoor air stream and removes the moisture from it (1→2). The most suitable rotor is fabricated of silica gel reinforced with inorganic fibres and formed into a honeycomb shape. Active silica gel is synthesized and combined in the honeycomb shape by chemical reaction. It has an excellent water adsorbing ability. The adsorption of moisture on the silica gel causes the temperature of the air to rise. The heat generated during the drying step is removed from the air by a rotating heat recovery wheel "B" with high efficiency (2→3). This heat recovery wheel is non-hygroscopic and fabricated of corrugated aluminium. The evaporative cooler "C" humidifies the dried air to further reduce the dry bulb temperature of the supply air stream (3→4).

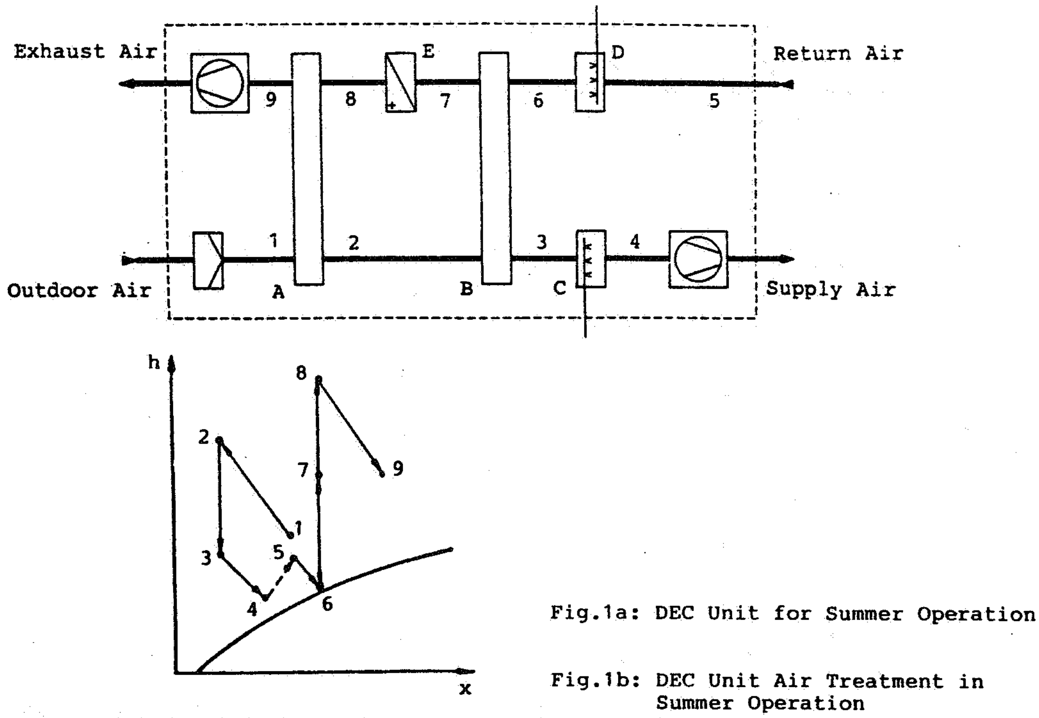


Fig.1a: DEC Unit for Summer Operation

Fig.1b: DEC Unit Air Treatment in Summer Operation

For the reactivation cycle the return air is used by first reducing the dry bulb temperature in the evaporative cooler "D" (5→6). The heat originally generated during dehumidification of the supply air is then removed and transferred back into the reactivation cycle by the heat recovery wheel "B" (6→7). In the heat exchanger "E" external heat energy brings the reactivation air to the required temperature for desorbing the desiccant wheel "A" (7→8). Due to the synthesized silica gel this temperature could be set at a minimum of 50°C which allows to use low level waste heat available from many heat processes, cogeneration processes and also solar energy. When desorbing the desiccant wheel "A" the exhaust air temperature is reduced with increasing the absolute humidity (8→9).

3. Facts and Figures

The new DEC units (8)(9) can operate during winter and summer seasons for heating, humidifying, cooling and dehumidifying the supply air in the same way as a traditional air conditioning device according to a DEC based control equipment (Fig. 2a,b). Therefore the customer and his consultant engineer can compare both air handling units not only on investment costs and on operation and maintanance costs but also on the thermal and electrical energy consumption together with the emissions of pollutants which are referred to thermal and electrical energy generation.

In the following such a comparison is presented. The specific results are based on the calculations of energy consumption according to the German Standard VDI 2067/3 (10). For the emissions of pollutants the specific figures of a German study on environmental protection (11) are used according to Fig.3.

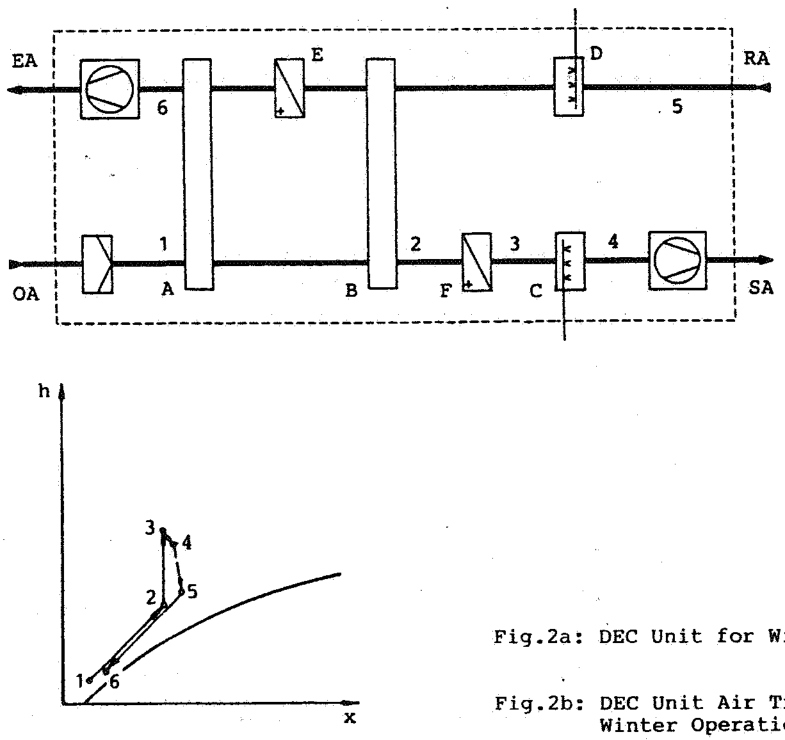


Fig.2a: DEC Unit for Winter Operation

Fig.2b: DEC Unit Air Treatment in Winter Operation

		SO_2	NO_x	Dust	CO_2
Oil fired Boiler		0,4	0,31	0,01	370
Electricity (Coal)		0,75	0,71	0,09	929
District Heating	Coal fired Cogeneration	0,1	0,1	0,01	345
Electricity		0,19	0,28	0,08	438

Fig.3: Specific Emissions kg/MWh Final Energy

The calculations are carried out for

- System A: Air conditioning system with adiabatic humidification and dew point control and CFC based refrigeration (Fig.4).

- System B: Air conditioning system with adiabatic humidification and dew point control, CFC based refrigeration and total energy recovery wheel with sensible and latent heat recovery (Fig.5).

- System C: Air conditioning system as a DEC unit operating the sensible heat recovery wheel and bypassing the dehumidification wheel in the winter season (Fig.6).

- System D: Air conditioning system as a DEC unit operating the dehumidification wheel as total energy recovery wheel with increased rotation speed in the winter season (Fig.7).

The total annual energy consumption for air treatment is calculated for the systems A, B, C and D for

- Thermal Energy (MWh_{th}): Heating and regeneration

- Electric Energy (MWh_{el}): Motors of blowers for internal and external pressure drops; conventional cooling compressor; cooling tower

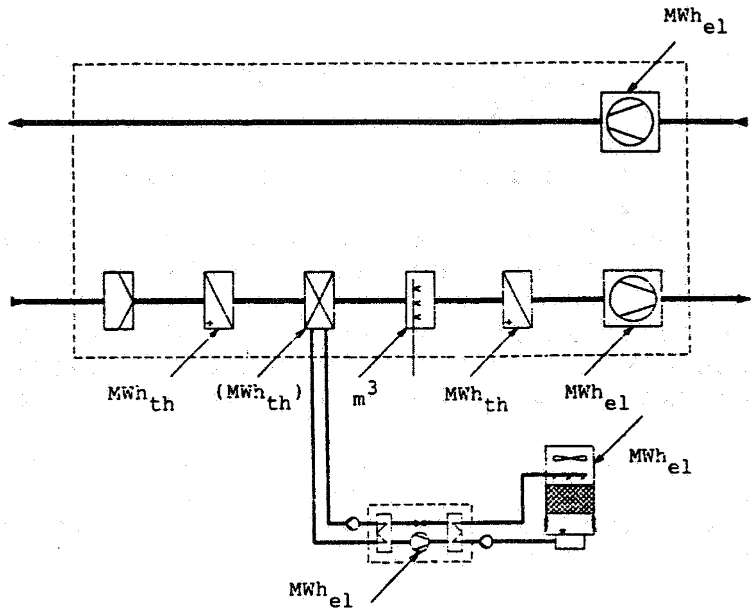
- Water Supply (m^3): Humidification and cooling tower

The total annual emissions of pollutants based on primary energy consumption for heating energy and electricity are calculated according to traditional energy supply systems:

- Systems A, B: Heating with oil, electricity from coal fired power plant

- Systems C, D: Heating and electric energy from a coal fired cogeneration power plant.

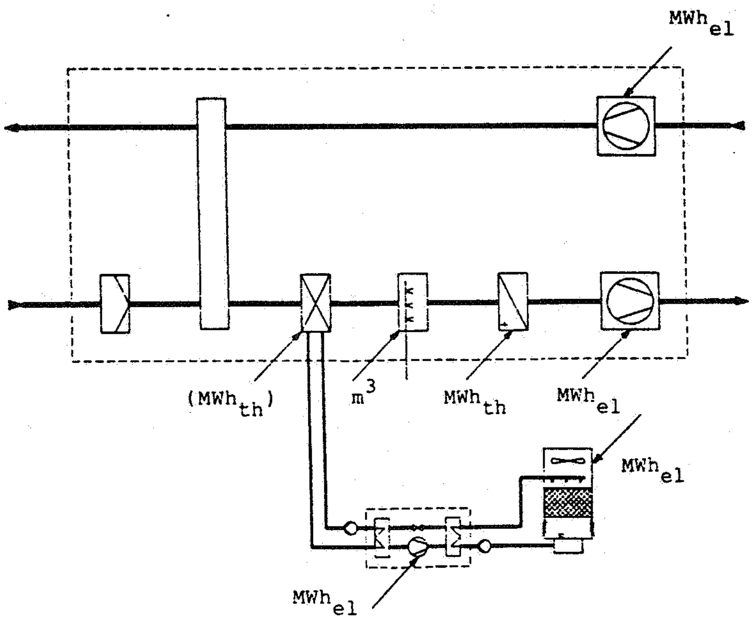
The calculations are derived from a German DEC installation with 25000 m^3/h supply air volume and 22000 m^3/h exhaust air volume for a convention hall operating 12 hours per day at 5 days per week. Supply air conditions are 18 ^oC/10,5 g/kg (16 ^oC after DEC). The results are presented as specific figures based on m^3/s air flow.



Energy Supply per m^3/s		
- Thermal Energy	81	MWh/a
- Electric Energy	14	MWh/a
- Water Supply	120	m^3/a

Emissions per m^3/s		
- SO_2	43	kg/a
- NO_x	35	kg/a
- Dust	2	kg/a
- CO_2	44000	kg/a

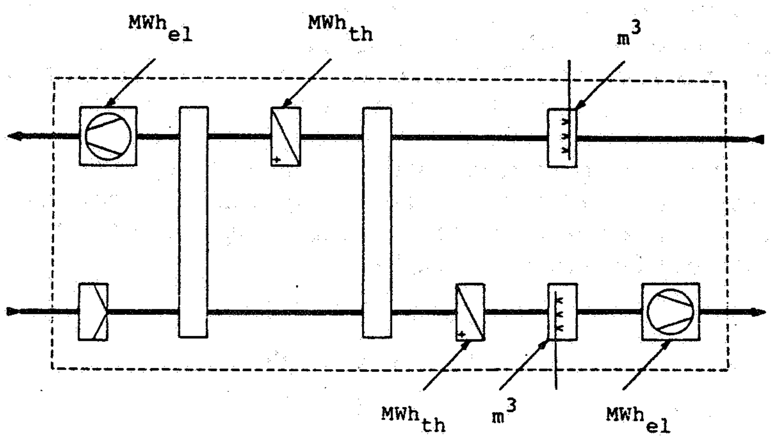
Fig.4: Air Conditioning System A



Energy Supply per m^3/s		
- Thermal Energy	52	MWh/a
- Electric Energy	16	MWh/a
- Water Supply	90	m^3/a

Emissions per m^3/s		
- SO_2	33	kg/a
- NO_x	27	kg/a
- Dust	2	kg/a
- CO_2	34000	kg/a

Fig.5: Air Conditioning System B



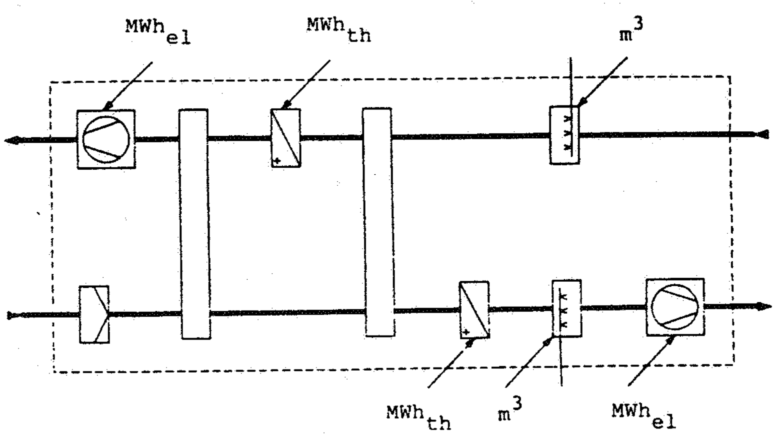
Energy Supply per m^3/s

- Thermal Energy 43 MWh/a
- Electric Energy 14 MWh/a
- Water Supply 90 m^3/a

Emissions per m^3/s

- SO_2 7 kg/a
- NO_x 9 kg/a
- Dust 2 kg/a
- CO_2 22000 kg/a

Fig.6: DEC System C



Energy Supply per m^3/s

- Thermal Energy 33 MWh/a
- Electric Energy 15 MWh/a
- Water Supply 45 m^3/a

Emissions per m^3/s

- SO_2 6 kg/a
- NO_x 8 kg/a
- Dust 2 kg/a
- CO_2 18000 kg/a

Fig.7: DEC System D

4. Comparison

As a new technology the DEC systems have to be compared with conventional air condition systems running with or without energy recovery. Because DEC systems replace the electric driven cooling compressor and CFC's are no longer needed for the cooling cycles the most important comparison is for the total energy consumption including the very attractive application of various cogeneration energy supply systems. As normally for DEC systems the peak electric capacity for summer operation can be reduced the comparison is extended to the annual operation costs with the following specific energy costs:

- Thermal energy 60,- DM/MWh
- Electric energy 180,- DM/MWh
- Electric capacity 250,- DM/kW
- Water 6,- DM/m^3

In Fig.8 the two conventional air conditioning systems A and B are compared with the two DEC systems C and D in relation to 100% for system A. It is evident that the DEC system D with the desiccant wheel operating in the winter season as a total energy recovery wheel in combination with cogeneration energy supply is under every aspect superior to the traditional air conditioning systems.

System	A	B	C	D
Energy				
- Thermal Energy	100	64	53	41
- Electric Energy	100	110	100	105
- Electric Capacity	100	77	29	29
- Water Supply	100	75	75	38
Emissions				
- SO_2	100	77	17	15
- NO_x	100	78	24	22
- Dust	100	100	100	100
- CO_2	100	77	50	40
Operation Costs	100	79	57	50

Fig.8: Comparison of Systems A,B,C,D

Regarding the annual heating energy consumption the DEC heat recovery system D needs only 40% of a conventional airconditioning system. The electric energy consumption is for all systems almost the same, because the DEC systems have higher internal pressure drops. But the DEC electric capacity is only 30% of a conventional airconditioning system. This allows to reduce the tarifs for electric energy.

The water supply can be reduced down to 40%.

Referring the combined application of thermal and electric energy in a cogeneration plant the DEC systems allow a more economical operation of these attractive energy supply systems.

DEC systems can apply solar thermal energy on low temperature level for the regeneration cycle. If 60% of the regeneration energy can be used by solar gains the total annual heating energy of a DEC system can be reduced to 33% of a conventional airconditioning system. This seems to be very attractive.

The total annual operation costs for a DEC system can be reduced to 50%.

Investment costs have to be calculated for every project. In general DEC systems as a total unit can be designed and installed with the same investment costs as traditional systems, because the refrigeration compressor and the cooling tower are no longer necessary. So the benefits of a DEC system of reduced energy consumption and reduced emissions as well as the reduced operation costs can be earned from the beginning.

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

The Testing and Rating of Terminals used on Ventilation Systems

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TESTING THE PERFORMANCE OF FREE STANDING VENTILATION TERMINALS

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Summary

Terminals are used on all types of ventilation system exhausts, often to prevent rain water and animal entry, but also to prevent wind induced flow-reversal and enhance wind induced up-draught. There are many different terminal designs available displaying a wide range of characteristics.

This report discusses a terminal testing and rating method. The tests highlight terminal wind performance as well as terminal resistance to the exhaust flow. The terminals are ranked according to loss coefficients and wind performance which allows them to be matched more closely to system requirements.

Whilst the data gathered here can help with the choice of terminal for any ventilation system, it is probably most applicable to those systems affected by the wind. Such systems include passive stack ventilation, passive gas extraction, combustion flues and chimneys.

This paper is intended as a test guide for manufacturers and as source of information to help system designers with terminal selection.

1. Introduction

Terminals are used on all types of ventilation systems, ranging from mechanical air-conditioning units to chimneys and passive ventilation systems, and their presence inevitably alters system behaviour. Often a particular design is installed for a specific reason; for example to increase wind induced up-draught or prevent wind induced flow-reversal.

Numerous designs are available displaying a wide variety of properties. Wind tunnel testing can highlight both desired and undesired properties, and by rating terminal performance installers can choose the best design for a particular application. For example, a passive gas extraction system may require a terminal which causes up-draught due to wind action whilst the chimney of an open-fire may benefit more from one which prevents wind induced flow-reversal.

This paper discusses a procedure for testing free standing terminals (as opposed to roof ridge vents or tile vents) which can be used as a basis for manufacturers to test their own products. A rating method is suggested which rates terminal performance relative to other designs.

2. Nomenclature

The following notation is used in this report.

C_p	-	a pressure coefficient defined as $C_p = 2\Delta P_{ab}/(\rho U^2)$
$C_{p\mid v=0}$	-	value of C_p when v=0
d	-	duct diameter (m)
g	-	acceleration due to gravity (m/s^2)

K	-	loss coefficient of a terminal
P_{dw}	-	dynamic pressure in the wind (Pa)
P_a, P_b	-	static pressure in the duct and wind respectively (Pa)
Re_{in}	-	Reynolds number relating to the flow inside the duct
Re_{ex}	-	Reynolds number relating to the wind and geometry of the terminal
S	-	suction coefficient
U	-	average wind speed (m/s)
v	-	average duct flow speed (m/s)
Z_a, Z_b	-	height of point 'a' and 'b' above reference level respectively (m)
ρ, ρ_d, ρ_w	-	density of air, of air in duct and of air in wind respectively (kg/m^3)
v_d, v_w	-	kinematic viscosity of air in duct and wind respectively (m^2/s)
ΔP	-	a static pressure difference (Pa)
ΔP_{ab}	-	static pressure difference between points 'a' and 'b' ie. $P_a - P_b$ (Pa)
ΔP_{loss}	-	pressure loss due to flow resistance of terminal between points 'a' and 'b'. $\Delta P_{loss} = \frac{1}{2}K\rho v^2$ (Pa)
$\Delta P_{suction}$	-	pressure change between points 'a' and 'b' caused by wind. $\Delta P_{suction}=\frac{1}{2}S\rho U^2$ (Pa)
ΔP_i	-	wind and duct flow interaction factor (Pa)

3. Theory

3.1 Flow resistance

Ventilation ducts, terminals, bends, and inlets all restrict flow to some extent and cause pressure drops. Loss coefficients give a measure of this resistance, the larger the factor, the greater the resistance.

The loss coefficient, K, for a terminal is defined by:

$$K = 2\Delta P/(\rho v^2) + 1 \tag{1}$$

where ΔP is the static pressure drop across the component and v is the average air velocity through that component. It is calculated under zero wind conditions.

For turbulent duct flow (Reynolds number, Re_{in}, >5000) terminal loss coefficients are approximately constant. It is this value that is calculated. For laminar flow (Re_{in}<2000) the loss coefficient varies inversely with Reynolds number(1).

3.2 Wind effects

Wind can have significant effects on the performance of a ventilation system, especially when the system operates passively. It can assist the flow by causing up-draught or it can restrict the flow and may sometimes cause flow-reversal. Such effects depend on the ventilation system, terminal design, wind angle and wind direction.

To analyse wind performance two separate flows need to be considered, ie the wind and duct flow. Dimensional analysis and Bernoulli's equation, discussed below, give an insight to how terminals perform in the wind.

3.2.1 Dimensional Analysis
Assume that the static pressure difference, ΔP_{ab}, between the inside of the duct and the wind is (for a particular wind direction, wind angle and terminal geometry) a function of v, U, ρ_d, ρ_w, ν_d, ν_w, d. That is:

$$\Delta P_{ab} = h(v, U, \rho_d, \rho_w, \nu_d, \nu_w, d) \quad (2)$$

which in dimensionless terms gives,

$$C_p = \Delta P_{ab}/P_{dw} = j(v/U, \rho_d/\rho_w, Re_{in}, Re_{ex}) \quad (3)$$

Experimental data(2) shows that the pressure coefficient C_p is independent of both Re_{in} and Re_{ex}. Hence if the two air densities are equal (ρ_d=ρ_w), C_p is a function of v/U alone, ie:

$$C_p = j(v/U). \quad (4)$$

3.2.2 Bernoulli's equation
Applying Bernoulli's equation between points 'a' and 'b' in Figure 1 we find,

$$P_a + \rho_d g Z_a + \rho_d v^2/2 = P_b + \rho_w g Z_b + \rho_w U^2/2 + L \quad (5)$$

where $L = \Delta P_{loss} + \Delta P_{suction} + \Delta P_i$.

After substituting the relevant expressions for ΔP_{loss} and $\Delta P_{suction}$ (see Nomenclature), and if ρ_w=ρ_d=ρ and $Z_a = Z_b$, it follows that:

$$C_p = 2\Delta P_{ab}/(\rho U^2) = C_{p|v=0} + (K-1)(v/U)^2 + (\Delta P_i/P_{dw}) \quad (6)$$

where the coefficient $C_{p|v=0}$ is a constant. ΔP_i is an unknown representing a pressure difference due to the interaction between the wind and duct flows. This will depend on terminal geometry, wind direction and wind angle.

4. Experimental details
It has been shown in the theory that the pressure coefficient C_p is a function of v/U. Thus the performance of a terminal can be found by measuring the variation of C_p as v and U vary.

The experimental rig, devised after the revision of the existing British Standard(3) and previous work in this area, is shown in Figure 1. A fan provides a constant flow through a duct. The flow rate, controlled by a butterfly valve, is monitored using a volumetric flow meter. Swirl, caused by the fan, is reduced prior to air entering the flow meter by using a flow straightener. Simulated duct flow velocities, v, are 0, 1, 2, 4m/s for the 110mm diameter duct and 0, 1, 2, 3m/s for the 150mm diameter duct. These represent typical flows found in most types of ventilation systems.

From the flow meter the air passes through a length of flexible pipe to the base of the duct and then through another flow straightener. The duct (either 110mm or 150mm in diameter depending on the terminal size) measures a minimum of 13 diameters from the flow straightener to the pressure tappings. This length, together with the flow straightener, provides for an approximately uniform velocity profile within the pipe at the pressure tapping location.

To avoid interference from air exiting the system through the terminal, the wall pressure in the pipe is monitored 0.5m from the terminal. The tapping points are made, located, and connected to the micromanometers according to BS848[4]. The static pressure difference, ΔP_{ab}, between the pipe wall and the wind is monitored by two identical micromanometers to ensure the results are both accurate and reliable.

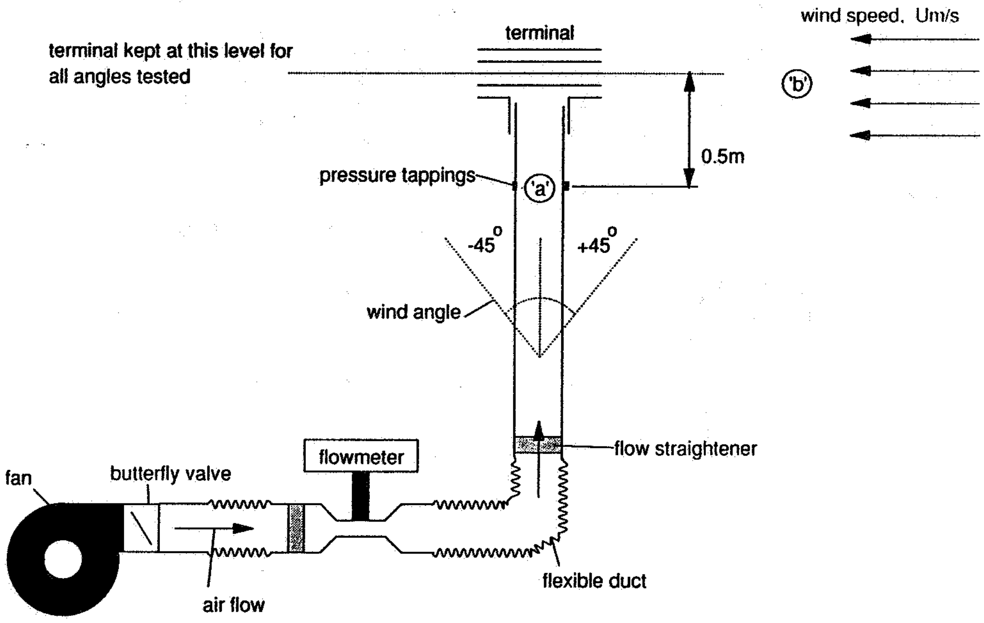


Figure 1: The experimental apparatus

A open-jet wind tunnel blows air directly at the terminal at speeds, U, between 0m/s and 10m/s, monitored using a pitot-static tube connected to a micromanometer. There are no blockage effects since the entire rig sits outside of the tunnel and the terminals are small relative to the tunnel cross-section.

The ventilation pipe can be tilted to monitor the terminals performance for various wind angles from the vertical. Angles range from -45° (away from the wind) to +45° (into the wind), in 15° steps. The terminal is kept at the same height (in line with the centre of the wind tunnel) for each angle by moving the ventilation pipe (see Figure 1). This ensures, as far as possible, that the wind speed conditions are the same for each angle.

Performance variation with wind direction is also investigated when necessary, depending on terminal geometry. For terminals that are symmetrical about the ventilation pipe centre line, only one wind direction is needed. For others, tests are carried out with the wind blowing onto the side with the minimum cross-sectional area and with it blowing onto the side with the maximum cross-sectional area. This is in accordance with BS715[3].

The pressure differences are monitored for a specific wind angle and wind direction as the duct flow and wind speed is varied. The tests are repeated for all combinations of wind angles and wind directions for each terminal. The terminals tested are shown in Figures 2 and 3.

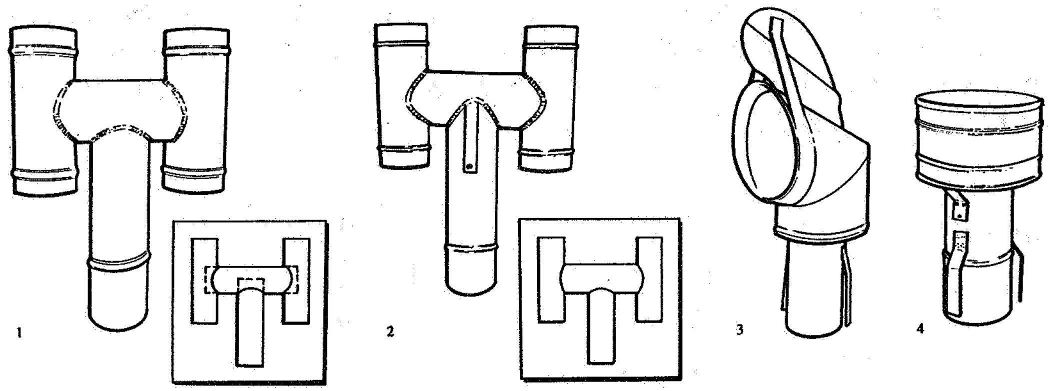


Figure 2: The 110mm terminal designs

1- H pot 1	2- H pot 2
3- Grey vane	4- Gas flue
5- Mushroom cap	6- Balloon

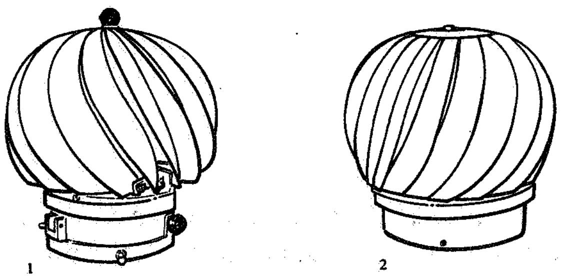
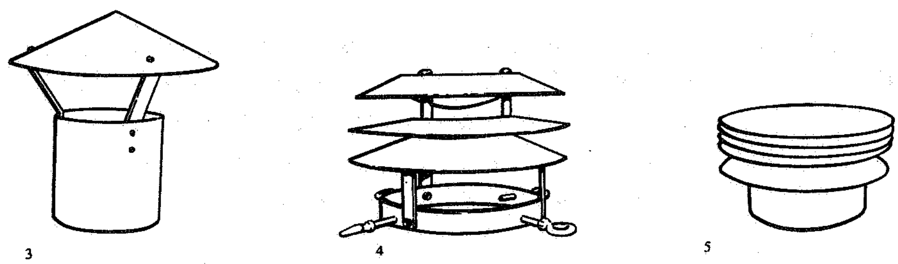


Figure 3: The 150mm terminal designs

1- Rotating cowl 1	2- Rotating cowl 2
3- Chinese hat	4- Aero-dynamic cowl
5- Gas flue	



5. Data analysis and results

5.1 Flow resistance

From Equation 1, a plot of ΔP_{ab} against $\frac{1}{2}\rho v^2$ will produce a straight line with gradient equal to K-1. Table 1 shows the loss coefficients for the designs shown in Figures 2 and 3. The figures are ranked in descending order of performance with the least restrictive first.

Terminal 110mm	Loss factor, K	Terminal 150mm	Loss factor, K
Open pipe	1.0	Open-pipe	1.0
Mushroom cap	2.1	Chinese hat	1.1
Balloon	2.2	Rotating cowl 1	1.1
H pot 1	2.7	Rotating cowl 2	1.3
Gas flue	3.2	Gas flue	2.0
H pot 2	4.7	Aero-dynamic cowl	2.3
Grey vane	6.7		

Table 1: Ranking terminals using loss coefficients only

5.2 Wind effects

5.2.1 Suction

The pressure coefficient C_p indicates how a terminal is performing for a specific duct flow and wind condition. Plotting C_p against v/U gives an approximate quadratic (Equation 6, assuming ΔP_i is small compared to the other components). When the data lie in the negative C_p region the extractive properties of the terminal exceed the resistive. When data are in the positive C_p region resistive forces exceed the extractive.

The two values discussed below can be used to describe a terminal's extractive performance (see Figure 4):

a) the value of C_p when v=0, [$C_{p|v=0}$]. This factor gives an indication of the greatest suction that can be produced by a specific terminal. The more negative this value the greater the suction. A range of $C_{p|v=0}$ values are collected from the experiment since this value varies with wind direction an wind angle. From this range an average value and an error value (two standard deviations) are calculated.

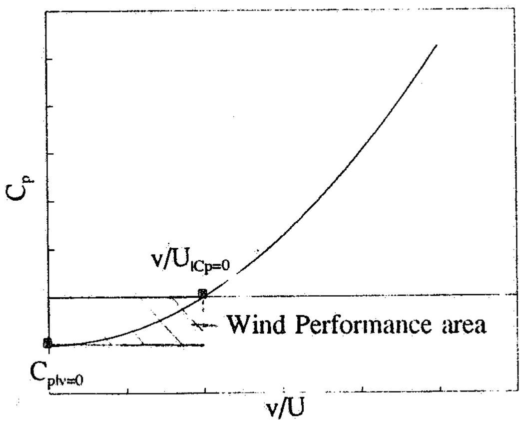


Figure 4: The suction performance indicators

b) the value of v/U when C_p=0, [$(v/U)_{|Cp=0}$]. This indicates the v/U range for which the extractive property dominates the resistive. As with $C_{p|v=0}$, this value varies with wind direction and angle, and an average value and error value (two standard deviations) are calculated.

Under specific conditions a terminal may not cause suction when v=0 (an example of this is when open pipe points into the wind causing flow-reversal). When this is the case the minimum of the $(v/U)_{|Cp=0}$ range is undefined, and in such cases flow-reversal is possibility.

These two figures are best combined by multiplying together the average of each range to produce a single *wind performance indicator*. This figure indicates the average performance over the conditions examined. Table 2 shows these values in descending order of performance, the first being the terminal that induces up-draught the most.

Terminal 110mm	Wind performance indicator	Terminal 150mm	Wind performance indicator
H pot 1	-0.23 ± 0.22	Rotating cowl 1	-0.53 ± 0.50
Balloon	-0.16 ± 0.09	Rotating cowl 2	-0.12 ± 0.07
H pot 2	-0.16 ± 0.13	Gas flue	-0.07 ± 0.04
Gas flue	-0.11 ± 0.11	Aero-dynamic cowl	-0.03 ± 0.04
Grey vane	-0.09 ± 0.04		

Table 2: Wind performance indicator ranking

5.2.2 Flow-reversal

The easiest way of identifying flow-reversal potential is to inspect the $C_{plv=0}$ range. If the maximum value is positive then there is flow-reversal potential. Alternatively flow-reversal is possible whenever the pressure drop across the terminal increases with wind speed ie. $d(\Delta P_{ab})/dU > 0$.

It is shown in Figure 5 how the wind angle alters the performance of a "Mushroom cap" terminal. Flow-reversal will not occur between the angle range 45° to -15°, but it can for angles -30° and -45°. Figure 6 shows the results for the "H pot 1". This terminal protects against flow-reversal for all wind angles investigated.

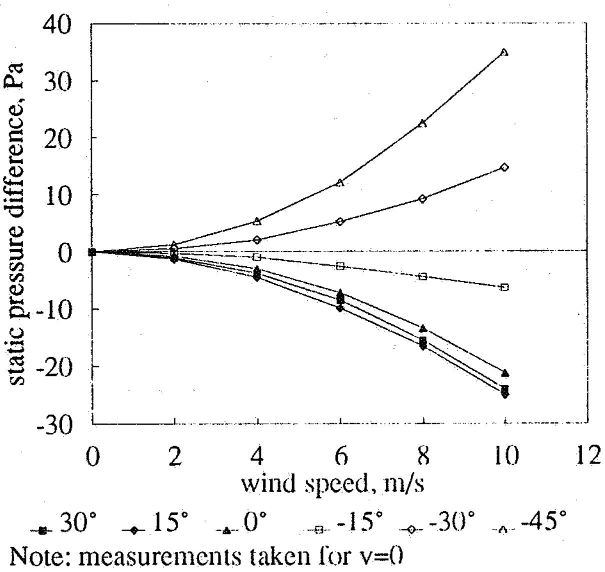


Figure 5: Flow-reversal for the "Mushroom cap" at -30° and -45°

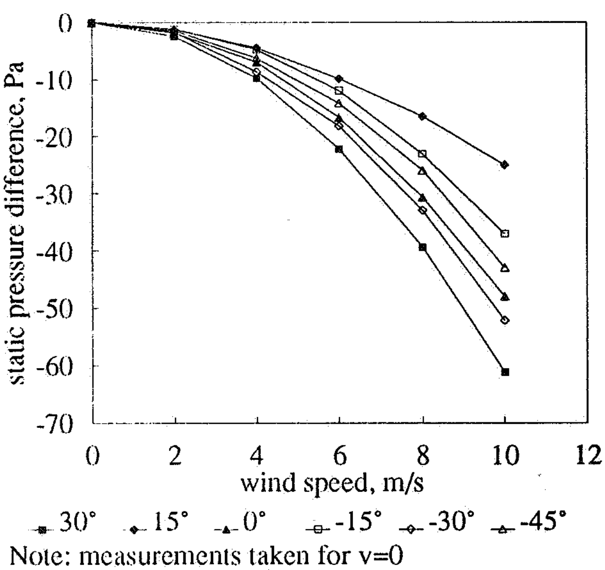


Figure 6: No flow-reversal for "H pot 1"

6. Discussion of results

Loss factors are seen to vary widely, some designs being considerably more restrictive than others. For mechanical ventilation systems the loss coefficient ranking (Table 1) may be used as a basis for terminal selection. Note the large difference between the loss factors for the two H type terminals which are visually similar.

For systems affected by the wind the wind performance indicator (Table 2) should be used to select terminals. A large 'error' is associated with this type of indicator because of the wide range of wind conditions examined. The error shows how consistent a terminal performs over the range of conditions. It is interesting that there is a large difference between the two rotating cowls.

Terminals capable of causing flow-reversal are the open pipe, "Mushroom cap" and "Chinese hat". This will only occur for specific wind directions and angles and it is most likely to occur in passive ventilation systems. Terminals which display such properties are undesirable especially when used on open-flued combustion appliances.

7. Conclusions

The test procedure discussed in this paper can be used to rank terminal performance. Such data allows terminals to be selected on the merit of their performance.

Terminals can be rated using three factors that are easy to establish ie loss coefficient, wind performance indicator and flow-reversal potential. Loss coefficients are most significant for mechanical ventilation systems whilst wind performance is more relevant for passive venting systems. Protection against flow-reversal is important for ventilation of open-flued combustion appliances.

The terminals examined in this paper may be grouped as follows:
(a) those with large loss factors (the most restrictive): "Gas flue" (110mm), "H pot 2", and "Grey Vane".
(b) those good at inducing up-draught: "Rotating cowl 1", "H pot 1".
(c) and those which may cause flow-reversal: open pipe, "Mushroom cap", "Chinese hat".

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The Role of Ventilation
15th AIVC Conference, Buxton, Great Britain
27-30 September 1994

Domestic Ventilation with Variable Volume Flows

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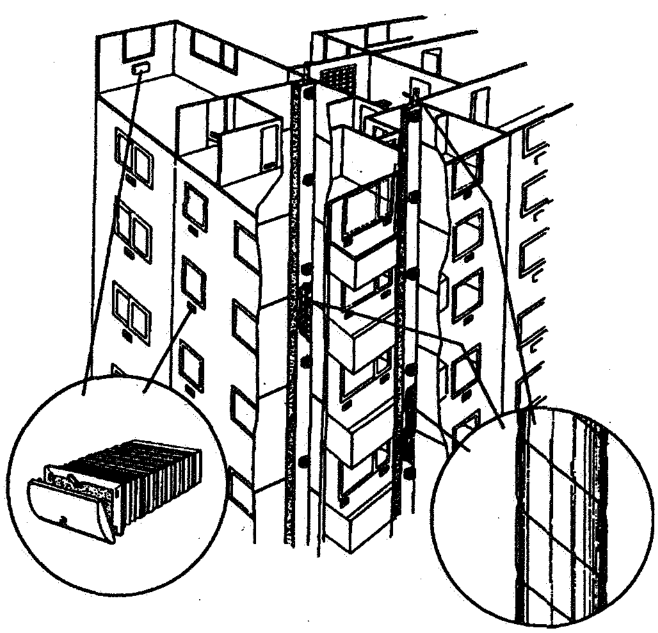
LUNOS

Decentralized Apartment Ventilation with the One-Pipe Ventilation System

by Dr. (Engr.) K. Ulrich Kramm

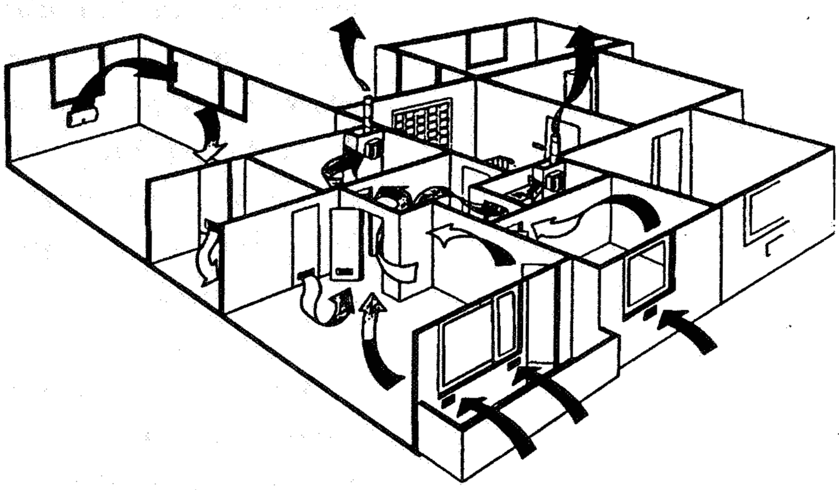
The system of decentralized apartment ventilation has been successfully used for several years in multiple-story apartment construction in the Federal Republic of Germany.
With this tried and tested system, the individual apartments are vented into a common exhaust shaft with decentralized apartment ventilating fans. The special designs of the individual fans ensure a constant volume flow of the outgoing air in the individual apartments, in spite of the large pressure variations into the outgoing air conduits. Non-return flaps prevent a back flow of the outgoing air. An extensive standardizing work safeguards the exactly defined installation conditions. /1/ /2/
Special fireproofing equipment prevents the spreading of fire longwise through the evacuation air ducts between the stories of the building. /3/
The additional air flows through the wind-pressure controlled outside air openings into the individual apartments. The air with pollutants is enriched there and comes into the central air evacuation duct through the individual fans. A building with the installed one-pipe system is depicted in Illustration 1.



Illus. 1: Diagrammatic view of decentralized apartment ventilation

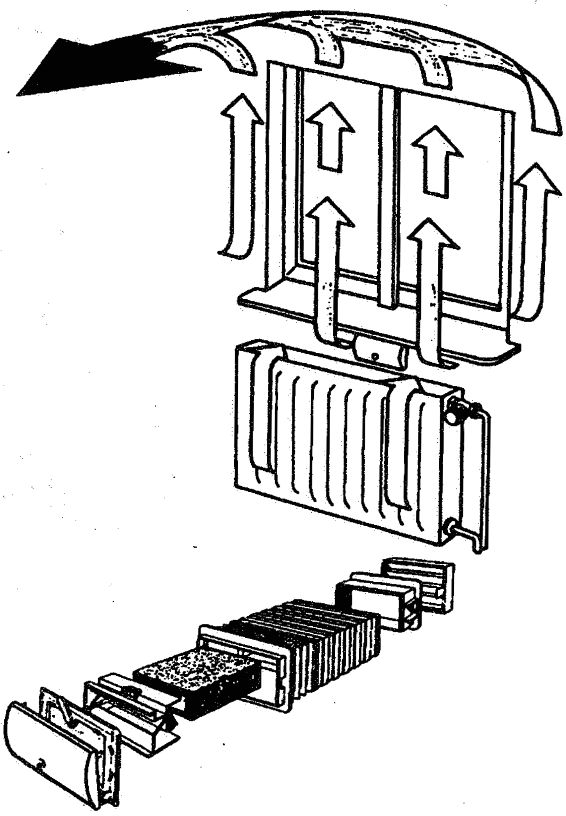
In Illustration 2, the air path in an apartment is depicted. The fans produce a slight under-pressure in the apartment. The fresh air flows into the apartments through the outside air openings.

The control units contained in the outside air openings ensure an additional air volume flow independent of the wind pressure. The additional air is filtered. The thermal insulation in the air inlet conduits prevents the formation of condensation in the air inlet element. Special installations bring about effective acoustic insulation.



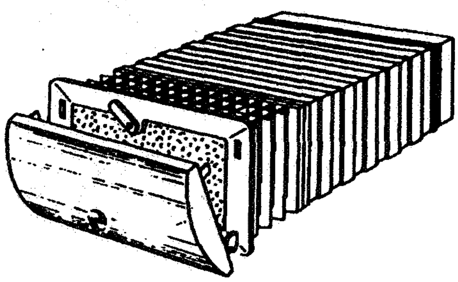
Illus. 2: Air currents with the decentralized apartment ventilation.

The outside air openings are installed over the heating radiators. The warmed air flows to the ceiling and mixes with the additional air.

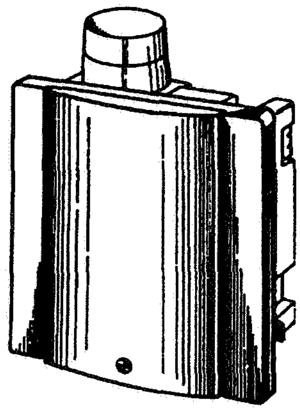


Illus. 3: ALD - Outside Air Opening Element
(ALD from the German)

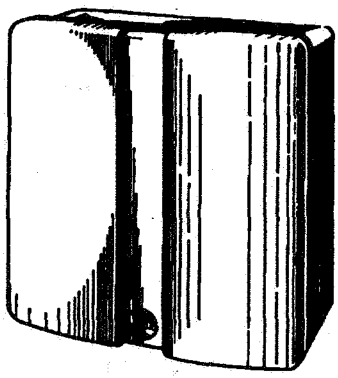
The occurrence of drafts is not possible. The flow speed in the outside air opening and to the outflow openings is very small.
A comfortable and healthy living climate ensues.
In Illustration 4, the devices belonging to the fan family are depicted. The Sapphire fan type is simply mounted on the wall with its housing (surface-type fan).



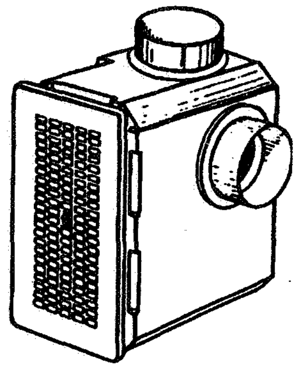
Type ALD



Type Scalar



Type LRK–2



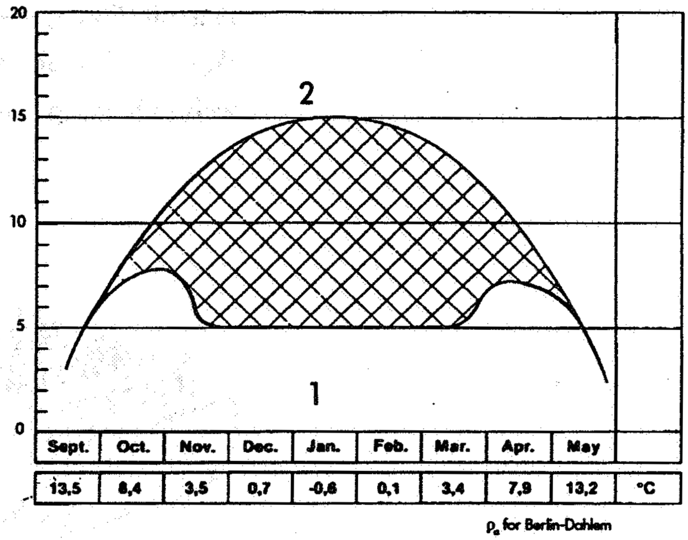
Type Sapphire

Illus. 4: The new generation of fans for surface and flush type mounting

The Scalar fan type was developed for flush mounting. The back side is mounted countersunk in the air shaft, so that only the flat and beautifully designed upper part projects into the room.
Both fans can be delivered as moisture controlled fans. The control input Air Moisture controls the volume flow sucked off.
The LRK2 fan type was developed for the simultaneous ventilation of two independent rooms. The ALD type outside air opening rounds out the program of delivery.

Heating energy savings through demand-controlled apartment ventilation

(W/m²)
spec.
heat con-
sumption



= saved ventilation heat consumption

1 = Ventilation heat consumption with demand controlled apartment ventilation

2 = Ventilation heat consumption with an 0.8-fold change of air

Illus. 5: Heating energy savings

It is known that substantial heating cost savings result with a demand-controlled apartment ventilation. /4/

The surface represented with hatching in Illustration 5 describes the saved ventilation heat consumption. Even larger savings result with respect to uncontrolled window ventilation.

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