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Ventilation System Performance

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Proceedings

Volume 2

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Preface

International Energy Agency

The International Energy Agency (IEA) was established in 1974 within the framework of the Organisation for Economic Co-operation and Development (OECD) to implement an International Energy Programme. A basic aim of the IEA is to foster co-operation among the twenty-one IEA Participating Countries to increase energy security through energy conservation, development of alternative energy sources and energy research development and demonstration (RD&D). This is achieved in part through a programme of collaborative RD&D consisting of forty-two Implementing Agreements, containing a total of over eighty separate energy RD&D projects. This publication forms one element of this programme.

Energy Conservation in Buildings and Community Systems

The IEA sponsors research and development in a number of areas related to energy. In one of these areas, energy conservation in buildings, the IEA is sponsoring various exercises to predict more accurately the energy use of buildings, including comparison of existing computer programs, building monitoring, comparison of calculation methods, as well as air quality and studies of occupancy. Seventeen countries have elected to participate in this area and have designated contracting parties to the Implementing Agreement covering collaborative research in this area. The designation by governments of a number of private organisations, as well as universities and government laboratories, as contracting parties, has provided a broader range of expertise to tackle the projects in the different technology areas than would have been the case if participation was restricted to governments. The importance of associating industry with government sponsored energy research and development is recognized in the IEA, and every effort is made to encourage this trend.

The Executive Committee

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

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

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, Italy, Netherlands, New Zealand, Norway, Sweden, Switzerland, United Kingdom and the United States of America.

VENTILATION SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 25

MEASUREMENT OF SUB-FLOOR VENTILATION RATES-COMPARISON WITH BREVENT PREDICTIONS

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ABSTRACT

The performance of ventilation provision in subfloor cavities is relevant to the fields of energy efficiency, condensation risk, and air quality. Thorough programs of site measurements of ventilation rates by means of tracer gas tests are in general protracted and expensive, and it is quite clear that would be highly desirable to be able to predict ventilation rates given details of building design, ventilation provision, and degree of exposure.

This paper describes a series of sub-floor cavity tracer gas ventilation measurements performed upon a low energy test house, and compares the results with a set of ventilation rates predicted for the sub-floor cavity by the Building Research Establishment ventilation prediction program BREVENT.

Agreement between measured and predicted values is generally good, although significant discrepancies are observed at certain incident wind angles. This is thought to be due to difficulties in obtaining reliable data pertaining to wind pressure effects on ventilation openings at very low levels.

INTRODUCTION

Knowledge of ventilation rates in subfloor cavities is assuming an increasing significance : methane seepage from landfill sites and radon gas ingress from certain rock formations come to mind as examples of comparatively new areas of research in which reliable subfloor ventilation data would be of value. It is unfortunately the case that the demand for data is not matched by the supply. Data sets produced within the United Kingdom are rare and far from comprehensive (see for example [1]). It could be reasonably supposed that two of the reasons for the deficiency in data are the relatively labour intensive, time consuming nature of tracer gas studies, coupled with the difficulties associated with achieving tracer gas injection and uniform mixing in cavities of extreme aspect ratios. It would be decidedly advantageous to have at one's disposal a means of predicting ventilation rates in subfloor cavities. To this end, BRE have extended the BREVENT computer program so as to cover subfloor cavities.

This paper describes a tracer gas study of subfloor cavity ventilation rates carried out by UMIST as part of a programme of research undertaken by The Timber Research and Development Association (TRADA) on behalf of the Department of the Environment. The data thus produced is compared with predicted ventilation rates generated by BREVENT using parameters for the test house used in the tracer gas study.

EXPERIMENTAL DETAILS

(i) Tracer gas tests

The test house is shown in figure 1. It is semi-detached, has three bedrooms, and is of low energy design. The whole house ventilation rate is of the order of 0.15 air changes per hour at a mean windspeed of 3m/s. It should be noted that the north side of the house is shielded by a 2 metre high wooden fence approximately 6 metres away.

The subfloor cavity is more accurately described as a crawl space, since its height is approximately 1 metre. The cavity volume is approximately $45m^3$. Ventilation is provided by means of four airbricks, each with 8no $675mm^2$ rectangular slots. On one of the airbricks, 5 of the slots are obscured by a concrete doorstep. The total open area of ventilation openings is $18225mm^2$, corresponding to $985mm^2$ open area per metre run of exposed external wall. This compares with the $1500mm^2$ per metre run <u>or</u> 500mm2 per m^2 of floor area (whichever is the greater) recommended by BS 5250[2].

Subfloor cavity ventilation rates were measured using the standard parallel column portable gas chromatograph developed at UMIST. This apparatus is well documented,(see for example[3]) and will not be described here. The height of the cavity enabled a standard tracer gas injection strategy to be used: mixing was achieved by means of oscillating desk-top fans. A set of 56 ventilation rate measurements were performed over a range of windspeeds and directions.

(ii) BREVENT simulations

BREVENT is a computer model, written in BASIC, that has been developed by the Building Research Establishment to predict ventilation rates in a building which is represented by a single zone. The program is most suited to domestic buildings. The model is based upon the work of Warren and Webb[4]. BREVENT calculates a ventilation rate for a building by considering the effects of temperature difference, windspeed and wind direction. the calculation procedure assumes an initial pressure difference, and then, on the basis of this pressure difference calculates the flows across each element of the building envelope. These flows are recalculated by means of an iteration procedure until all airflows are balanced.

Recent developments allow the program to include a wide range of flow elements, such as vertical ducts, extract fans, flues and combustion appliances, and windows. It is also possible to include a ventilated subfloor space within a building configuration, and in this circumstance the model has two zones. For more information on the current BREVENT model, reference [5] is recommended.

The key input variables to BREVENT are as follows:

(i) The air leakage of the building fabric at a reference internal/external pressure difference (obtained by fan pressurization testing);

(ii) pressure coefficient data for the building surfaces and air bricks (in the case of these predictions derived from BRE wind tunnel measurements and summarised in Tables 1 and 2 respectively);

(iii) floorboard and skirting air leakage data (based on site measurements);

(iv) the area of ventilation openings to the subfloor cavity.

	Wind angle											
Face No.	0	30	60	90	120	150	180	210	240	270	300	330
2	-.09	-.12	-.13	.04	-.11	-.13	-.09	-.13	-.06	.08	-.06	-.14
3	.29	.34	.23	.08	-.072	-.15	-.15	-.13	0	.1	.16	.24
4	-.2	.02	.29	.32	.29	.06	-.2	-.33	-.07	.07	-.08	-.33
5	-.18	-.18	-.12	.08	.31	.47	.38	.3	.19	.13	0	-.16
6	0	0	0	0	0	0	0	0	0	0	0	0

TABLE 1 - Surface pressure coefficients (nb face numbers refer to Figure 1)

For the purposes of the validation exercise, BREVENT was run with the following temperature and wind parameters:

(i) internal and external temperatures set equal at 17 deg C (bearing in mind that the site measurements were performed in early summer);

(ii) wind direction varying in increments of 30 degrees from 0 to 330 degrees (note that 0 degrees corresponds to a Northerly direction);

(iii) wind speed at 10 metres height varying in increments of 0.25m/s between 1.5 to 4.0m/s.

Wind angle

Face No.	0	30	60	90	120	150	180	210	240	270	300	330
3	.33	.4	.3	.12	-.05	-.14	-.15	-.12	.05	.15	.21	.29
4	-.17	.02	.24	.25	.25	.04	-.18	-.31	-.05	.09	-.07	.29
5	-.14	-.14	-.06	.1	.31	.46	.35	.29	.19	.15	.04	.1
6	0	0	0	0	0	0	0	0	0	0	0	0

Table 2 - Pressure coefficients at air bricks

RESULTS AND DISCUSSION

(i) Tracer gas tests.

The results obtained are presented graphically in figures 2 and 3. It should be noted that during the test period, the wind was not observed to blow from either the east or north-east, meaning that the range of experimental data did not completely match the range of BREVENT predictions.

From the results, it can be seen that the ventilation rates associated with each wind direction tend to merge together at windspeeds approaching 1m/s: the range of ventilation rates exhibited at this windspeed seems to lie between 0.15 and 0.2 air changes per hour. The clear directional influence of prevailing wind direction can be seen as windspeed increases. Northerly and Southerly winds give ventilation rates which are substantially higher than for any other wind directions: of the two, Southerly winds appear to induce ventilation rates which are approximately 30% higher than for a Northerly wind of the same strength. It is suggested that this is a reflection of the sheltering afforded by the 2 metre high fence which surrounds the north side of the test house.

Of all wind directions, the West, parallel to the ridge, gives the lowest ventilation rate for a given windspeed. At 3.5m/s, the ventilation rate for a westerly wind is approximately 0.4 air changes per hour, that is, only of the order of 22% of the ventilation rate associated with a Southerly wind of the same mean speed. At higher windspeeds, the difference becomes more pronounced. These results demonstrate that to optimise ventilator performance, ventilation openings should ideally be distributed between all external walls. The distribution of ventilation openings on two sides of a building only should, if possible, be avoided. Ventilation rates for other wind directions fall between the extremes shown by the North, South, and Easterly directions.

(ii) Comparison of tracer gas tests and BREVENT predictions.

A detailed comparison was carried out between the tracer gas test results and the BREVENT predictions by plotting graphs showing the measured and predicted ventilation rates over a range of windspeeds for a range of wind directions. These are shown in figures 4 to 9.

Reasonable agreement can be seen between measured and predicted values for most of the wind directions compared. The notable exceptions to this are the 180 degree (South) and 315 degree (North West) directions, for which measured ventilation exhibited a greater straight line increase with increasing windspeed than did the BREVENT predictions. It is suggested that the principle reason for this under-estimation by BREVENT is the choice of surface pressure coefficients made in order to try and make allowance for the 2 metre high boundary fence to the North of the property. The sets of surface pressure coefficients used by BREVENT are a reflection of the general terrain conditions around the building in question: any allowances for shielding on individual faces has to be made by the program user. In the particular case of these predictions, the presence of the fence was estimated to result in a 20% reduction of the standard surface pressure coefficient at 10% housing density for the North face of the building. Clearly this estimate may not have been as accurate as might have been desired:however, no experimental data existed upon which to make a more reliable assessment of the fence. The consequences of the inaccuracy in the assessment of the shielding effect is accentuated by the close proximity of the ventilation openings to the ground. There is an obvious need for further research into the effects of shielding on the performance of ventilation openings.

CONCLUSIONS

Based upon the findings of this limited validation exercise, BREVENT would appear to provide reasonable estimates of the air change rate within a ventilated subfloor cavity. However, some discrepancies exist between predicted and measured ventilation rates for certain wind directions which clearly indicate that the reliability of BREVENT predictions could be enhanced by a better knowledge of the influence of local shielding upon surface pressure coefficients, particularly in the vicinity of ventilation openings at low levels.

ACKNOWLEDGEMENTS

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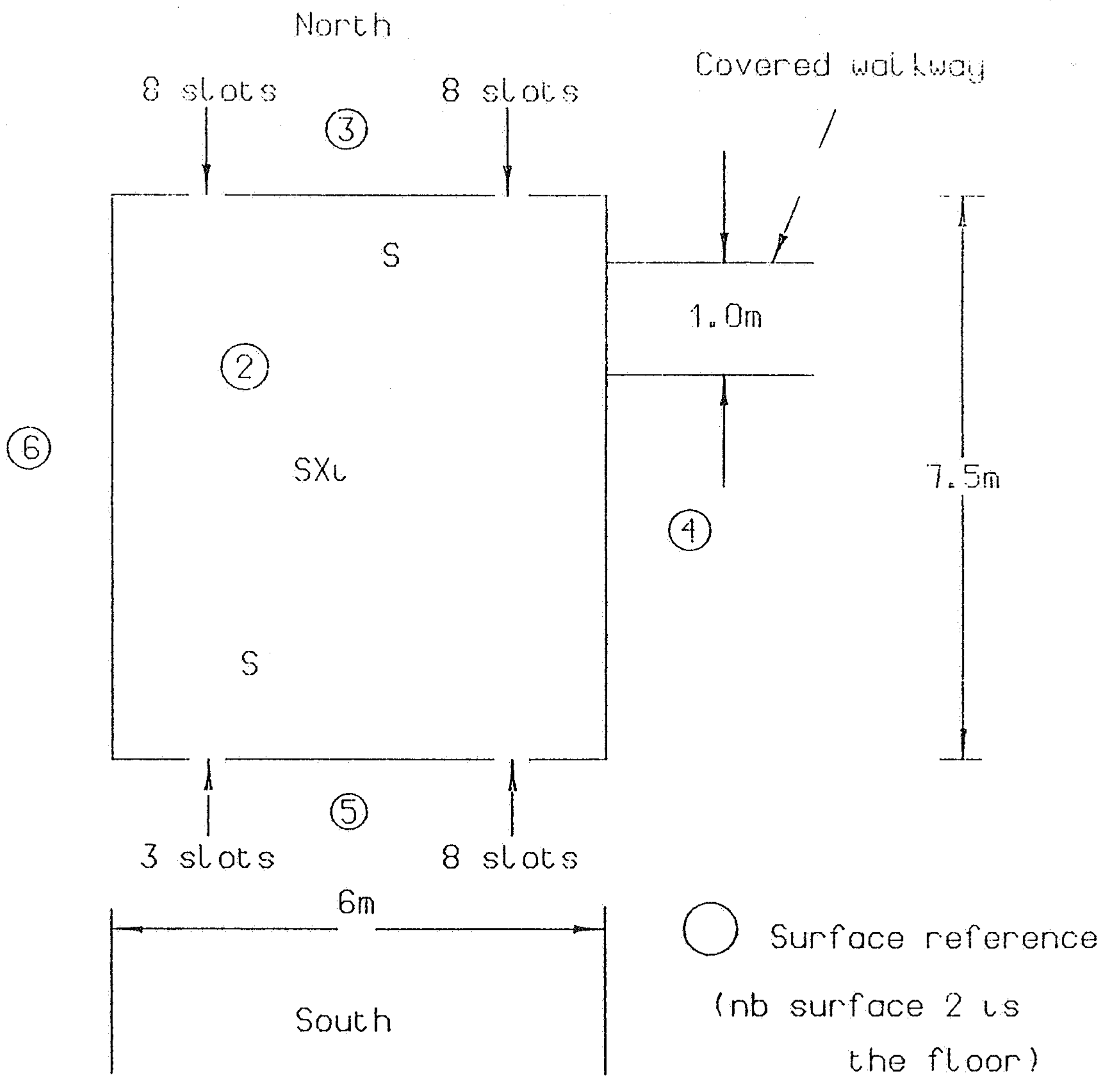


SUBFLOOR VENTILATION PROVISION

Figure 1.

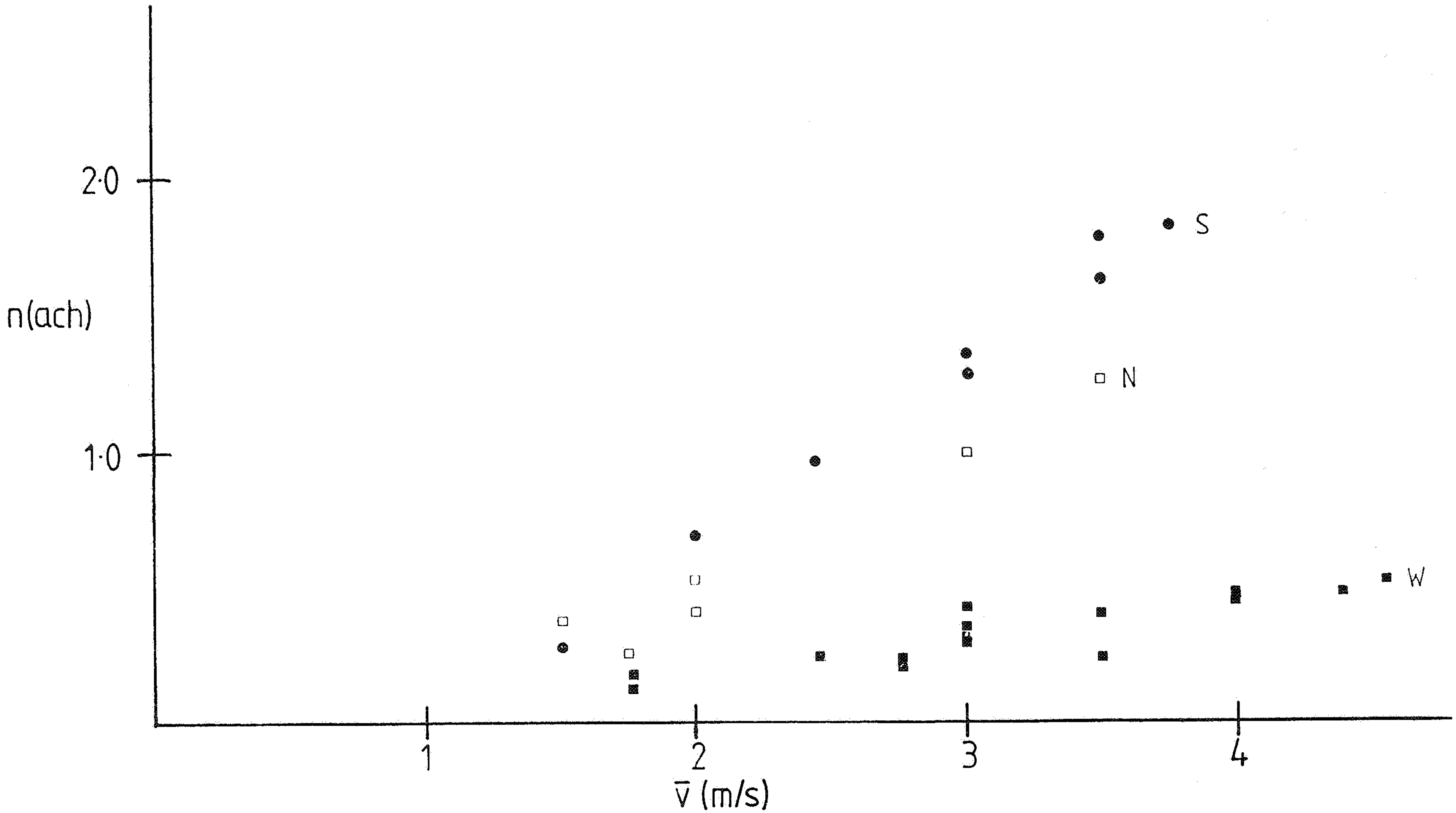


Figure 2.

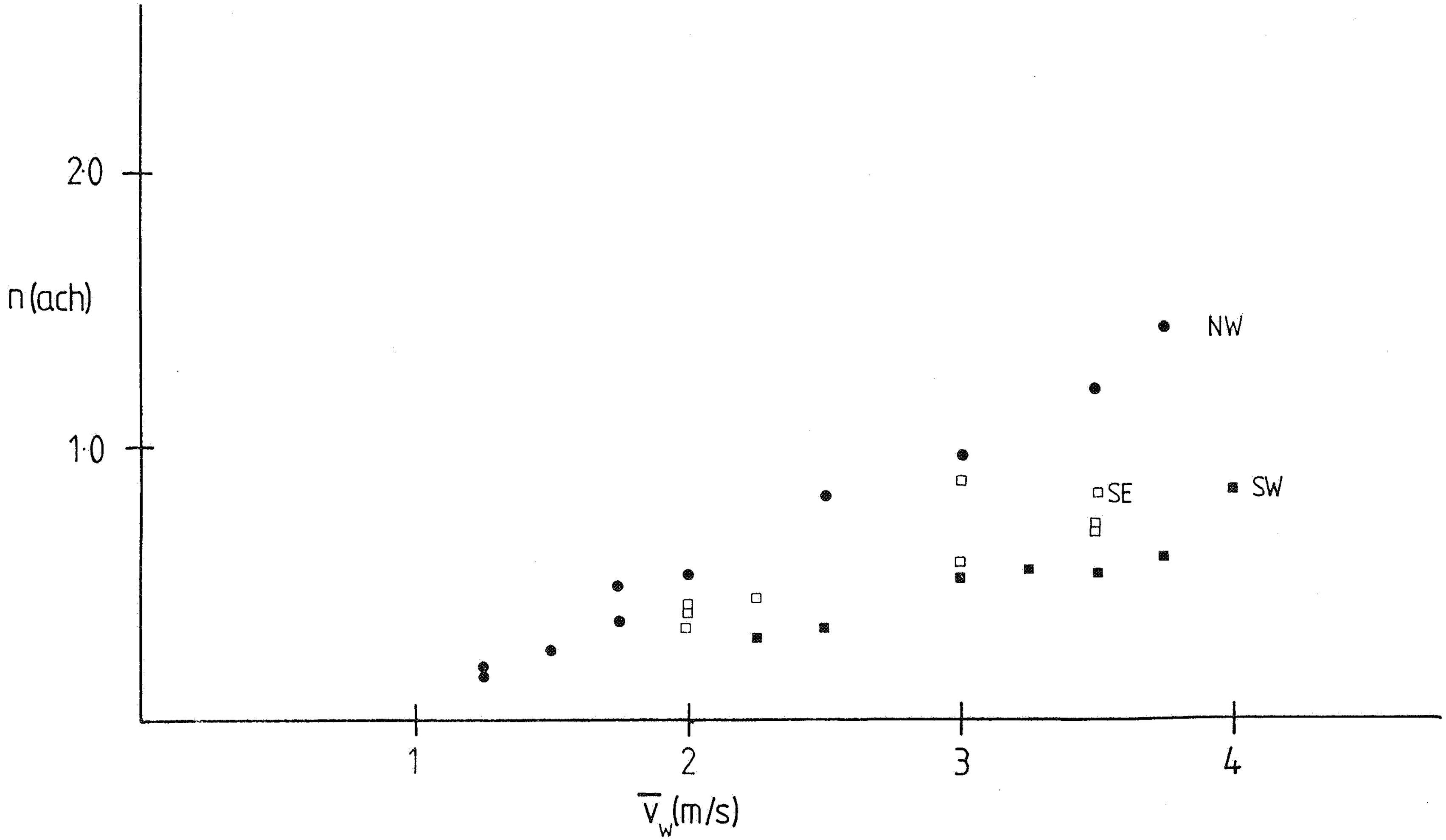


Figure 3.

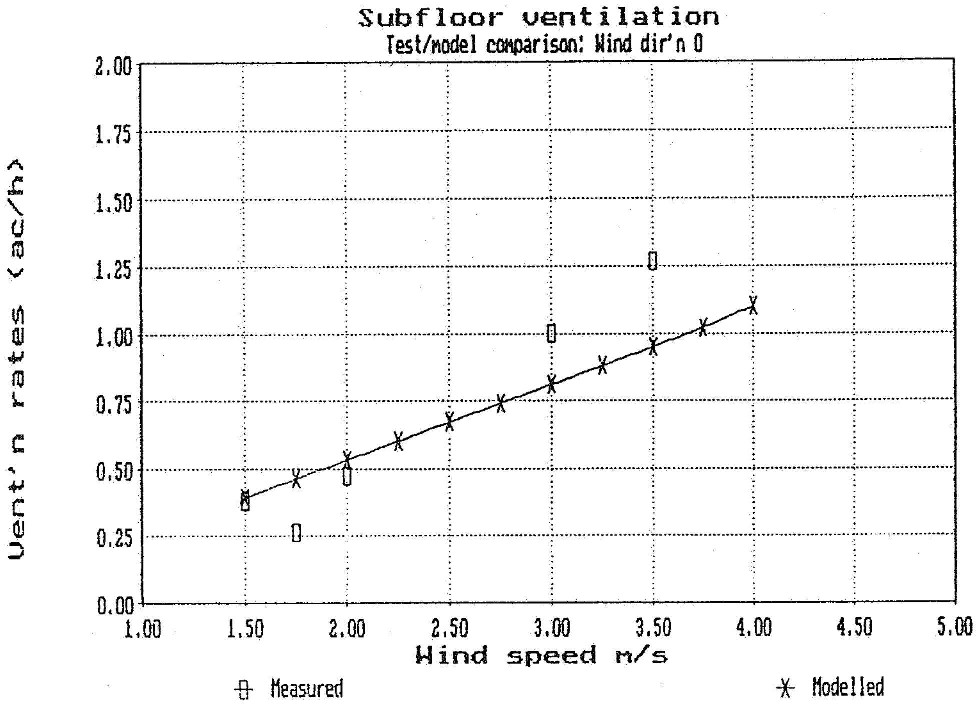


Figure 4.

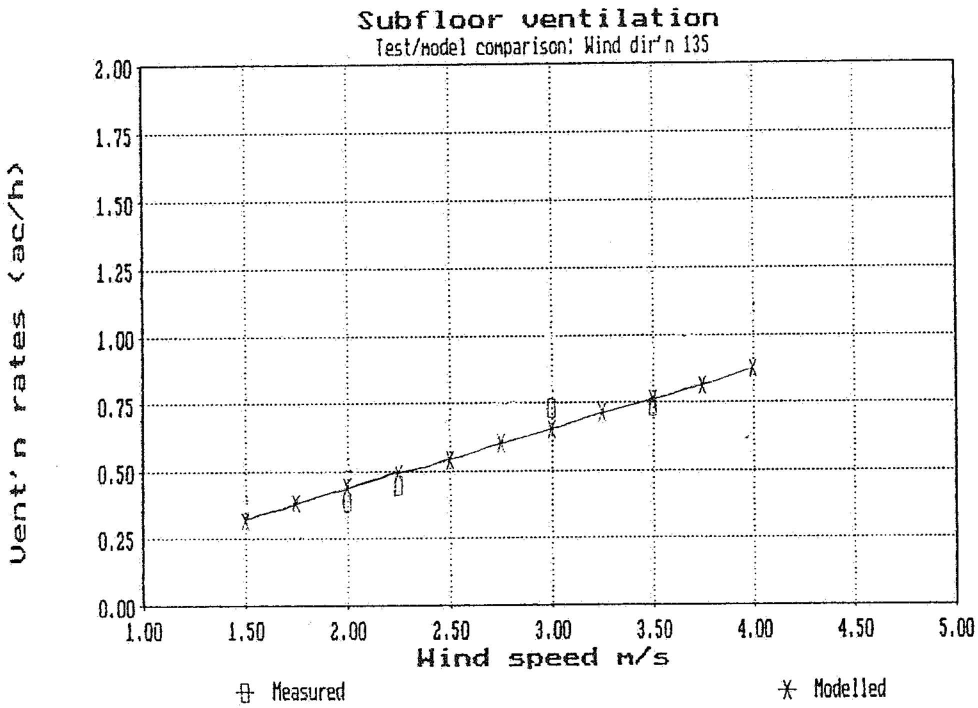
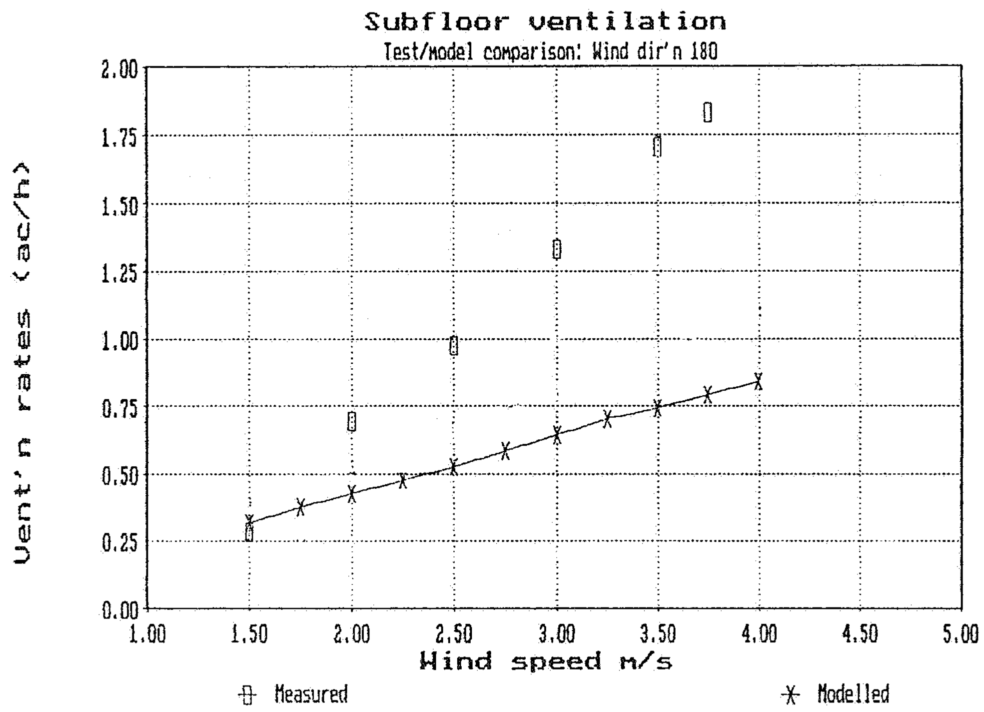


Figure 5.



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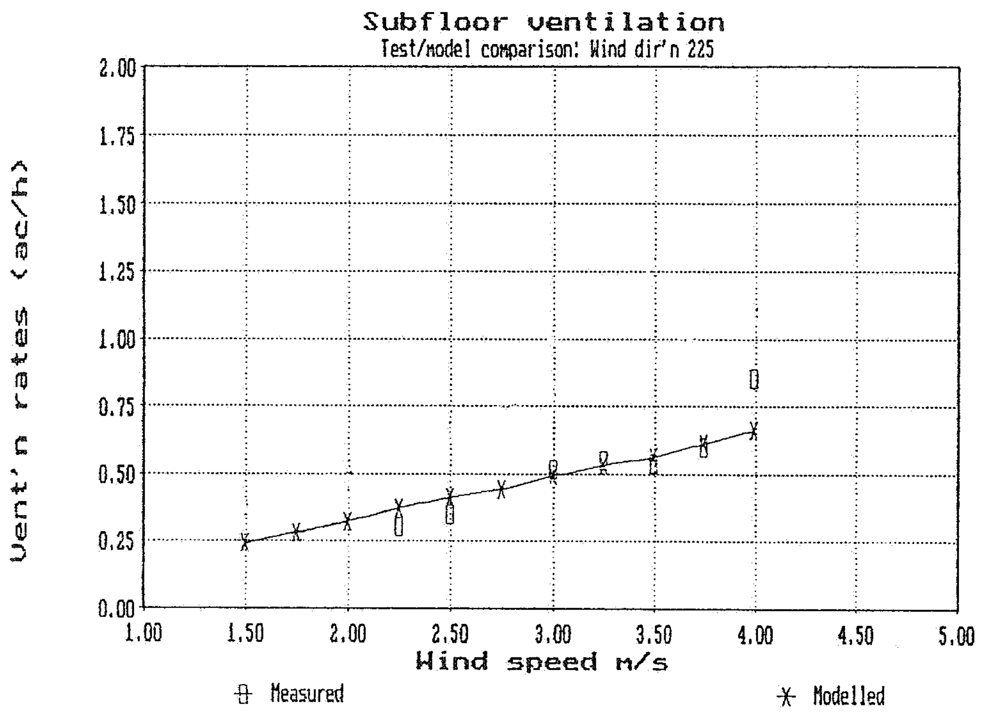

Figure 7.

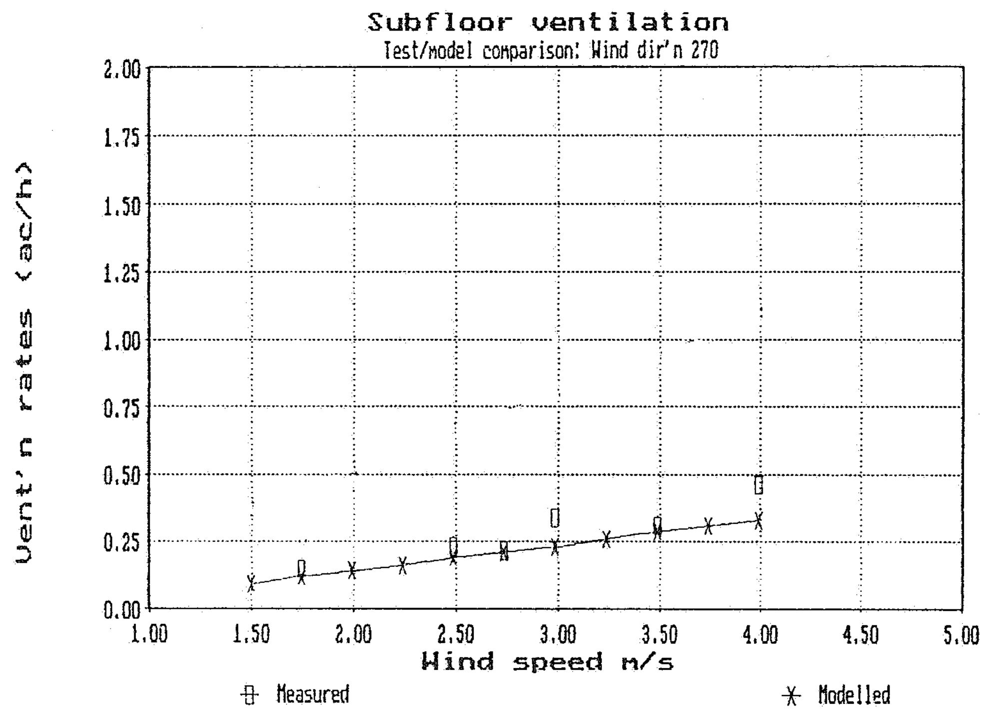

Figure 8.

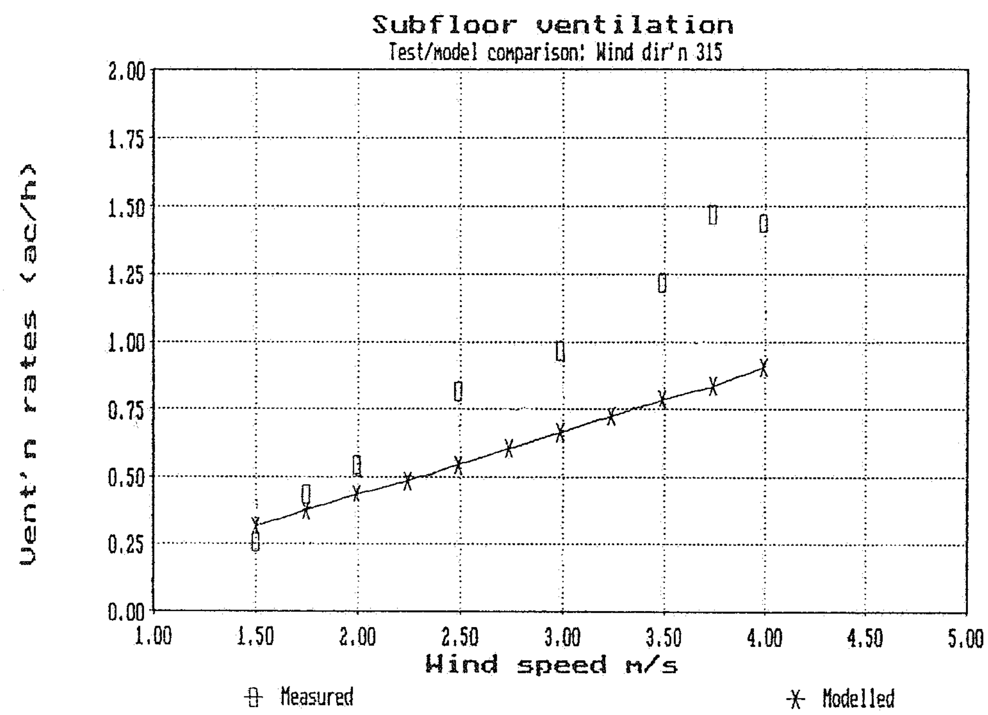

Figure 9.

Discussion
Paper 25

H Hens (Leuven, Belgium)

Groundfloor - first floor ventilation loop. Be careful that loop is established between living space and kitchen and sleeping rooms. Not only in-transfer but also a very active vapour flow and heat flow.

R Edwards (UMIST, UK)

We know.

H Hens (Leuven, UK)

Ventilation of crawl spaces - condensation problems in winter time? Reason: ventilation = lower temperatures, through that: refer RV, etc.

R Edwards (UMIST, UK)

The test was only a calibration test for the BREFAN programme.

VENTILATION SYSTEM PERFORMANCE

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Paper 26

PREDICTION OF THE PERFORMANCE OF VARIOUS STRATEGIES OF SUBFLOOR VENTILATION AS REMEDIAL ACTION FOR RADON PROBLEMS

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ABSTRACT

In order to reduce the convective flow which is the principal responsible for the high indoor ^{222}Rn concentrations, several mitigation technics have been developed and used in many countries. Since they don't always respond as expected, there is a need of instruments helping in their design and their evaluation. This paper suggests the use of a numerical code, based on the finite difference method, for the evaluation of ^{222}Rn mitigation strategies in dwellings. It is supposed that ^{222}Rn transport from soil into a dwelling occurs mainly by pressure-driven air-flow. The programme used calculates the pressure fields under the buildings, supposing a laminar air-flow in the soil and adopting the steady-state condition. Clear graphic outputs are delivered. The results of sample calculations are presented in order to illustrate the possibilities of the code. These calculations concern a house without basement, with an entry route for soil-gas : the floor-wall joint. A particular subslab depressurization system is included in the calculations. The code appears to be a powerful tool for the prediction and the evaluation of the performances of subfloor ventilation strategies.

1. INTRODUCTION

Radon (^{222}Rn) is a radioactive noble gas which decays by alpha-emission with a half-life of 3.8 days. It is the unique gaseous element (at normal temperature) of the ^{238}U natural radioactive decay family, whose elements are present all over the earth's crust with concentrations which vary as a function of the nature of the soil. The fact that ^{222}Rn is chemically unreactive under normal conditions implies that as soon as it is produced by the disintegration of radium (^{226}Ra, another element of the ^{238}U family), the ^{222}Rn atom is free to move away from its place of birth. In the soil, the fraction of radon atoms that succeeds in escaping to the exterior of the solid medium - the emanating fraction - will be mixed with the soil gas present in the pore space. Then, the ^{222}Rn atoms can move throughout the void space and, because of their relatively large half-life, a part of them will reach the soil-air interface. This transport of radon is related to two distinct mechanisms : molecular diffusion and forced flow (pressure-induced flow).

It is known that radon can be trapped in the interior of buildings, where it can reach elevated concentrations (much above the concentration outdoors) : ^{222}Rn produced in the soil, near the building foundations, migrates inside the building by a combination of the two mechanisms mentionned above. However, it is believed that forced air flow through the soil and across the building substructures is, in most cases, the principal responsible of the high indoor ^{222}Rn concentrations[1]. This forced flow is induced either by climatic conditions (such as wind and stack effects) and operational conditions (use of exhaust ventilation, HVAC system, ...) which create pressure fields favoring soil-gas infiltration through any opening (cracks, construction joints, ...) that connects the house, particularly the basement, with the underlying soil or rock. Other factors affecting these indoor radon concentrations are the radon production in soil and the physical characteristics of the soil, especially the permeability which affects the soil-gas entry rates[1, 2].

In order to reduce the flow of air through the building substructure, several techniques have been used during the last decade : sealing of entries, basement pressurization, soil depressurization, ...[3]. For example, it is well known that a fan-and-pipe system that depressurizes the layer of soil or gravel immediately below the substructure can reduce the entry of radon. However, it seems that a large number of such mitigation systems do not perform up to expectations, and there are many cases of houses which remain above the recommended radon level after subslab-depressurization[4].

So it is clear that there is a need of instruments helping in the design of radon-mitigation systems and in the evaluation of the performances of various strategies of subfloor ventilation as remedial action for radon problems.

We are presently involved in a CEC program dedicated to the modelling of the entry of radon from soil into dwellings, through the basement, to better perform remedial actions[5]. Within this framework, we need instruments allowing us to model the pressure field around the basements. In the present paper, we examine the possibility of using a three-dimensional finite difference code (which solves any kind of linear flow models)[6] to calculate pressure-driven flow rates in the ground, taking into account some environmental conditions (like soil permeabilities and outdoor-indoor pressure differences) and some structural conditions (cracks and joints in the basement, presence of a mitigation system, ..). The program we use runs on a PC and delivers very simple and useful graphical outputs (2 and 3 dimensions) so it could be particularly well suited for practical evaluations of mitigation systems performances.

2. MODELS OF RADON ENTRY FROM SOIL INTO RESIDENTIAL BASEMENTS BY PRESSURE-DRIVEN FLOW

As far as we know, much work has been done concerning the molecular diffusion transport mechanism (see e.g.[10], [11]), but only a few papers are devoted to the modelling of radon transport by pressure-driven air flow through the soil. Nevertheless, the approximation of a negligible diffusion is generally good, and the forced flow mechanism is believed to be the major factor responsible for high ^{222}Rn indoor concentrations. In fact, diffusion can be neglected if[1] :

$$\frac{k\,\Delta P}{\mu} \gg \frac{D_e}{\varepsilon} \qquad (1)$$

where :
k [m^2] is the soil permeability,
ΔP [Pa] is the driving pressure difference,
μ [Pa.s] is the dynamic viscosity of air,

ε is the soil porosity and
D_e [m^2/s] is the effective diffusion coefficient of radon in the soil pores.

In reference [1], W.W. Nazaroff applied a combination of analytical and numerical methods to the problem of computing ^{222}Rn transport by pressure-driven air flow from soil into a dwelling having a basement. The building was represented by a very idealised physical system. The steady-state ^{222}Rn transport in the soil was described by a set of equations expressing the conservation of air mass in the soil pores, Darcy's law, and the radon activity balance in the soil pores. It was shown that for small flow rates of air through the soil, the radon entry rate into the basement increases in proportion to the outdoor-indoor pressure difference ΔP at the soil level. For large flow rates, the entry rate increases with $\Delta P^{2/3}$. It was also shown that soil with ordinary radium (^{226}Ra) content can cause high indoor radon concentrations if it is even moderately permeable (say $k \sim 10^{-10}$ m^2), because of this pressure-driven air flow. Results of this work are essential for our understanding of the radon entry process. However, the method employed in order to calculate the pressure field in the soil pores is of a limited interest for practical extensions : it is an analytical calculation, so one needs to consider very simple and probably unrealistic geometries (basement, cracks, joints, ...).

T.A. Reddy *et al.*[4] addressed themselves to the general problem of modelling the pressure-induced air flow below the slab, in order to calculate pressure fields and to optimize the design of subslab depressurization systems. More specifically, they proposed a mathematical formulation for modelling the pressure field induced by a single suction point, with the particular hypothesis that air flows radially through a porous bed contained between two impermeable disks (one of them being the concrete slab, the other a very impermeable soil bed) centered at the suction point. The authors pointed out that the nature of the subslab air flow, under operation of mitigation systems, is related to the nature of the subslab medium : turbulent flow conditions will generally prevail through subslab gravel beds, whereas laminar flows are likely to occur in houses having soil (no gravel) below the concrete slab. They specially concentrate on a model predicting analytically the pressure field of turbulent flows in homogeneous circular porous beds (gravel) when suction is applied at the centre of the circle.The model coefficients of the pressure drop versus flow were determinated empirically for different types of gravels. For the laminar case, Darcy's law was used in the calculations. Here again, as in reference [1], the calculation of analytical solutions restricts the configurations for which pressure fields can be predicted. However, some of the authors are presently working to see how their simplified model and their empirical coefficients could be used by professional mitigators, for practical purposes, in a large number of situations[8].

In the work of C.O. Loureiro[7], numerical methods were used to solve the basic equations of a model which simulates the steady-state transport of radon from soil into houses with basements under constant negative pressures. The model simulates the generation and decay of radon within the soil, its transport throughout the soil due to diffusion and convection induced by the pressure disturbance applied at a crack in the basement, its entrance through the crack and the resultant indoor radon concentration. It supposes a steady-state condition, a fixed geometry (a house with a basement and a crack at the wall-floor joint in the basement), a constant indoor-outdoor negative pressure applied at the crack and a laminar flow through the soil (Darcy's law holds). Two three-dimensional finite difference computer programs were used to solve the mathematical equations of the model. It was concluded that the most important parameters involved in the transport of radon into the house are the soil permeability (k), the inside-outside pressure difference and the ^{226}Ra concentration in the soil particules. For a pressure difference of 5.0 Pa, it was shown that the entry rate of radon into the house was dominated by diffusion for $k \leq 1.0\ 10^{-12}\ m^2$ and by convective transport for $k \geq 1.0\ 10^{-12}\ m^2$, in which case the indoor radon concentration was found to be strongly dependent (almost linerarly) on the soil permeability. Among other results, we note that the indoor radon concentration was found to be directly (though not linearly) related to the pressure difference. The fact that the two programs of reference [7] used numerical methods for solving the basic equations of the model allows one to treat problems related to more realistic configurations (irregular boundary conditions, ...). However, these programs are related to the geometry considered by their author. To adapt them to other configurations (different crack geometries, presence of one or several subslab depressurization pipes with different possible geometries, ...) would be a hard task for professional mitigators who wants practical and rapid advices for the design and the evaluation of mitigation systems applied to each particular building type.

3. THREE-DIMENSIONAL FINITE DIFFERENCE CALCULATIONS

As it was said above, radon from the ground is the principal accountable for buildings high indoor radon concentrations. For moderate or high soil permeabilities (say $k \geq 10^{-12}$ m^2), the radon entry rate depends primarily on the possibilities of air entering the building from the soil, by convection, and it was shown that the indoor radon concentration depends directly on the indoor-outdoor pressure difference. So it seems reasonable, when evaluating the performance of a mitigation system, to concentrate on the effect of such a system on the air flow rates between the ground and the building and to calculate the pressure fields below the building, especially around any opening connecting it with the underlying soil. In order to perform such calculation, one needs to know the model relating the air flow per unit cross-sectional area to the pressure gradient. According to T.A. Reddy *et al.*[4], air flow should generally be laminar when soil (instead of gravel) is present under the slab. In this case, Darcy's law holds :

$$\vec{v} = -\frac{k}{\mu}\vec{\nabla}P \qquad (2)$$

where

$\vec{v}$ [m/s] is the Darcian velocity,

k [m^2] is the permeability,

μ [Pa.s] is the dynamic viscosity and

$\vec{\nabla}P$ [Pa/m] is the pressure gradient.

As this is a linear relation, we have decided to use the numerical code TRISCO[6], based on the finite difference method, which solves any kind of linear flow models (like three-dimensional heat transfer problems for example). This program runs on a PC-compatible under MS-DOS and achieves high performance, is user-friendly, provides high capacity (up to 60 000 nodes) and has a clear graphic output. An important merit of the code is the simple data structure, allowing one to describe even complex geometries in a fast and easy way. Note that the steady-state condition is adopted, for reasons of simplicity and reduction of computing costs; in fact, for most purposes, one can say that flows stabilise relatively quickly after a change in pressure[2].

As an input to the code, one has to formulate the problem, ie., describe the geometry (the object must be decomposed in homogeneous, beam shaped blocks), soil and material permeabilities and boundary conditions. One also has to select a grid of nodes, for discretisation purposes, which consists of perpendicular planes. Test calculations have been done for the following geometrical configuration : the soil block is represented as a parallelepiped in the centre and upper part of which there is a house, also considered as a small parallelepiped. This house has no basement and there is only one entry route for soil gas into the house : a gap located at the joint between the floor-slab and the wall, along all the perimeter of the slab. A subslab depressurization system is represented by a vertical pipe, through the slab, connected to an horizontal pipe situated just below the slab. The system is located at the center of the house floor. This configuration is presented in figure 1.

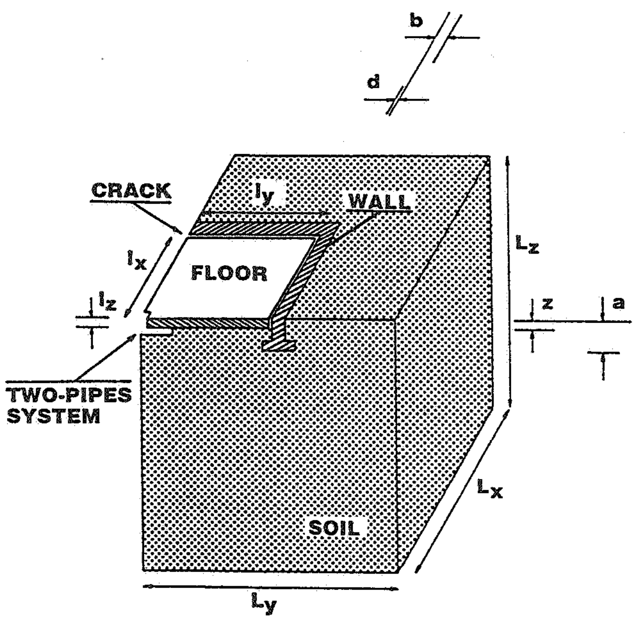


Figure 1 : Geometrical configuration adopted for the first set of calculations (see text)

In fact, only one quarter of the geometrical configuration is presented in the figure, because of the symetry in the plane parallel to the floor slab. Calculations have to be done only in this reduced configuration and are, in this case, greatly simplified. Of course, it is also possible to consider non-symetrical situations and a greater number of cracks and depressurization pipes. All those details can be defined to the program in a very friendly way. However, the number of nodes necessary to discretisate the problem and the calculation time will depend on the complexity of the configuration.

The calculations suppose implicitely that the soil is isotropic with respect to permeability. The pressure at the soil surface was assumed to be uniform and constant. The indoor pressure was also supposed uniform and constant, at a slightly lower value than the outdoor pressure. The same situation was assumed to hold for the two-pipes system, but with a greater pressure difference with regard to the outdoor pressure. Concerning the crack, it can be shown[9] that the air-flow through the crack is related to the pressure difference by the following expression :

$$\Delta P = A.Q + B.Q^2 \qquad (3)$$

where :
ΔP [Pa] is the pressure drop
Q [m^3/s] is the volume flow rate and
A and B are constants for a given crack.

With values of A and B given in the work of P.H. Baker *et al.*[9] for the crack geometry considered here it appears that, for the typical pressures of our problem, the evolution of Q with ΔP (predicted by expression (3)) can be modeled by a linear relation similar to expression (2). So it is also possible to define a permeability for the crack, which allow the program to calculate the pressure field inside the crack.

Table 1 presents the soil and building parameters used in the first set of calculations. During these calculations the entire soil block was supposed homogeneous with respect to a fixed value of the permeability (see table 1). Figures 2 and 3 present the results of these calculations for a difference of -50.0 Pa (fig. 2) and -100.0 Pa (fig. 3) between the outdoor pressure and the two-pipes system pressure. These figures illustrate some of the output possibilities of the code. The solid isobar line corresponds to -3.0 Pa (the difference between outdoor and indoor pressures). A comparison of figures 2 and 3 immediately indicates that for the considered mitigation system, a pressure difference of -50.0 Pa is not sufficient to ensure that the soil immediately below the floor-slab is at a lower

Soil permeability	10^{-10} m^2
Crack permeability	6.2 10^{-8} m^2
Floor and wall permeability	10^{-15} m^2
Pressure difference (building)	-3.0 Pa
Pressure difference (two-pipes system)	-50.0 Pa or -100.0 Pa
Soil block dimensions	L_x = 9.62 m L_y = 10.70 m L_z = 5.0 m
Floor-slab dimensions	l_x = 4.22 m l_y = 5.30 m l_z = 0.15 m
Wall characteristics	a = 0.80 m b = 0.30 m
Crack depth	z = l_z = 150.0 mm
Crack with	d = 1.0 mm
Vertical pipe dimensions (mm)	70 x 70 x z
Horizontal pipe dimensions (mm)	70 x 140 x 1200

Table 1 - Soil and building parameters used in the first set of calculations (see also figure 1)

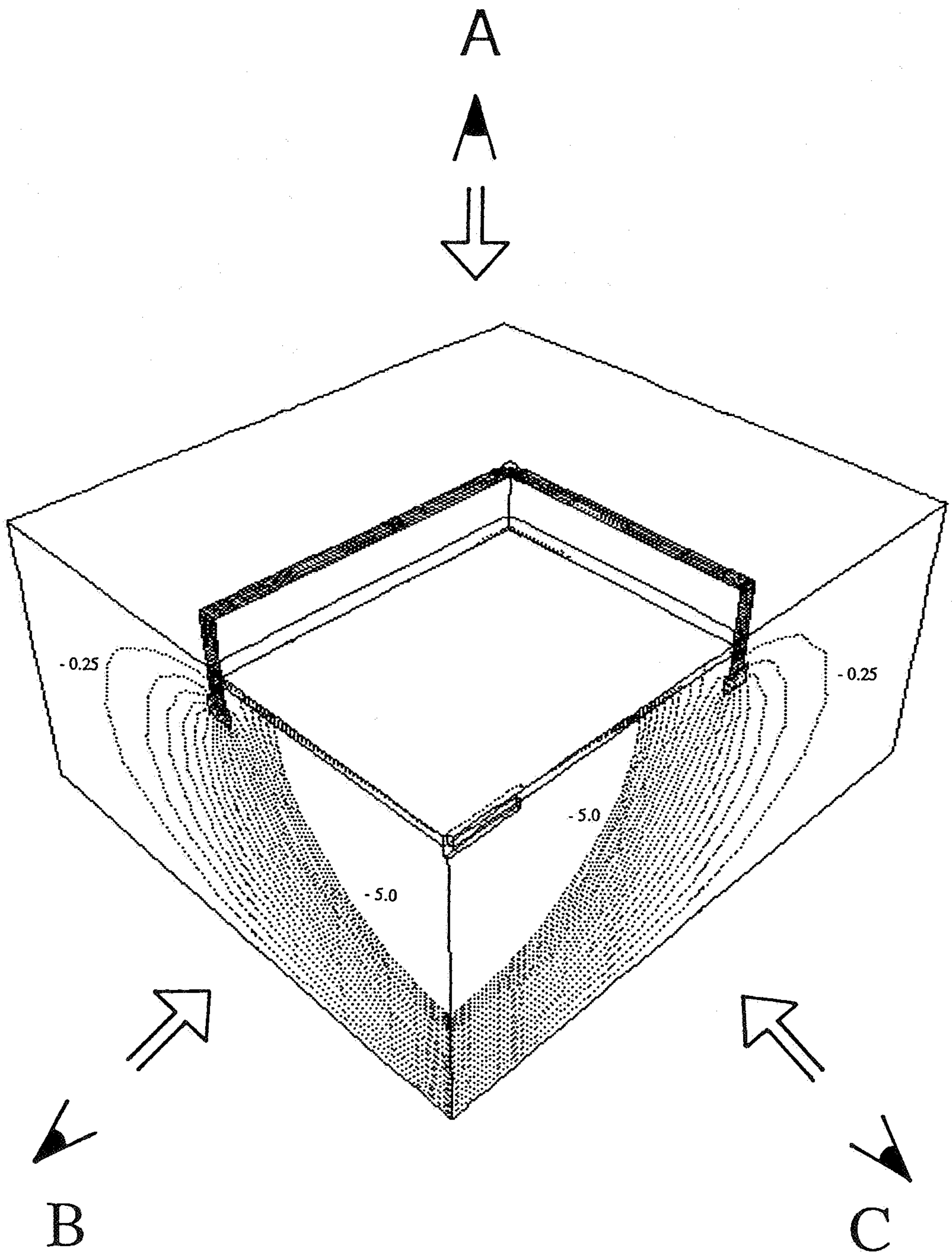


Figure 2.a. : Isobars (Pa) in the soil for a pressure difference (two-pipes system) equal to -50 Pa : three-dimensional view of the configuration (see table 1 for the soil and building parameters used in the calculation). For reasons of clarity only the isobars between -5.0 Pa and 0.0 Pa are presented, with a step of 0.25 Pa.

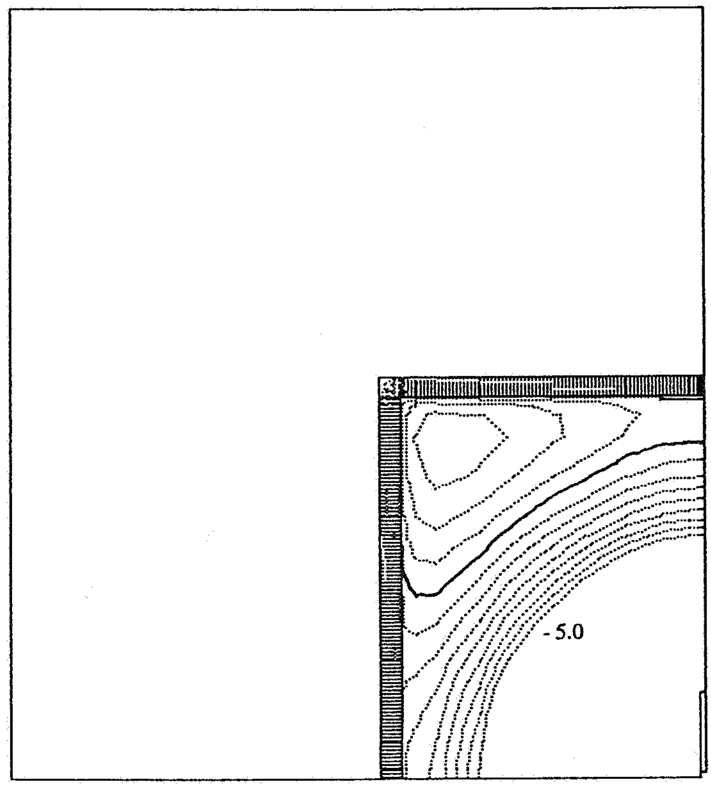


Figure 2.b. : Isobars : view A (immediately below the floor-slab, see fig. 2.a.)

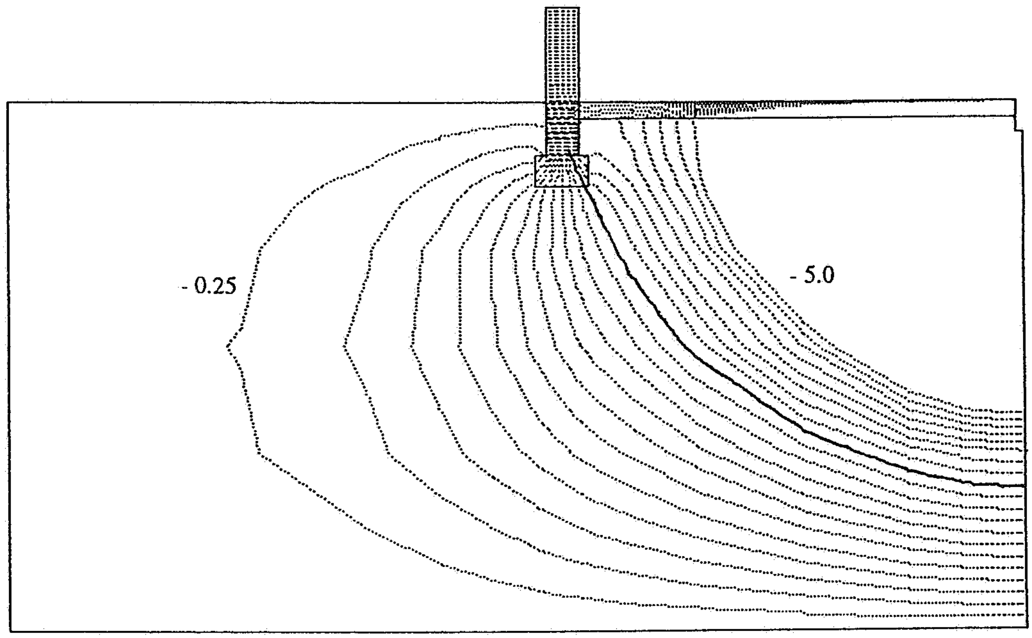


Figure 2.c. : Isobars : view B (see fig. 2.a)

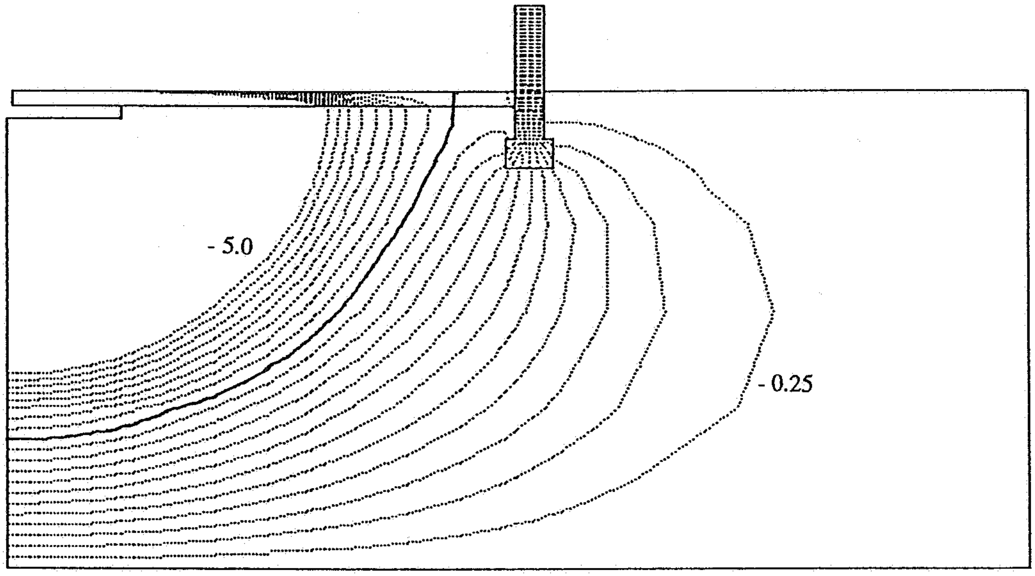
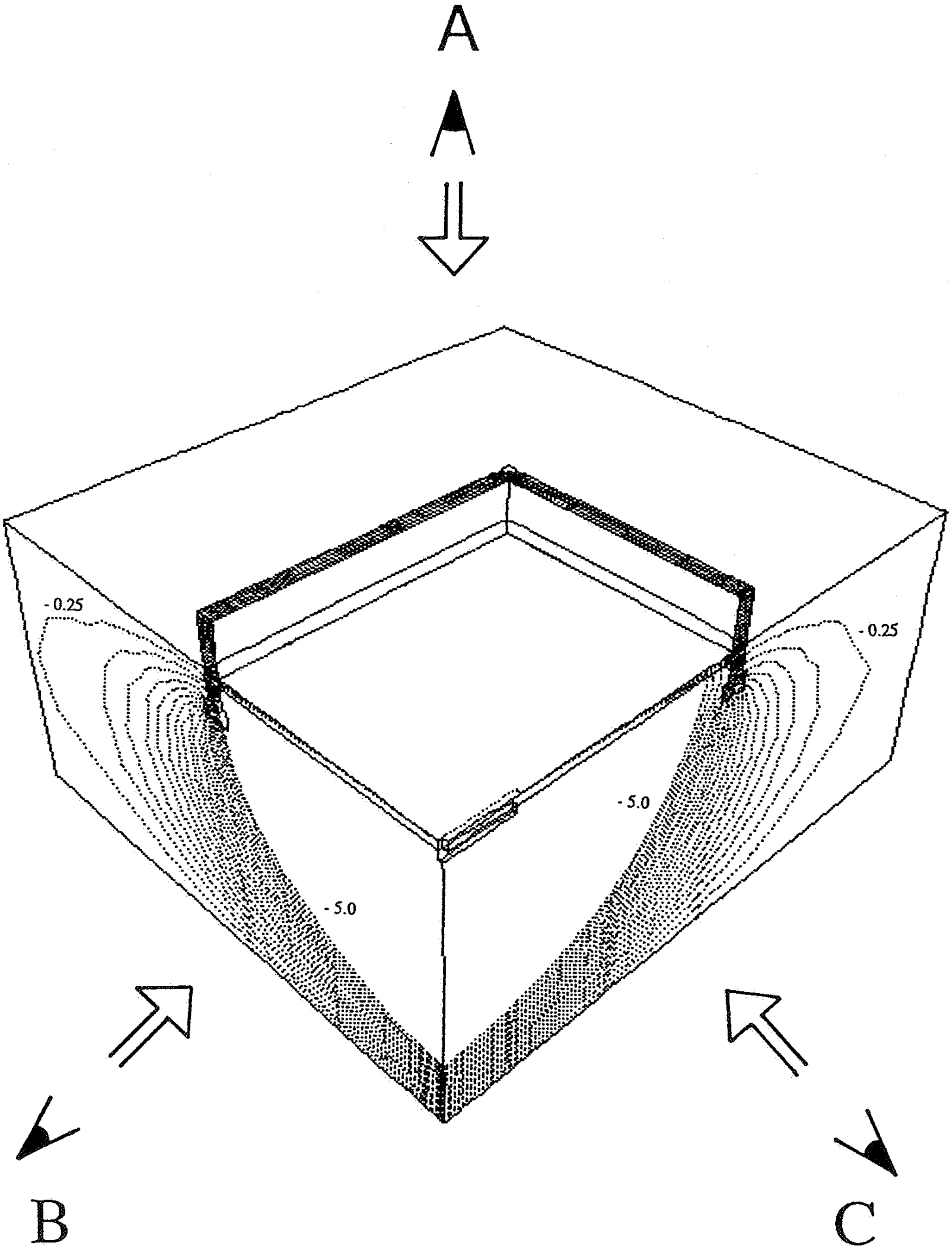


Figure 2.d. : Isobars : view C (see fig. 2.a.)



<u>Figure 3.a.</u> : Isobars (Pa) in the soil for a pressure difference (two-pipes system) equal to -100 Pa : three-dimensional view of the configuration (see table 1 for the soil and building parameters used in the calculation). For reasons of clarity only the isobars between -5.0 Pa and 0.0 Pa are presented, with a step of 0.25 Pa.

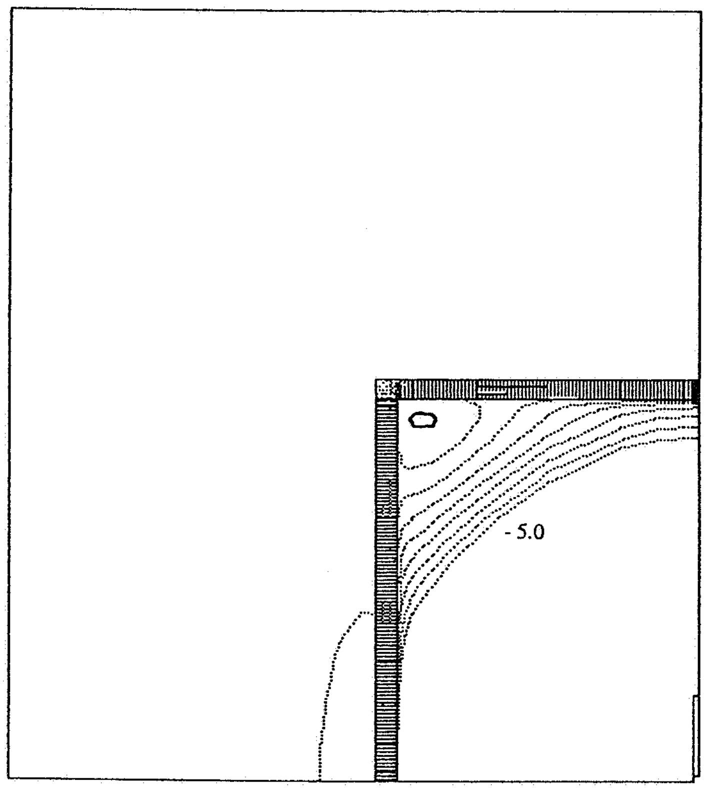


Figure 3.b. : Isobars : view A (immediately below the floor-slab, see fig. 3.a.)

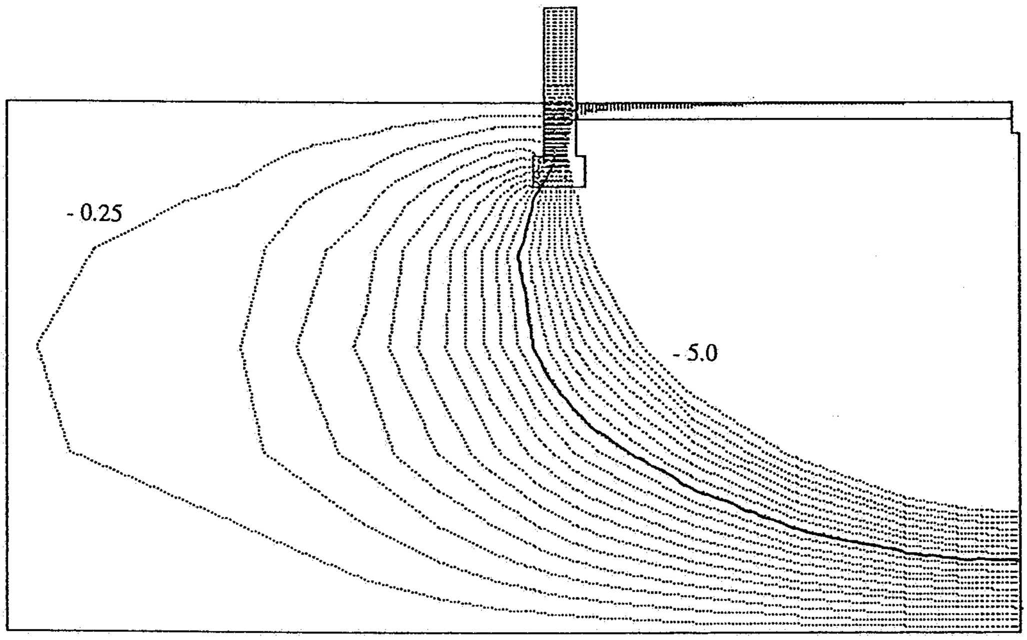


Figure 3.c. : Isobars : view B (see fig. 3.a)

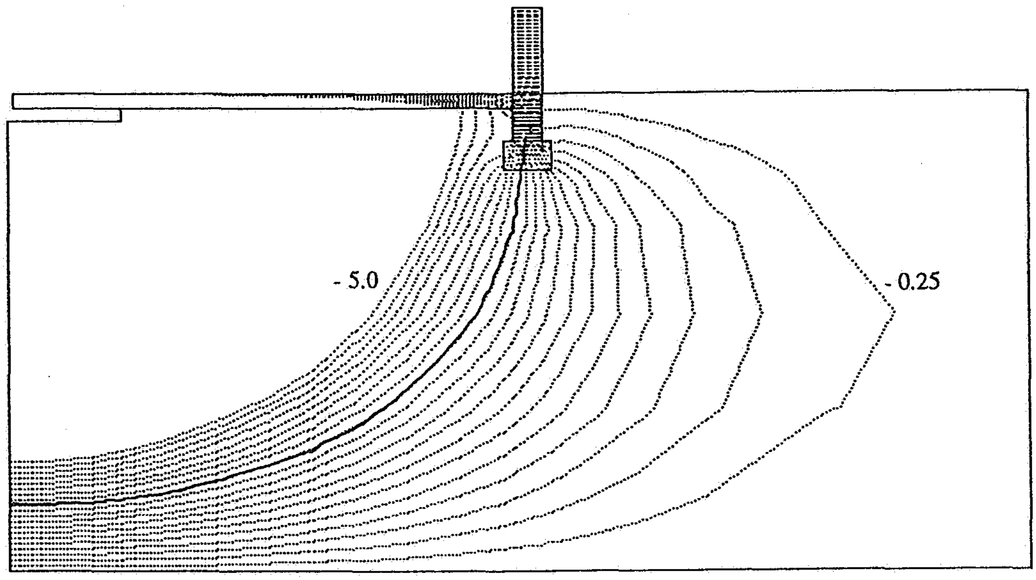


Figure 3.d. : Isobars : view C (see fig. 3.a.)

pressure than the indoor pressure, especially around the corners. Figure 3 indicates that if the pressure difference is set to -100.0 Pa, the situation looks much better from the point of view of the ability of the mitigation system to avoid the penetration of the soil-gas inside the house. Of course, an alternative solution would be to adopt another mitigation system : several suction points located closer to the corners for example. All those alternative mitigation strategies may be easily simulated and evaluated with the code TRISCO.

In figures 4 and 5 we present another calculation illustrating the possibilities of the programme. For this second calculation, we have adopted the same geometrical configuration and the same mitigation system than for the first set of calculations. However, it is supposed that a layer of gravel is present just below the floor-slab. This layer is separated from the soil by a plastic foil located just below the horizontal pipe of the mitigation system. Figure 4 illustrates this situation. All the parameters were set as in table 1, and the difference between the outdoor pressure and the two-pipes system pressure was fixed to -100.0 Pa. The permeability of the gravel was supposed to be ten times greater than the soil permeability. We must stress that it may be delicate to consider much greater values for the gravel permeability because of the implicit hypothesis of a laminar flow adopted by the programme. For pratical purposes, the plastic foil was supposed to be one centimeter thick, with a permeability equal to 10^{-12} m^2.

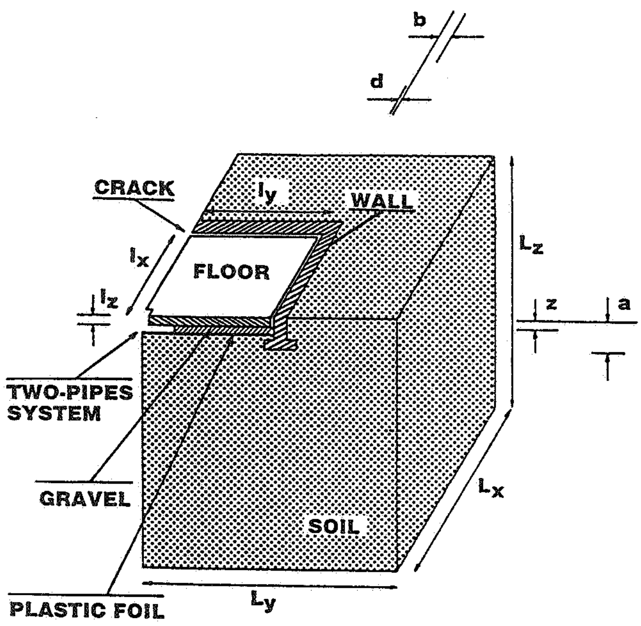


Figure 4 : Geometrical configuration adopted for the second calculation (see text)

Figure 5 presents the results of this calculation. The solid isobar line corresponds to -3.0 Pa. It appears, as expected, that the situation looks even better than for the case presented in figure 3 from the point of view of the performances of the mitigation strategy.

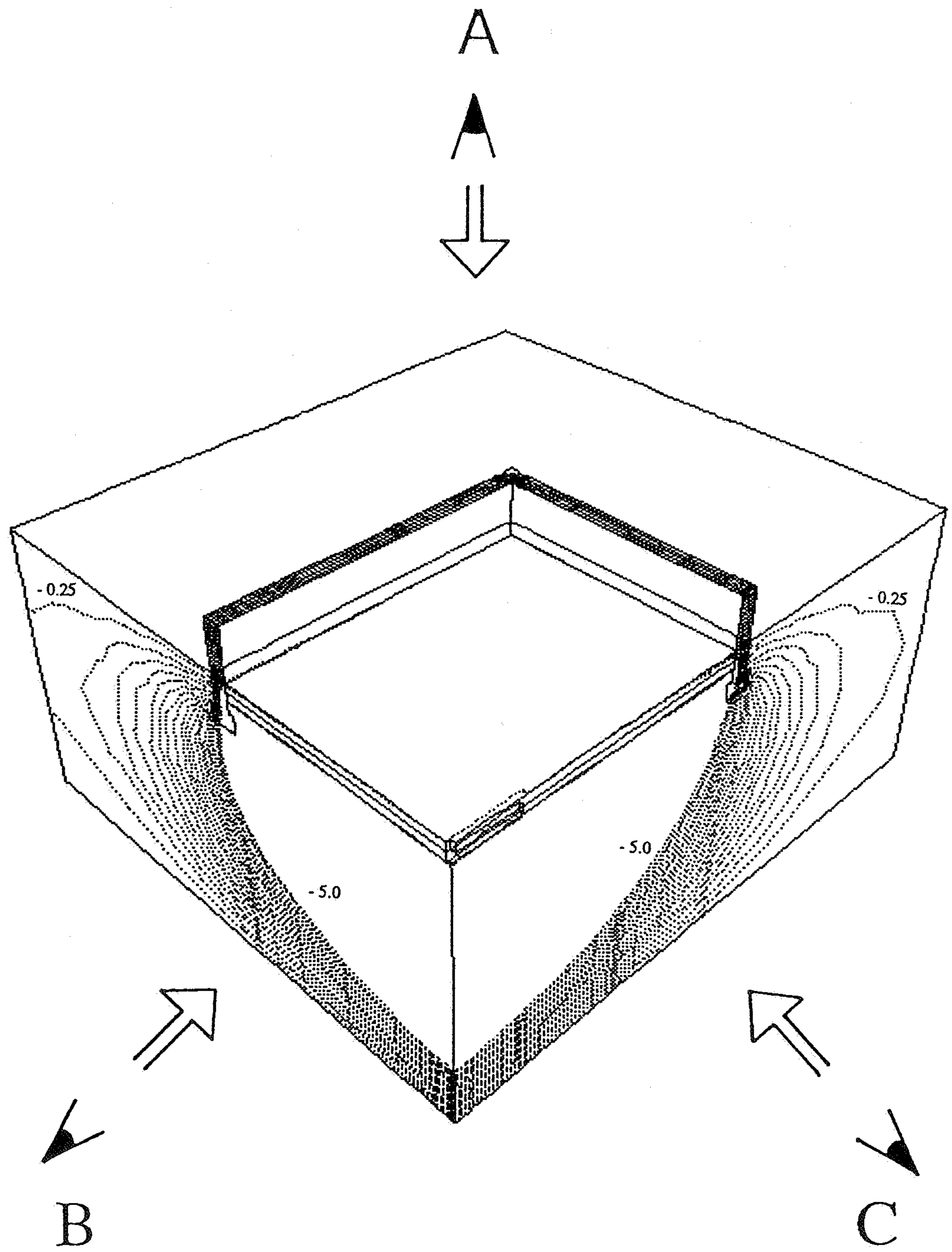


Figure 5.a. : Isobars (Pa) in the soil for a pressure difference (two pipes system) equal to -100 Pa : three-dimensional view of the configuration (see text for the soil and building parameters used in the calculation). For reasons of clarity only the isobars between -5.0 Pa and 0.0 Pa are presented, with a step of 0.25 Pa.

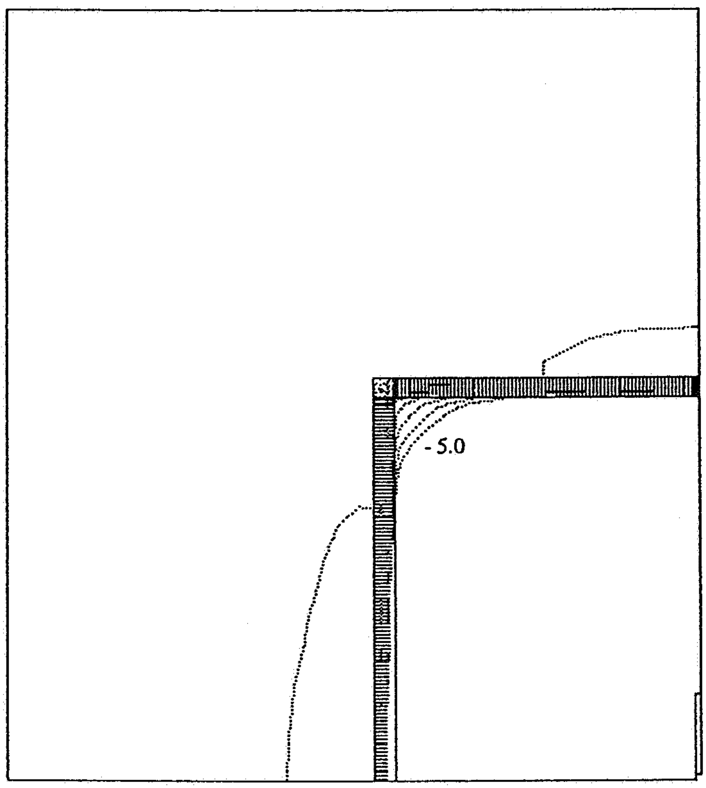


Figure 5.b. : Isobars : view A (immediately below the floor-slab, see fig. 5.a.)

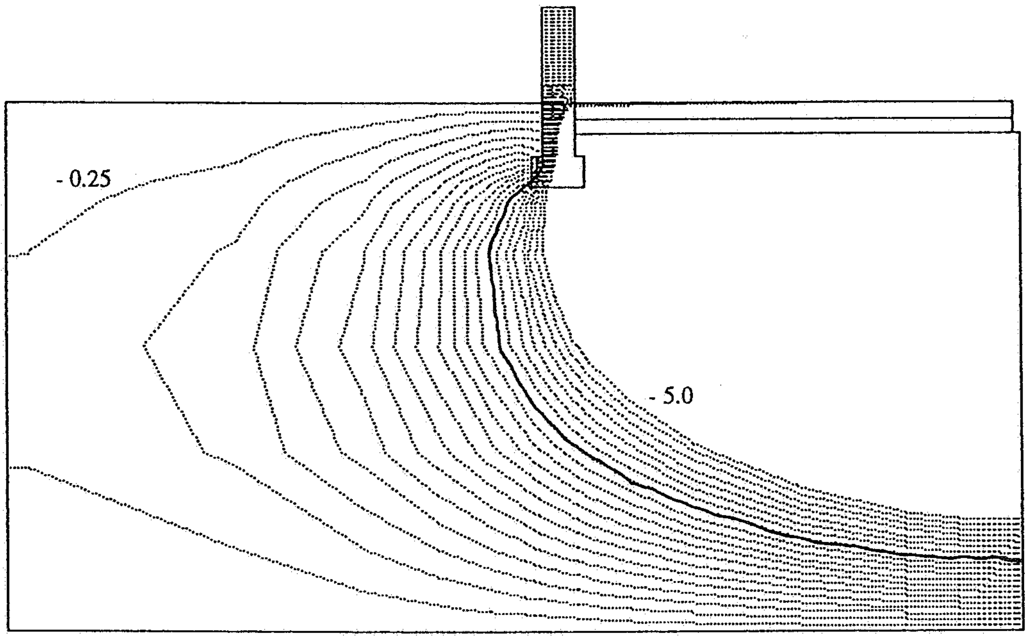


Figure 5.c. : Isobars : view B (see fig. 5.a)

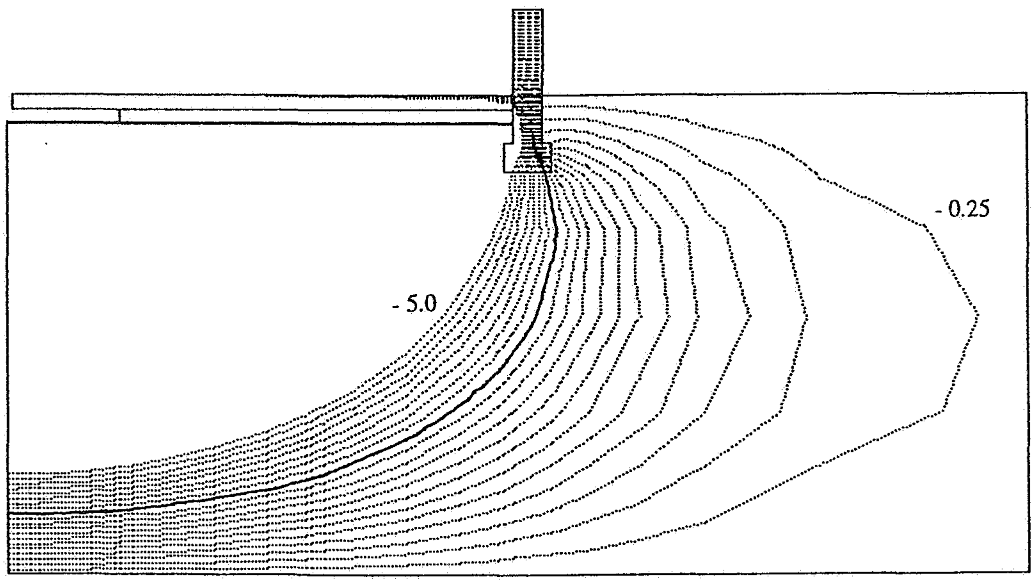


Figure 5.d. : Isobars : view C (see fig. 5.a.)

The calculation time depends on the complexity of the configuration. For cases similar to those presented here, typical calculation times are 10 hours for a 80486/33MHz system and around 100 hours for a 80286/10MHz system.

4. SUMMARY

There is a need for instruments helping in the design and the evaluation of radon mitigation systems in dwellings. This paper suggests the use of a three-dimensional computer code, based on the finite difference method, to perform such evaluations.

-With the hypothesis that the pressure-driven air flow through the soil is the major factor responsible for high ^{222}Rn indoor concentrations, the used programme calculates the pressure fields under the floor-slab. It allows the control of the ability of mitigation systems to avoid the penetration of the soil-gas into the house.

The sample calculations presented in the paper illustrate the possibilities of the programme. Because one can take into account all the details of the configuration in a very simple way, and because of the simple and useful graphical outputs (2 and 3 dimensions), the code appears to be particularly well suited for practical evaluations of mitigation systems performances.

As it is implicitly supposed that the linear Darcy's law holds, it has to be kept in mind that a particular hypothesis is made about the nature of the air flow under the slab : it is supposed to be laminar. This is the case when soil with not too large permeabilities (say $k < 10^{-9}$ m^2) is present under the slab. However, it is believed that this is not a too serious limitation to the usefulness of the proposed approach, because the cases for which the design of a mitigation system may be the most delicate are probably those for which soil (instead of gravel) is present under the slab.

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VENTILATION SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 27

DESIGN OF VENTILATION SYSTEMS IN RESIDENTIAL BUILDINGS

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ABSTRACT

Building regulations prevailing in France made it compulsory to use specific ventilation systems in new residential buildings since 1969. Different kind of ventilation systems (mechanically powered, temperature driven, hybrid systems,...) have been developed. This paper reviews these systems and outlines their advantages and drawbacks in single-family or multi-family buildings, with respect to architectural flexibility and comfort or safety requirements. In a second part, the paper introduces and discusses the scientific grounds (for instance, wind effect) which govern the ventilation rates. Results of theoretical work are given and design methods in use in France, with their influence on architecture conception, are featured.

1 - INTRODUCTION

Ventilation systems in residential buildings are primarily intended to remove air pollutants (moisture, combustion products, odours,...) and prevent building damage due to condensations and mould growth. This was previously achieved by natural means : window opening and also air change when fireplaces are working. In the sixties, new architectural trends appeared : water-closets and bathrooms are often in a central position with no window, and fireplaces have almost disappeared in multi-family houses. On the other hand, greater concern has been paid to energy conservation and comfort matters (protection against outdoor noise and cold draught). This has led to the spread of specific ventilation systems such as : natural ventilation using vertical shafts, mechanical systems ; or new techniques such as : humidity controlled ventilation. The usual air flow pattern, using such systems, is shown in fig 1.

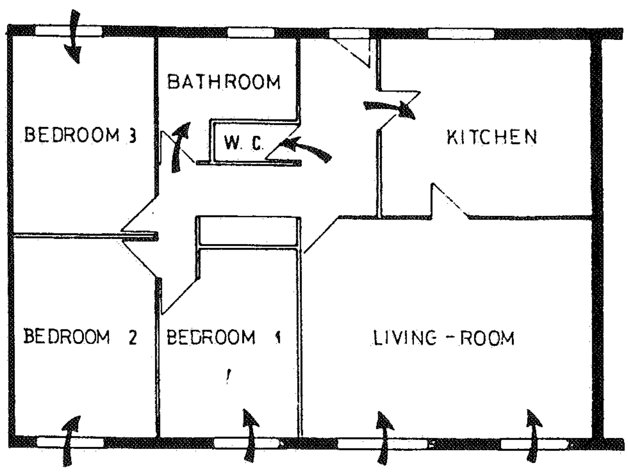

Fig. 1 : *Air flow pattern in dwellings :*
Since service rooms are ventilated by exhaust vents, there are no air inlets in these rooms.

These systems must comply with additional requirements :

. limit heat losses to a reasonable value, independent of climatic conditions (wind, outdoor temperature) ;

. prevent acoustic annoyance, due either to noise emission or noise transmission through ventilation components ;

. prevent reverse flow through a common ventilation network ;

. design and install the air inlets in order to avoid cold draughts ;

. avoid drift of components characteristics due to fouling or aging ;

. allow access for maintenance operations and provide safety in case of vented or unvented combustion appliances.

2 – VENTILATION SYSTEMS IN USE IN FRANCE

Mechanical exhaust systems which appeared in France in the sixties are now the most commonly used systems. As opposed to natural ventilation, they provide a much greater flow rate steadiness, whatever the outdoor climatic conditions. Mechanical systems have been improved in recent years. New systems, with flow rates depending on the prevailing indoor climatic conditions, enable energy conservation and indoor air quality to be further improved.

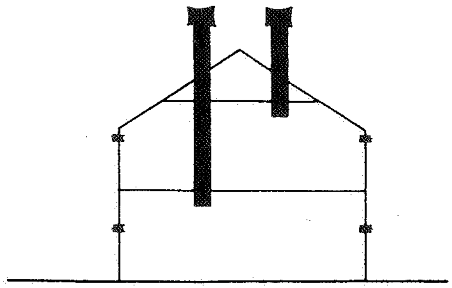

Fig. 2 : *Scheme of a natural ventilation system a single-family house.*

In natural exhaust systems (fig.2) the flow rate is generated by buoyancy and wind effects. Accordingly, flow rates vary a lot. Other systems (mechanical balanced systems,...) are also in use and many new systems (humidity controlled vents, hybrid systems combining natural and mechanical effect,...) have been recently developed.

3 – VENTILATION SYSTEMS DESIGN AND DIMENSIONING

3.1 – Dimensioning of air inlets

Air inlet sizing must lead to a good balance between air quality and energy conservation requirements. As a matter of fact, air inlets are intended to ensure as equal as possible fresh air distribution amidst the different rooms. Therefore, their sizing must be sufficient with regard to non-uniformity of flow due to building shell leakage : fig. 3 depicts the influence of air leakage increase on fresh air flow in a room.

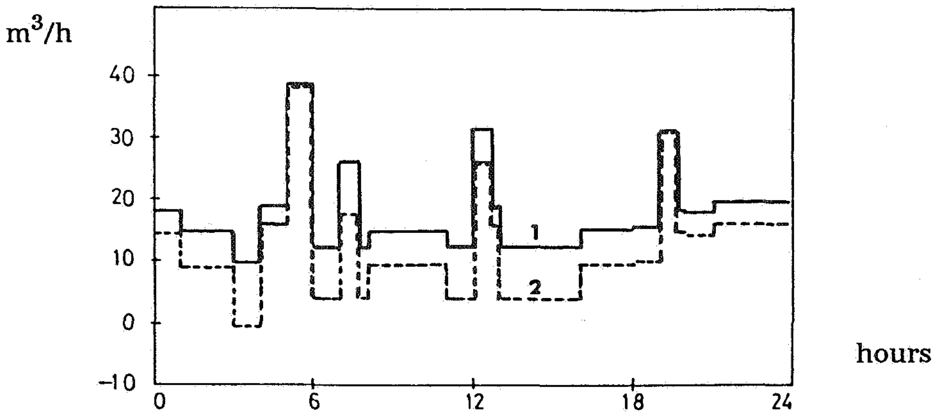

Fig. 3 : *Influence of building shell air leakage on air flow pattern*

The curves represent the calculated values of fresh air entering a bedroom (cf fig. 1) according two different assumptions :

Curve 1 : *external walls of the kitchen are assumed to be perfectly airtight.*

Curve 2 : *air leakage value of the kitchen external walls is assumed to be equal to 20 m^3/h under 1 Pa pressure difference.*

On the other hand, air inlets size must be as low as possible, in order to reduce heat losses due to cross ventilation in case of wind. Calculations [1] have shown that cross ventilation heat losses are a function of the negative pressure inside the dwelling, according to the following equation :

$$Q_S = P. \frac{e}{1 + d . \left(\frac{Q}{P}\right)^2}$$

where :

Q_S (m^3/h)	flow rate corresponding to cross ventilation
Q (m^3/h)	exhaust flow rate due to ventilation system operation
P (m^3/h)	flow rate through the building shell air leakage and air inlets when the pressure difference is 1 Pa
e, d	numerical coefficients, the values of which depend on building wind exposure.

Taking into account the mean air leakage values of the building shell in new buildings, the above considerations lead to a proposal that the area of air inlets should be equal to 30 cm^2 in each room, when exhaust flow rate is achieved by a mechanical systems. When a natural exhaust system is used, the air inlet area must be higher.

3.2 – Sizing of mechanical exhaust network

fig. 4 depicts an example of an air flow calculation in an exhaust network. Air flows are mainly depending on the pressure-flow rate curves of the fan, exhaust vents and network itself. However, the negative pressure in the network, (hence the flow rates at each exhaust vent), may vary considerably. As a matter of fact, thermal or wind effect may substantially increase the flow rates. Moreover, according to French regulation, it is allowed that exhaust vent aperture area may be set by the occupant either to a low value, or to a peak value. When most of the exhaust vents are at their peak value, pressure drop in the ducts increase and exhaust flow rates in other vents are lower.

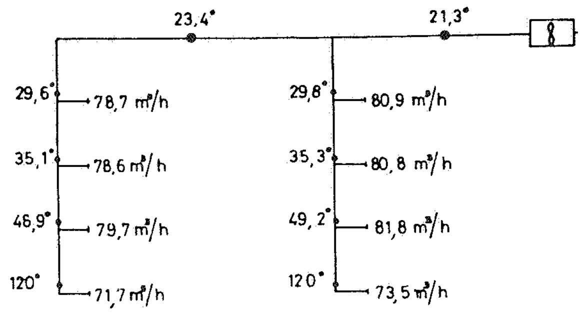

Fig. 4 : *Mechanical exhaust network* :

this figure depicts calculated flow rates and temperatures [6] in an exhaust network. The exhaust vents of the first floor are linked to vented gas appliances, the flue temperature of which is assumed to be equal to 120°C.

The pressure variation inside the network may be, as illustrated in fig. 5, quite important and cause either noise production, when the pressure is high, or unsufficient ventilation, when this pressure is low. The allowed values of noise production (acoustic pressure level) in the kitchen is 35 dB(A), which can be usually achieved, provided the negative pressure in the network does not exceed a value which usually is in the range of 120 to 140 Pa. On the other hand, the pressure must be high enough to limit flow rate variations due to wind or thermal effect. Moreover, the exhaust vent aperture area must not be too large in order to reduce noise transmission from one flat to another one.

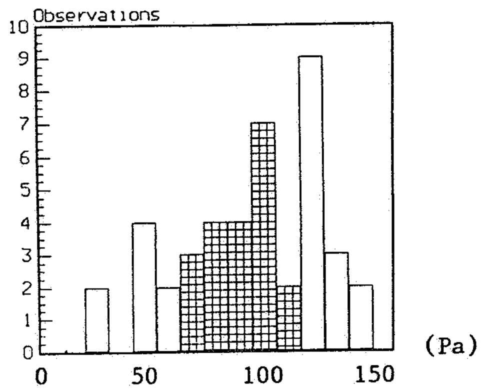

Fig. 5 : *Pressure variation in a mechanical exhaust network :*

this histogramme shows the distribution of negative pressure which were measured in an exhaust mechanical system. Crossed lines indicate the range of pressure for which air flow and acoustic requirements of the exhaust vent are met. A large number of measured values are not in the authorized range ; the sizing of this particular exhaust network must therefore be improved.

For all these reasons, a typical pressure range is 70 - 120 Pa. Most of the exhaust vents in use in France are designed to meet the flow rate steadiness and noise requirements in this range of pressure.

The design principle of mechanical exhaust systems in multi-family buildings is therefore to dimension the network in such a way that the pressure drop of exhaust vents remains in the authorized range of values, regardless of the climatic data and the occupant behaviour. This is generally achieved either by using purpose-designed fans or, preferably, in using ducts with sufficiently large diameters, which allow accommodation of the pressure drop variation due to human behaviour (i.e : variation of required exhaust flow rate). A design method [3], based on this principle, is under preparation and will hopefully help designers improve exhaust network sizing.

3.3 Dimensioning of natural ventilation systems

Proper dimensioning of natural ventilation system is a delicate matter, because the air flow is dependent, to a great extent, on the outdoor temperature and also on the wind effect, which still needs a lot of investigation. The need for more research can be illustrated by considering the example of shunt ducts which were in common use in multi-family buildings in the sixties : As it appears in fig. 6, the shunt duct is a double duct composed of a small duct used for individual air exhaust of each dwelling and a larger one used to collect individual air flows up to the top, where a cowl is installed to transform wind velocity into pressure head. From an architectural point of view, the shunt duct makes it possible, compared with individual ducts, to save space and decrease the building cost. On the other hand, it may lead to higher likelihood of reverse flow [4].

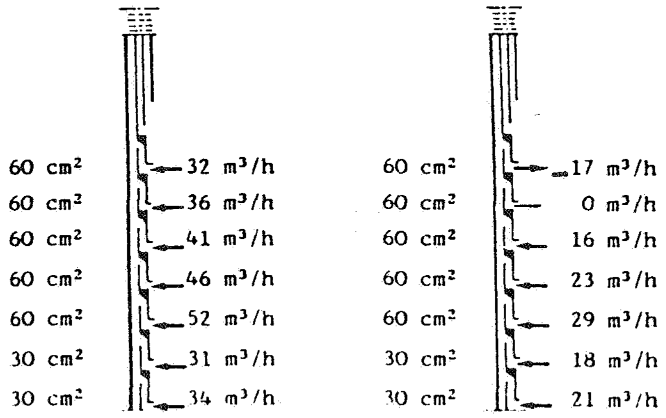

Fig. 6 : *Shunt ducts : influence of wind velocity and outdoor temperature on exhaust flow rates.*
first column : outdoor temperature : 0°C ; wind velocity : 0 m/s
second column : outdoor temperature : 10°C ; wind pressure : -3 Pa

It may be noticed that reverse flow rates occur under unfavourable climatic conditions. These reverse flow rates may be alleviated using high performance cowls. A practical difficulty is that, for a given cowl, the negative pressure caused by the wind is a complex function of wind velocity and air flow rate, thus making design calculation difficult to handle. Gonzalez [5] investigated that point, and showed that a relationship exist between two non-dimensional quantities :

$$X = \frac{DP}{1/2.\rho e.V^2} \qquad Y = \frac{1/2.\rho i.U^2}{1/2.\rho i.V^2}$$

Where :

DP (Pa)	:	difference between total pressure at the cowl inlet and outdoor static pressure
U (m/s)	:	air velocity in the duct
V (m/s)	:	wind velocity in the cowl neighbourhood
ρ i, ρe (kg/m^3)	:	volumic mass of indoor and outdoor air

The simplest way of formulating the relationship between X an Y is to assume that suction effect due to wind and pressure drop due to air flow are independent, which leads to :

$$X = \frac{Y}{A} + C \qquad (3)$$

It may be observed (fig.7) that the agreement between this formula and experimental points is far to be good. Thus a new relationship was derived :

$$X = \frac{Y}{A} + C + \sqrt{\frac{- C.Y}{2.A}} \qquad (2)$$

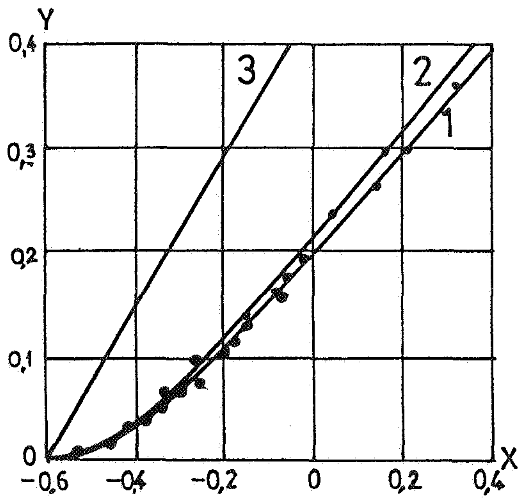

Fig. 7 : *Cowls : agreement between experimental data and different formulas*
1 best fit
2 fit using equation (2)
3 fit using linear equation (3)

This relation leads to a better agreement and is quite easy to handle because the coefficients, A and C may be readily measured, according standardized methods :

A is related to the cowl pressure drop when there is a no wind effect,

C is related to the suction effect due to wind velocity when there is no flow rate.

More work is still needed to correlate flow rates test results on commercially avaible cowls with theoretical analyses.

A last difficulty arises from wind turbulence. Computer simulations [6] [7] showed that taking into account wind turbulence makes it possible to explain reverse flow at the upper floor.

4 - CONCLUSION

Design methods based on computer codes makes it possible to improve the design of ventilation systems. More research work is still needed in order to ascertain results relevant to natural ventilation.

ACKNOWLEDGEMENTS

The author wish to express his thanks to R.MOUNAJED who developed computer models, J.GANDEMER who provided wind data and J. RIBERON for general assistance. Participation of AFME (Agence Française pour la Maîtrise de l'Energie), to research support is gratefully acknowledged.

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11 th AIVC conference. Ventilation system performance
Belgirate, 18 - 21 Septembre 1990

Discussion
Paper 27

Bas Knoll (TNO, Netherlands)

Did you do any research on flow rates and flow directions in leeward-side rooms? The high cross-ventilation you showed may cause under ventilation in these rooms.

D.Bienfait (CSTB, France)

We have not done this research yet, but I know it is important to reduce the airleakage to prevent these problems. Also in Canada tests are done on this item.

E.Arens (U.C. Berkeley, USA)

How are you proposing to solve the problem of residents closing the supply vents to control draughts? Is it a detail in the design of the supply vent itself?

C-A Roulet, (LESO, Switzerland)

In many countries it is shown that air inlets are often taped or closed by the occupants. Is it not time to take this fact into account and act in order to avoid this behaviour? (e.g. hide the inlets, avoid the draughts, inform inhabitants).

D.Bienfait (CSTB, France)

Use of closeable supply vents is not considered to be a good solution because the ventilation system would not work properly when closed. On the other hand, uncloseable supply vents should provide good comfort conditions, otherwise they are likely to be taped, which would be even worse. The design of supply vent should therefore be improved. This can be achieved by better qualification of the component with respect to draught problem. This is presently discussed inside CEN TC 156 Committee "Ventilation of residential buildings". On the other hand some unconventional components such as - pressure difference, or moisture, or outdoor temperature controlled devices; - dynamic walls or windows; - supply vents with heat exchanges or additional heater should be encouraged because they may help to solve the draught problem.

B.Fleury (ENTPE/LASH, France)

Industrial ventilation products are of very high quality but they are so poorly installed that the final building performance is terrible. What do you suggest to improve building quality and especially the leakage characteristics?

D.Bienfait (CSTB, France)

I would not give so definitive an appreciation as you on the quality of ventilation installations. However, some improvements may be necessary. Two complementary suggestions can be made: - better information for installers through guidelines or calculation methods; such documents are currently being issued in France and concern for instance building leakage or installation rules; - develop methods to check the performance of the installation, measurement of air leakage, airflows,... in France measurement of airflows at exhaust vents for checking purpose is an acknowledged practice. On the other hand it may be noticed that not all the ventilation systems have the same sensitivity to poor installation.

VENTILATIOAN SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 28

HUMIDITY CONTROLLED NATURAL VENTILATION WITHOUT AUXILIARY ENERGY SUPPLY

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1. Introduction

As a consequence of measures required for reducing the heating energy consumption in residential buildings, there have been more and more complaints in the last few years on the appearance of mould in dwellings. In most cases, it is retrofitted or renovated old buildings which are affected [1]. Mould growth is frequently the result of a severe reduction in the natural air change rate in old buildings following the installation of airtight windows, while user habits remain the same as before. Each day, an average amount of 8 to 15 liters of moisture is generated in dwellings, which is usually conveyed to the outside through window joints. However, airtight windows and insufficient ventilation cause indoor air humidity to rise. This may lead to surface humidity on cold external walls, e.g. at thermal bridges, thus providing ideal conditions for mould growth. The effect is enhanced unless the insulation level of the external wall is greatly improved so that the surface temperature of the exposed areas is increased. According to [2], mould growth is influenced by the following parameters: nutrient availability, temperature, ph-value of the substrate and, in a decisive manner, the amount of water in the substrate. According to [3], in one third of all cases, the damage is obviously caused by user-related, high indoor air humidity (see Table 1). This is the result of tests performed in 300 old buildings, where several examinations were

Tab.1: Compilation of the parameters most frequently analysed in the test. Often, the cause of damage is not clearly attributable to a single parameter. Therefore, the sum of the parameters is a value above 100 %.

Influential factors for moisture damage			Frequency [%]
Building	Thermal bridges	Parapet	21
		Window reveal	18
		Others	5
	Rain protection device	Small cracks	37
		Large cracks	15
	Rising moisture		9
Occupant	Indoor air humidity (mould growth on furniture)		31

carried out in different dwellings. Besides, structural deficiencies such as thermal bridges or insufficient humidity protection of the building envelope were identified as being responsible for the remaining cases of damage. It can therefore be concluded that, supposing the construction is sound, it is the user-related indoor moisture load that should be reduced to prevent humidity damage and resulting mould growth.

2. Ventilation

While items such as missing or unsufficient thermal insulation and bad humidity protection as well as insufficiently insulated thermal bridges are inherent in the construction itself, humidity production and release can be influenced by the user. In the absence of building defects (sufficient thermal insulation and humidity protection), it is indoor air humidity that becomes the major cause for mould growth. Free ventilation through windows and untight window joints is the kind of ventilation most frequently performed in dwellings. An increased air change through window ventilation is something that is done subjectively by the user. As continuous ventilation results in increased heat losses during the heating period (often with mould contamination of the window reveal caused by convective cooling), it seems that short periods of intensive ventilation would be most efficient for energetic and hygric reasons. However, as the assessment of the indoor air quality is a subjective thing, ventilation controlled by the user may result in air change rates which are either excessively high (heat losses!) or dangerously low (risk of mould!).

3. Free ventilation according to actual parameters

A humidity controlled ventilation unit based on the principle of free ventilation (see Fig. 1) has been developed and patented at the Fraunhofer Institute of Building Physics. The device is a ventilation valve which may be installed in external walls or window frames, thus establishing a connection between indoor and outdoor air. It is automatically opened and closed by way of a sensor, without mechanic or electric auxiliary energy being required. Based on the principle of the hair hygrometer, the opening works according to humidity dependent length variations of suitable natural or chemical fibres employed to provide a mechanic force. For this reason, the unit does not require auxiliary energy and will operate automatically for years. The unit will stop the air change between indoor and outdoor air, if the indoor air humidity decreases to values which do not lead to critical surface humidity at the external wall. Simultaneously, increased heat losses are avoided.

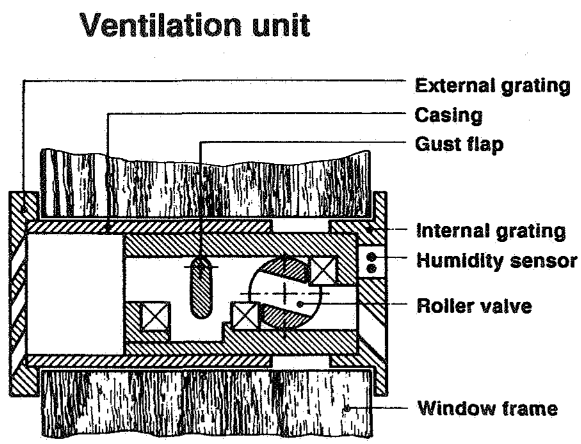

Fig.1: Schematic representation of the humidity controlled ventilation unit.

4. Function

The cross section of the unit is controlled by the sensors according to actual parameters, i.d. in case of high indoor air humidity, the valve of the unit will open to enable an air change. The opening characteristic is illustrated in Fig. 2.

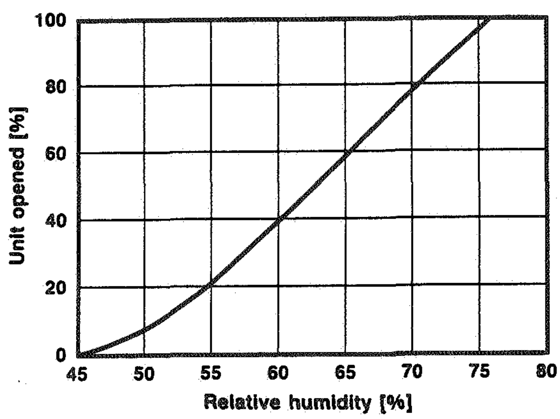

Fig.2: Opening characteristic of the ventilation unit developed by the Fraunhofer Institute of Building Physics.

The air change rate is dependent on the pressure difference between inside and ambient environment. Figure 3 presents the air flow rate through the valve in closed and open state dependent on the existing pressure difference for the prototype of a wall unit. Pressure differences of 10 to 20 Pa already lead to air flow rates of app. 15 to 20 m^3/h through the ventilation unit.

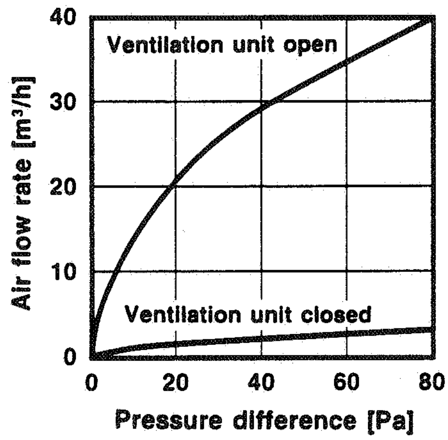

Fig.3: Air flow rate through the ventilation unit while open and while closed. The measurements were carried out according to German standard DIN 18055.

In order to examine the process of free ventilation by means of this device, four units were mounted in the unoccupied test house [4] as it can be seen in Fig. 4.

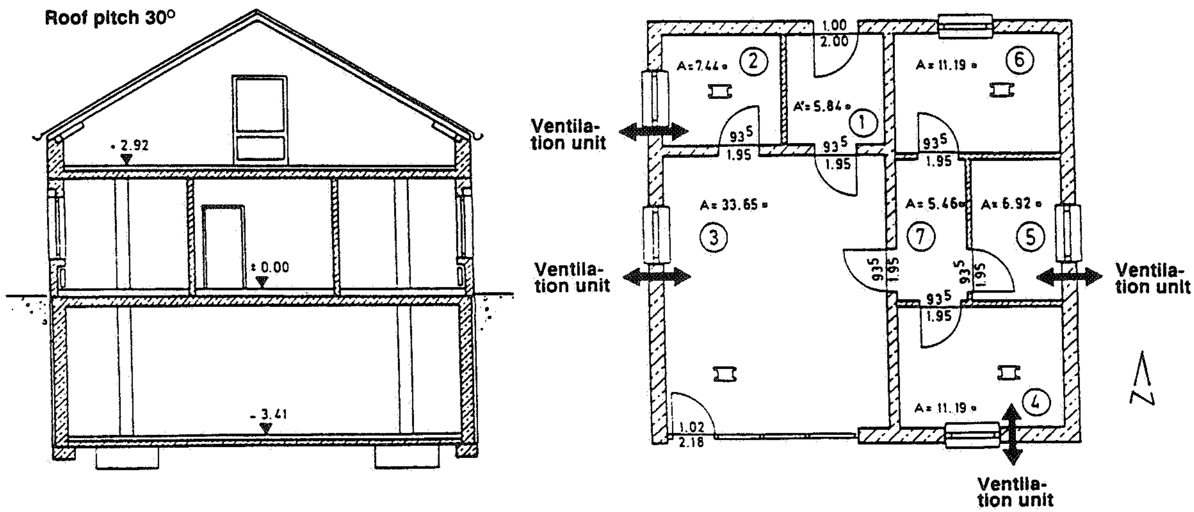

Fig.4: Floor plan and section of the test building and location of the ventilation units according to [5].

In this connection, different installation situations were subject to investigation. Using tracer gas measurements, it was possible to determine the air change rate in single rooms under realistic wind pressure conditions with the units fully open or closed. It was observed that closed units hardly increased the airtightness of the entire building in comparison to the basal air change rate without units [5]. The measurement results are shown in Fig. 5. The values recorded for each room are plotted dependent on wind speed for open and closed units respectively. The hatched section between the curves represents the increased ventilation rates which result. The values for room 2 in Fig. 5 examplarily demonstrate that the unit increases the air change rate from 0.3 to 1.0 h^{-1} at a wind speed ranging from 2 to 7 m/s.

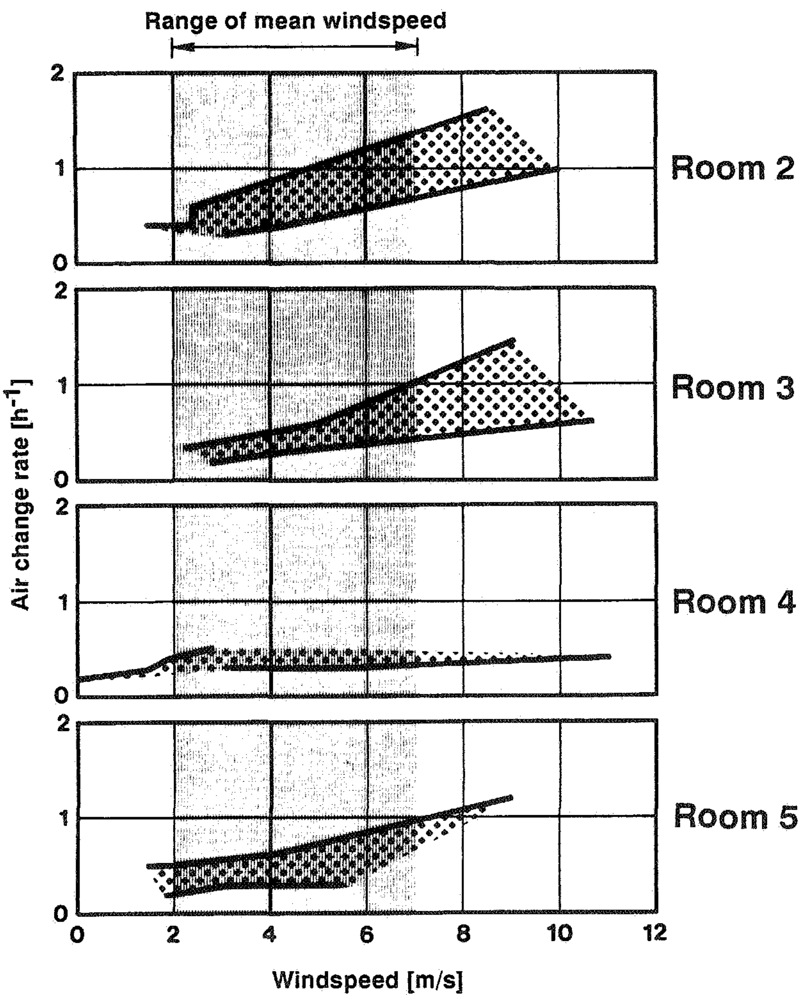

Fig.5: Comparison between the air change rates recorded within the test building for the rooms 2 to 5 with the unit being closed (lower curve) and open (upper curve). The hatched section demonstrates the increased air change rates obtained by way of the ventilation unit according to [5].

Figure 6 displays the mean wind speeds for the heating periods according to the 12 test reference years developed by [6] at different reference locations in Germany.

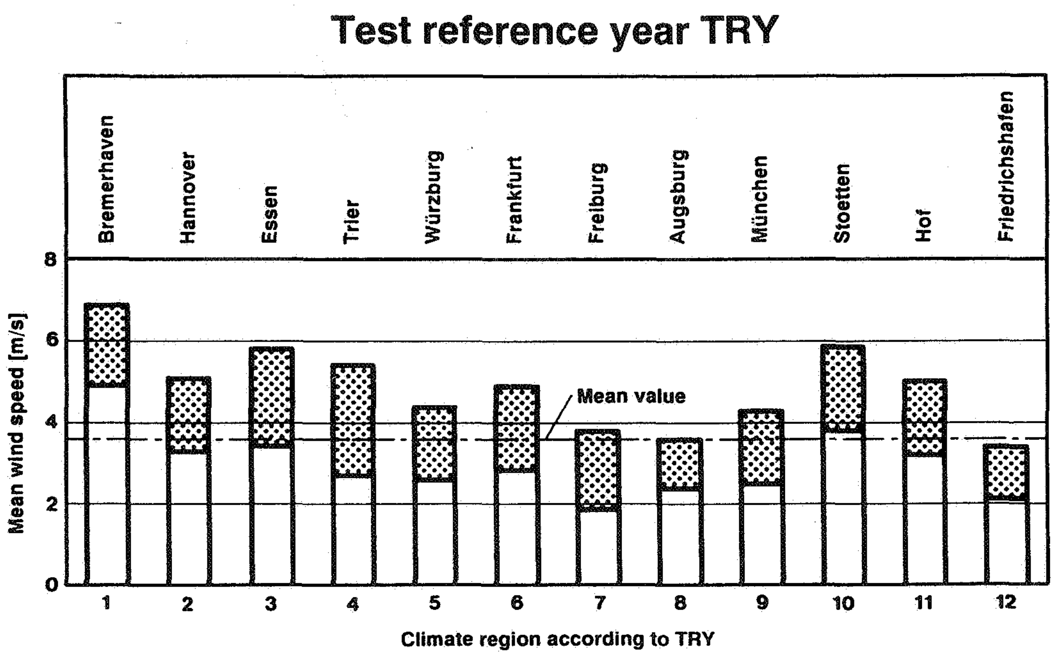

Fig.6: Mean wind speeds during the heating period as they were recorded by the 12 stations in the test reference years developed by [6] for the Federal Republic of Germany.

The mean wind speed of all locations is 3.6 m/s; the lowest mean value being 2.5 m/s, the highest 5.4 m/s. At the location with the lowest wind speed, the air change rate is 0.6 h^{-1} for room 2 with the unit open.

It can be concluded from the measurements that given mean wind speeds, free ventilation by way of the units described results in air change rates of > 0.5 h^{-1} which are recommended in [6] to prevent mould growth caused by too humid indoor air.

Besides, the measurement demonstrates that it is useful to install several units in order to provide supply and exhaust air valves depending on wind pressure.

5. Conclusion

To prevent mould in dwellings with a sound building fabric, it is necessary to convey humid air to the outside. To this end, the paper at hand presents an efficient device for free ventilation developed at the Fraunhofer Institute of Building Physics. The unit functions automatically (without auxiliary energy). Depending on the relative air humidity, it can adjust air change rates such that the r.h. will not remain critical for a longer period of time. It is advisable to test this unit in practice in order to prove the intended effect of preventing mould due to excessive indoor air humidity.

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Discussion
Paper 28

C-A Roulet (LESO, Switzerland)

In many countries it is shown that air inlets are often taped or closed by the occupants. Is it not time to take this fact into account and act in order to avoid this behaviour? (E.g. hide the inlets, avoid the draughts, inform inhabitants, etc).

M Szerman (Fraunhofer Institut fur Bauphysik, Germany)

I agree completely that the occupant should not be able to adjust or close demand controlled internal ventilation units. In fact our developed unit cannot be closed by the user.

W. Raatschen (Dornier GmbH, Germany)

a) Your system works about in the same way as the French AERECO system. Do you expect the same performance? b) Why was there a need for you to develop a new system if such systems do already exist?

M Szerman (Fraunhofer Institut fur Bauphysik, Germany)

a) Respectively the air change rate due to this element, we expect nearly the same performance as the AERECO. But our element avoids problems with thermal and acoustic bridge effect due to the element, by the special construction. AERECO elements don't consider this, so that it is expected that this element will get problems, even in the German market. b) When we develop our system, AERECO was not on the market. Besides that, our element takes into account that much different outside temperature/humidity conditions different air change rates are required, to reduce the indoor air humidity by the same amount.

H Hens (Leuven, Belgium)

Comment: 1) Tricky paper, proving that the larger an air inlet opening, the higher the ventilation rate. 2) Humidity control = correct solution. To avoid mould one should monitor or the RH against the coldest surface or ********* (outside temp, inside temp, RH)

VENTILATIOAN SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 29

PERFORMANCE ASSESSMENT OF A HUMIDITY CONTROLLED VENTILATION SYSTEM

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SYNOPSIS

Demand controlled ventilation systems have recently become an interesting opportunity to achieve acceptable indoor air quality while minimizing energy consumption. Although they are usually designed for buildings showing relevant variations of occupancy (e g, office buildings, schools, etc.), there are now examples of applications also in residential buildings. One example is the passive humidity-controlled ventilation system recently developed in France. This type of installation has been tested in a five- storey apartment building located in Torino, Italy, during the winter 1989. Preliminary results, concerning air temperature and relative humidity data and system operation, are presented in this paper. Analysis of data shows that the system is capable of maintaining air humidity levels below the limit values in most situations, reacting effectively to changes in the occupancy patterns and activities. The energy savings compared to a conventional constant flow ventilation system have also been calculated.

LIST OF SYMBOLS

c_p = specific heat of air at constant pressure (J/kgK)
m = air mass flow rate (kg/s)
m_v = moisture production rate (kg/s) or (g/h)
n = number of airchanges per hour (h^{-1})
p_s = saturation pressure of water at temperature T (Pa)
p_t = total atmospheric pressure (Pa)
Q_v = ventilation heat load per unit volume (W/m^3)
RH = air relative humidity (%)
T = air temperature (°C)
T_d = dew point temperature of indoor air (°C)
T_s = surface temperature of window frame (°C)
x = air humidity ratio (kg/kg)
v = specific volume of air (m^3/kg)
V = room volume (m^3)

Subscripts

i = indoor
o = outdoor
1 = ambient no. 1
2 = ambient no. 2

1. INTRODUCTION

Mechanical ventilation is seldom adopted in residential buildings in Italy. However, the synergic effect of recently developed factors (e g, supertight windows, lower indoor temperatures, and cold bridges frequently caused by incorrectly placed thermal insulation) are now often creating condensation problems, particularly in Northern Italy, where cold and rather humid winters are common. As a consequence, mechanical ventilation is now been considered as a useful technique to avoid condensation. Among ventilation techniques, novel technologies such as Humidity Controlled Ventilation (HCV) systems (i.e. demand controlled ventilation, based on humidity control) appear particularly interesting.

A multifamily building, equipped with a passive humidity controlled mechanical ventilation system, has been instrumented in order to assess the performance of this type of installation under field conditions in the climate of Torino (northwestern Italy). The building -- which is five storeys high with two flats at each level -- is a good example of current practice in the residential sector, in terms of size, construction technology, and type of heating system.

The specific aims of the investigation were:

- to check the resulting air humidity levels in terms of preservation of the building constructive elements and thermal comfort of the occupants;
- to determine if the air change rates resulting from the adoption of this ventilation strategy are sufficient to provide an acceptable indoor air quality (IAQ);
- to compare the adopted ventilation strategy with natural ventilation and traditional (i.e., without feedback) mechanical ventilation systems on the grounds of energy savings and IAQ;
- to verify the subjective reactions of the occupants to the adoption of an unconventional ventilation system.

The measurement campaign started on October 20, 1989 and ended two months later; although the extension of this campaign was rather limited, a significant range of winter climatic conditions was covered, thanks to the unusually cold weather that occurred in early December. Temperature and humidity profiles were recorded continuously outdoors and in nine representative rooms of three of the ten flats. Questionnaires were also employed to collect information about the occupants' behaviour.

2. THE SITE AND OBJECT OF INVESTIGATION

The climate of Torino can be concisely defined through the following data:

- length of the heating season: 180 days, typically from October 15 to April 15;
- number of degree-days: 2700, base 20°C;
- wind speed: usually very low during the coldest months ($\leq$1.0 m/s), scarcely exceeding 1.0 m/s during the central hours (from 12:00 to 16:00) of the day in the other months of the heating season;
- relative humidity: generally high (above 70%) from November to February.

A plot of temperature vs. humidity ratio for the typical months in the heating season is shown in Fig. 1.

The investigated building is part of a group of three buildings which can be considered identical under any point of view. Each building accomodates eleven flats (two at each floor plus a small "conciergerie" at the ground floor), has an overall volume of 3500 m^3, and a heated area of about 1400 m^2. Each flat is equipped with an individual hydronic heating system, including a gas boiler and hot water radiators.

The ventilation system is centralized, with one extraction fan in each building having a nominal power of 0.55 kW and a nominal flow rate of 3,000-4,000 m^3/h with a pressure head of 150-200 Pa. Air is evacuated from each flat through three extraction grilles, located in the two bathrooms and in the kitchen. Exhaust air from each flat is driven through two vertical ducts (I.D. = 125 mm) into the attic, and then collected by a horizontal duct (I.D. = 250 mm) to the fan. Silencing devices are located at each extraction grille and at the top of each column. Fresh outdoor air is introduced into the flats through the hygro-controlled immission grilles located in the roller blind boxes of the living room and the bedrooms. Ambient air is exctracted through hygro-controlled exctraction grilles installed in bathrooms and kitchen.

3. THE PASSIVE HUMIDITY CONTROLLED VENTILATION (HCV) SYSTEM

The passive HCV system, as well as the entire ventilation system, is manufactured by the French company ALDES. The system is based on a very simple principle: the relative humidity level indoors is controlled by means of a sensor, which is itself an actuator, i.e. a device regulating the inlet area of the air immission grilles. This concept is interesting because one device only replaces all the electromechanic chain from the transducer to the controller and finally to the actuator, with a lower investment cost and lower risks of failure.

The inlet and outlet grilles are made of PVC, as most other components of the system. The size of the grille opening varies with relative humidity, due to a humidity sensitive tissue which varies its length with relative humidity (RH). The tissue is made of a polyammidic fibre, treated and stabilized by the producer. The grille opening characteristic shows a linear variation between RH = 40 % (A = 5 cm^2) and RH = 75 % (A = 30 cm^2). In the exctraction grille the airflow rate is controlled by a rubber membrane which modifies the cross section of the air passage according to ambient air humidity.

3.1. Theoretical evaluation of HCV systems

HCV systems show an interesting capability of automatically controlling the ventilation heat load. This may be explained as follows: as the outdoor air temperature diminishes, air humidity ratio usually tends to decrease, even with increasing relative humidities (see Fig. 2). Therefore, provided the indoor conditions (air temperature, and moisture production) do not change, the amount of outdoor air required to maintain the RH setpoint will decrease with diminishing air temperature (see Fig. 3). As a consequence, ventilation losses will not increase linearly with decreasing temperature, but will keep stable within a large range of temperatures and will even decrease with decreasing outdoor temperatures (see Fig. 4).

There is an obvious theoretical limit to the possibility of controlling indoor humidity in this way: when moisture content outdoors is greater or equal than the required humidity ratio indoors no rate of air change will be sufficient. Usually, as can be seen from Figures 1 and 2, this is not the case for the heating season in Torino. However, there may be a risk of this kind during the remaining months of the year. For example, if the indoor conditions are $T_i = 20°C$ and $RH_i = 50\%$ (i.e., $x_i = 0.0073$ kg/kg), the limit values of outdoor RH above which HCV will not work are given in Table I.

Table I - Limit outdoor RH values for HCV Systems

T_o (°C)	≤9.1	10.0	12.0	14.0	16.0	18.0	20.0
RH_o (%)	100	93	82	72	65	55	50

The analytical explanation of the phenomenon for a "perfect" HCV system is given in the following. The mass flow rate of outdoor air required to maintain a constant RH indoors is given by:

$$m = \frac{m_v}{x_i - x_o} \qquad (1)$$

where

m = outdoor air mass flow rate (kg/s)
m_v = moisture production rate (kg/s)
x_i = indoor air humidity ratio (kg/kg)
x_o = outdoor air humidity ratio (kg/kg)

Eq. 1 can be rewritten in terms of airchanges:

$$n = \frac{3600 m_v}{(V/v)(x_i - x_o)} \qquad (1')$$

where

n = number of airchanges per hour (h^{-1})
V = room volume (m^3)
v = air specific volume (m^3/kg)

In the expressions above, it may be assumed that m_v and x_i are both constant. Under such circumstance, m (or n) will be a function of x_o only. This, in its turn, will be a function of outdoor temperature T_o and relative humidity RH_o through the well known relationship:

$$x_o = 0.622 \frac{RH_o p_s(T_o)}{p_t - RH_o p_s(T_o)} \tag{2}$$

where

$p_s(T)$ = water saturation pressure at temperature T

p_t = total atmospheric pressure

Since the ventilation heat load per unit volume (W/m^3) is given by:

$$Q_v = mc_p(T_i - T_e)/V \tag{3}$$

one finally obtains:

$$Q_v = \frac{mc_p(T_i - T_o)/V}{x_i - 0.622 \dfrac{RH_o p_s(T_o)}{p_t - RH_o\, p_s(T_o)}} \tag{4}$$

Equations (1') through (4) were solved assuming

T_i = 20°C

RH_i = 50 % (from which x_i = 0.0073 kg/kg)

m_v/V = 10 g/(hm^3).

Results of the analysis are shown in Figures 2, 3 and 4.

4. RESULTS OF THE MEASUREMENT CAMPAIGN

4.1 Indoor Air Quality considerations

The first aim of the investigation was to verify the system capability of maintaining relative humidity below a certain level. Figures 5 and 6 show respectively the frequency distribution and the cumulated frequency distribution plots of indoor RH (relative to about two months of hourly data) in the different rooms of Flat # 1 (kitchen, bathroom, living room, and bedroom). From these pictures it can be seen that, although the highest vapour production occurs in the kitchen and the bathroom, in these two rooms only 10 % of RH values are above 50%, and around 2-3% values are above 55%. The lowest RH values have been detected in the living room, and the highest ones in the bedrooms.

By means of questionnaires it was possible to define the typical daily and weekly activity schedules of the tenants in the instrumented flats, and from these the water vapour production was estimated. The questionnaires included questions about the location of the vapour producing electrical equipment, the cooking habits, the use of sanitary hot water, the presence of plants in the rooms, the daily schedules of the tenants, etc.

This also allowed to interpret the time plots of RH and estimated water vapour production. As an example, Fig. 7 shows the situation in a kitchen for a typical day: the plot indicates that the system was fully able to offset the increase in water vapour production even during cooking times.

4.2 Surface condensation problems

A second type of analysis refers to surface condensation problems and, in particular, to the condensation events on the aluminum frame of the windows, which is usually the coldest internal surface of the envelope. (Such events were indeed the most frequent problem that was pointed out by the occupants in the questionnaires.)

The analysis consists of three steps:

- determination of the indoor frame surface temperature (T_s) as a function of outdoor and indoor temperature, using a numerical heat transfer code;
- determination of dew point temperature (T_d) as a function of indoor air temperature and relative humidity;
- construction of frequency distribution plots for (T_s - T_d).

Results for the kitchen of Flat # 1 are given in Figures 8 and 9; the two bar graphs respectively show the absolute frequency of condensation events as a function of outdoor temperature and time of the day. From these data, it can be argued that it is no longer possible to control indoor humidity through ventilation in order to avoid surface condensation when two concurrent factors are present, i.e. a high vapour production rate (preparation of meals: see Fig. 9) and high outdoor humidity (which is typical of mild weather, with air temperature well above the winter minima: see Fig. 8).

4.3 Flow rate and energy savings evaluations

The evaluation of energy savings requires the determination of i) the actual number of air changes, and ii) the theoretical number of air changes required by a "perfect" HCV system to keep the indoor RH constant.

As an example, the problem was solved for Flat # 1 (which is the smallest of the three flats that were instrumented). The flat was divided into two zones: a night-zone including bedroom and bathroom, and a day-zone including living room and kitchen. Assuming that air flows between zones are zero, the problem can be solved for each zone independently. Knowing the measured values of air humidity ratio in the two rooms and outdoors, and the estimated values of water vapour productions in the two rooms, the air flows from outdoors to each room and between the rooms can be determined. For instance, for the night-zone (bedroom 1 + bathroom 2), reminding that the extraction grille is located in room 2, the following system of mass balance equations can be written:

$$x_o m_{o1} - x_1 m_{12} = - m_{v1} + (V_1/v)(dx_1/dt)$$
$$x_o m_{o2} + x_1 m_{12} - x_2 m_{2o} = - m_{v2} + (V_2/v)(dx_2/dt) \qquad (5)$$

where

$m_{o1} \quad = m_{12}$ (air flow in and out of room 1)

and

$m_{2o} \quad = m_{12} + m_{o2}$ (air flow extracted from room 2)

are the two unknowns.

The system of equations (5) was solved as an example for a ten hours period, and the results were compared with the theoretical air flow rate needed by a "perfect" HCV system to maintain 50% RH in the two rooms. Results in terms of total extracted air flow (m_{2o}) for the actual case and the perfect system case are presented in Fig. 10. Although the trend is qualitatively the same there is a large difference between the two values, which can be explained by the fact that the actual RH in the two rooms was well below 50%. Figures 11 and 12 show the disaggregation of flow rates (i.e., m_{2o}, m_{12}, and m_{o2}) for the actual case and the perfect case respectively. The average flow rates during the considered period for the two cases are reported in Table II.

Table II - Average flow rates and ach for actual/perfect cases.

Flows: Total				From outd. to room 1				From outd. to room 2			
actual		perfect		actual		perfect		actual		perfect	
m_{2o} m^3/h	n_{tot} h^{-1}	m_{2o} m^3/h	n_{tot} h^{-1}	m_{2o} m^3/h	n_1 h^{-1}	m_{2o} m^3/h	n_1 h^{-1}	m_{2o} m^3/h	n_2 h^{-1}	m_{2o} m^3/h	n_2 h^{-1}
29.8	0.50	13.3	0.22	16.3	0.35	10.8	0.23	13.5	1.00	2.5	0.19

The results show that the theoretical airchange rates necessary to maintain 50% RH are normally well below 0.5 ach, i.e., the recommended airchange rate for IAQ control in dwellings; this seems to indicate that by adopting a RH set-point slightly lower than 50%, values of ach would be achieved which are more suitable to offset the other normal pollutants such as carbon dioxide and body odours.

In other examples, which are not reported here for the sake of brevity, apparently meaningless results were found, such as negative inter-room flow rates or flow rates which are greater than the design flow rate of the extractors. The first fact may be interpreted as an inversion of flow (from room 2 to 1 instead than 1 to 2, as supposed in the equations 5). In the second case the overestimate of extracted flows may be due to the hypothesis of perfect mixing adopted in the equations (5). In effect, due to the location of the extraction grilles (e.g., right above the cooking equipment in the kitchens), there is an obvious "hood effect" which increases the ventilation efficiency requiring less air than would be necessary under perfect mixing conditions.

An evaluation of energy savings based on this limited amount of data only is probably not meaningful. However, it can be observed that, while the actual average airchange was 0.50 ach, a constant ventilation system maintaining the same maximum indoor RH would have required 0.83 ach, i.e., 66% more than the installed HCV system.

4.4 Occupants' acceptance

In order to assess the performance of this type of installation it was decided to gather information about the "subjective" acceptance of the system by the occupants. For that aim a questionnaire (see Table III) was developed and distributed to the occupants of both buildings equipped with the HCV system.

Table III - Questionnaire filled by the occupants.

1. Did you notice humidity problems in the building components?
2. Are you satisfied with indoor temperature levels?
3. Are you satisfied with indoor humidity levels?
4. Did you notice any malfunctioning of the ventilation system, such as:
   a - noise
   b - air draughts
   c - insufficient ventilation
   d - eccessive ventilation
   - other
5. Did you attempt to modify the operation of the ventilation system?
6. Did you modify your habits regarding window opening for airing?
7. Are there any modifications you would like to suggest about the installation or the use of the ventilation system ?

The results of the questionnaire are listed in Table IV. A total of 20 questionnaires were distributed to the tenants. Twelve families did not reply. Six questionnaires were returned, one of which incomplete. Two tenants refused to fill the questionnaire and declared to be globally unsatisfied with the system, without explaining their reasons (which probably indicates an "a priori" bias against any technological innovation!). In general, the tenants that answered the questionaire (probably, those who paid more attention to the operation of the system) expressed a global satisfaction, while pointing out some relatively minor problem.

The most frequent problem that was detected is the condensation of water vapour on the aluminum window frames in the bathroom; obviously, this is an intrinsic limitation of the system, which cannot detect the presence of cold spots due to thermal bridges: therefore, condensation cannot be avoided if, as in the case which has been investigated, window frames are made of a highly conductive material and exhibit thermal continuity between indoors and outdoors. A few of the tenants have expressed some annoyance, especially in the coldest days, for the cold draughts creeping into the bedrooms through the grilles during the night, and have also tried (successfully) to outdo the system by taping the inlet grilles. Other problems, such as temperature differences between rooms, cannot be attributed to the ventilation system.

On the positive side, several tenants noticed that the HCV system allowed them to reduce airing and that indoor humidity was acceptable even under "severe" (e.g., cooking time) conditions. The quality of indoor air was also evaluated satisfactory.

Table IV - Results of the questionnaire

Quest. #	YES # repl.	YES %	NO # repl.	NO %	NOTES
1	4	67	2	33	Condensation on bathroom window frames
2	4	67	2	33	Temperature differences between rooms
3	6	100	0	0	
4a	1	17	5	83	
4b	0	0	6	100	
4c	1	17	5	83	
4d	0	0	6	100	
5	1	20	4	80	Air inlet plugged due to low temperature
6	4	80	1	20	Reduced need of airing
7	0	0	5	100	

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Raatschen, W. (Ed.), *Demand Controlled Ventilating System - State of the Art Review.* Annex 18, IEA. Swedish Council for Building Research D9:1990. Stockholm, 1990.

Fig. 1 — Humidity ratio vs. temperature

October–March typical values in Torino

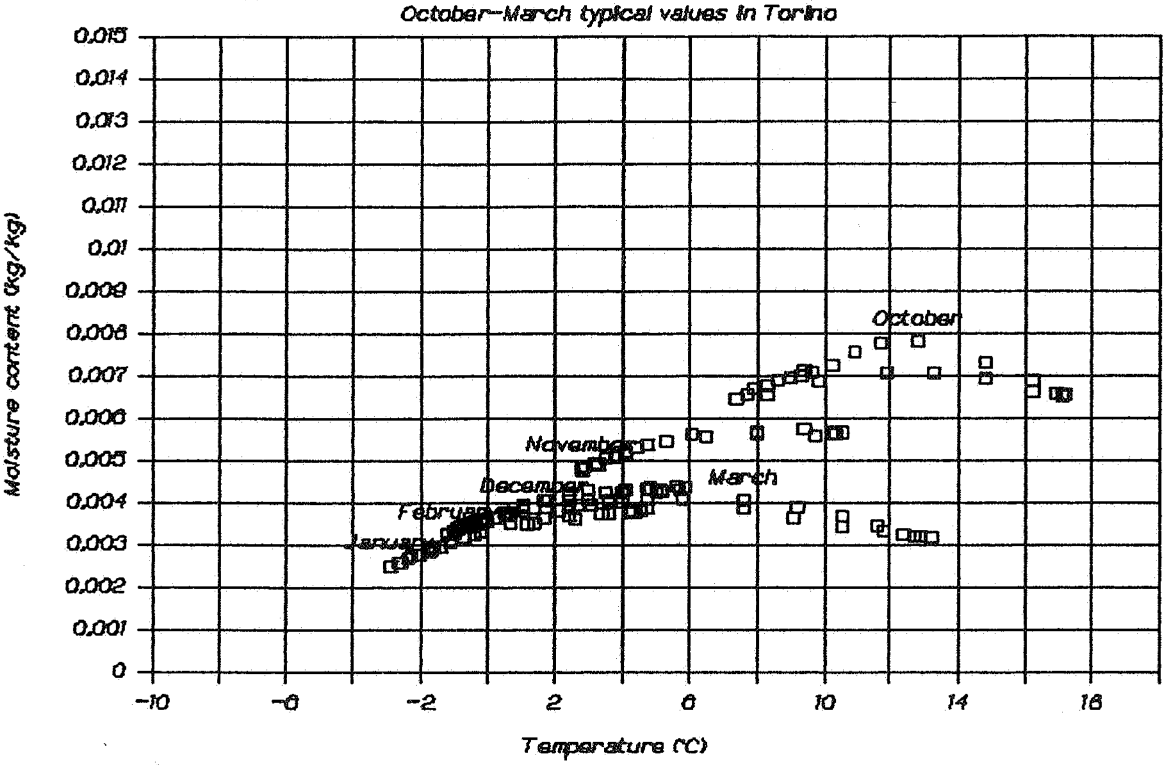


Fig. 2 — Humidity ratio vs. temperature

Parameter: relative humidity

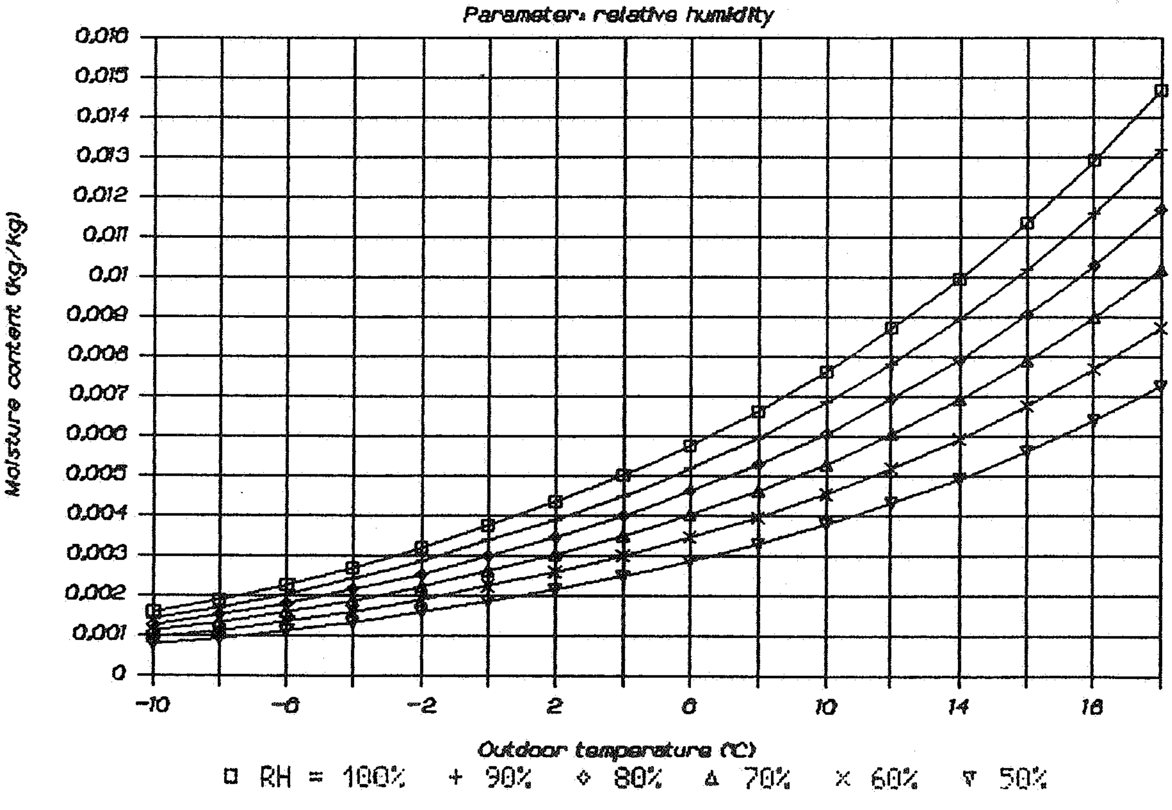
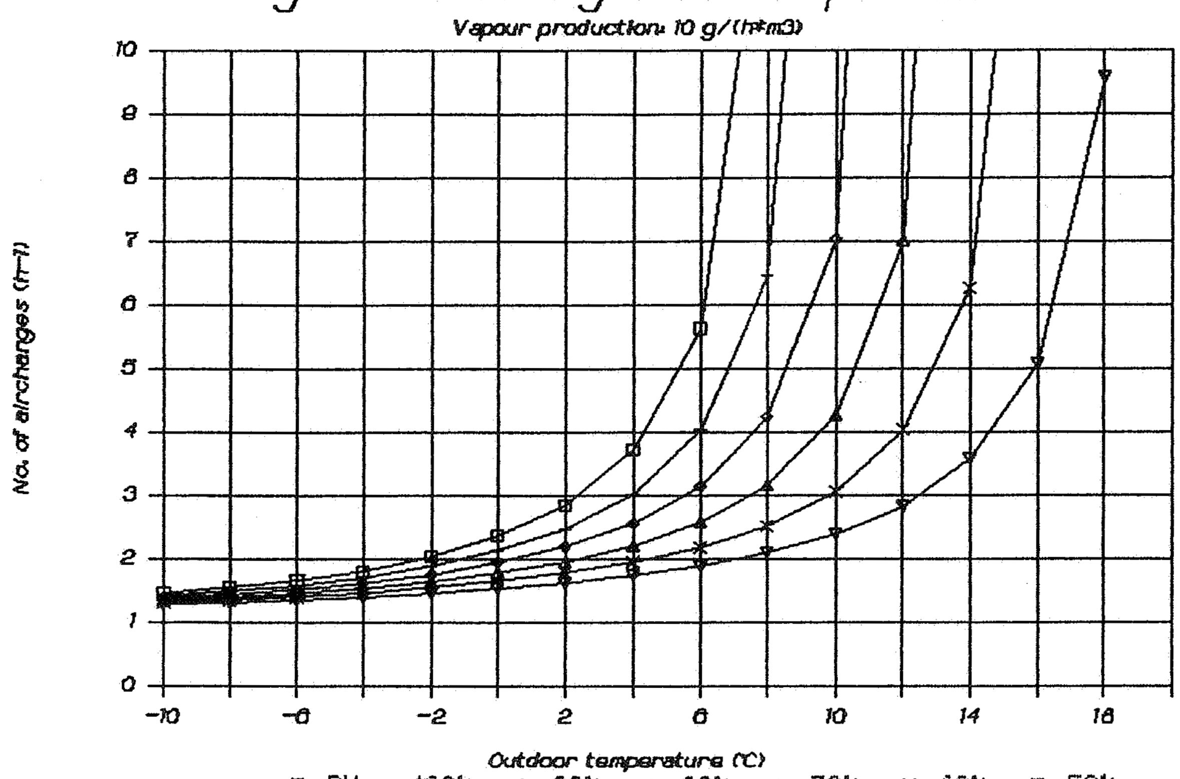
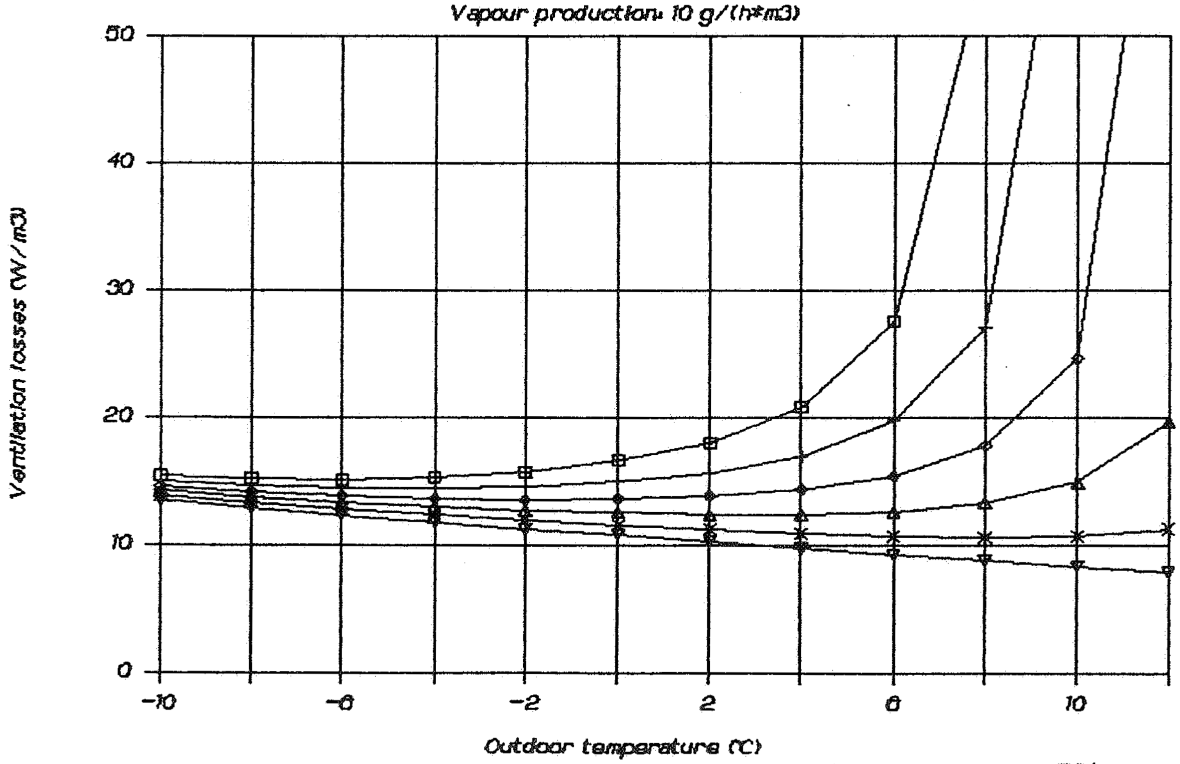


Fig. 3 – Airchanges vs. temperature



□ RH = 100% + 90% ◇ 80% △ 70% × 60% ▽ 50%

Fig. 4 – Ventilation losses vs. temp.



□ RH = 100% + 90% ◇ 80% △ 70% × 60% ▽ 50%

Fig. 5 – Frequency distribution of RH

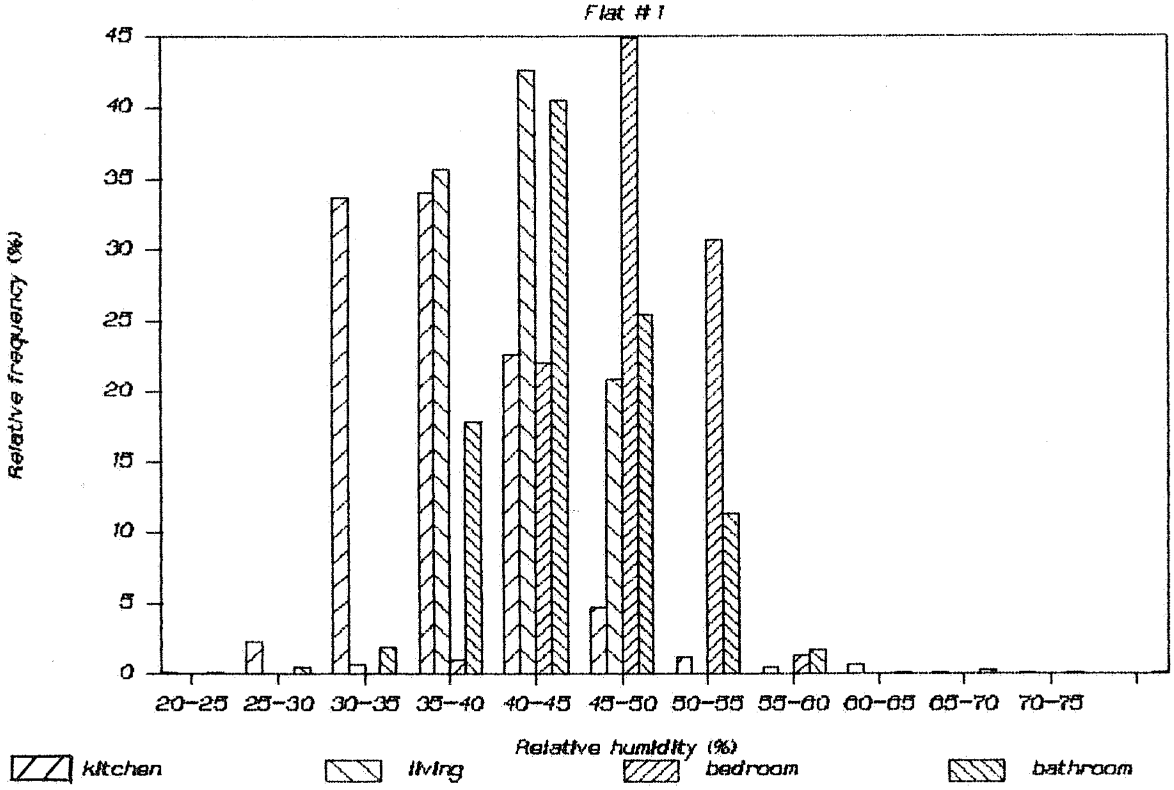


Fig. 6 – Cumulated freq . distr. of RH

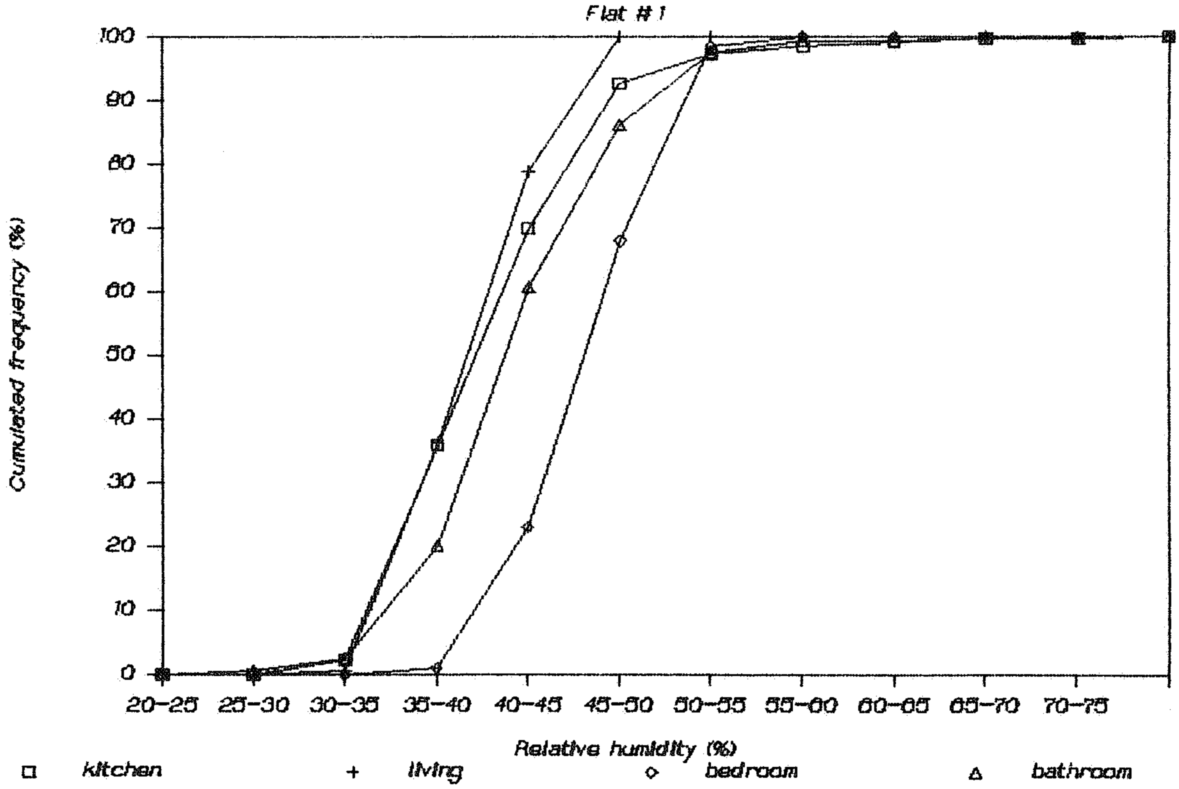


Fig. 7 - RH and Vapour production

vs. time of day

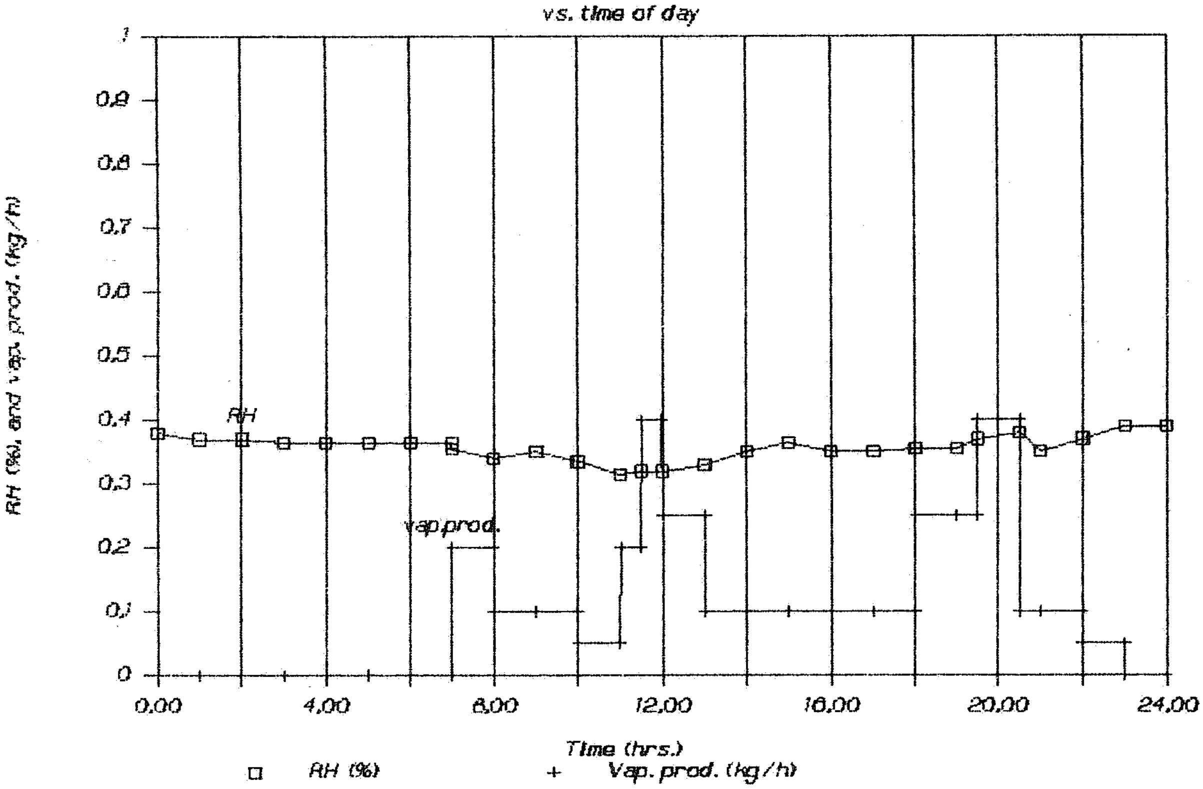


Fig.8 - Condensation events

Frequency vs. outdoor temperature

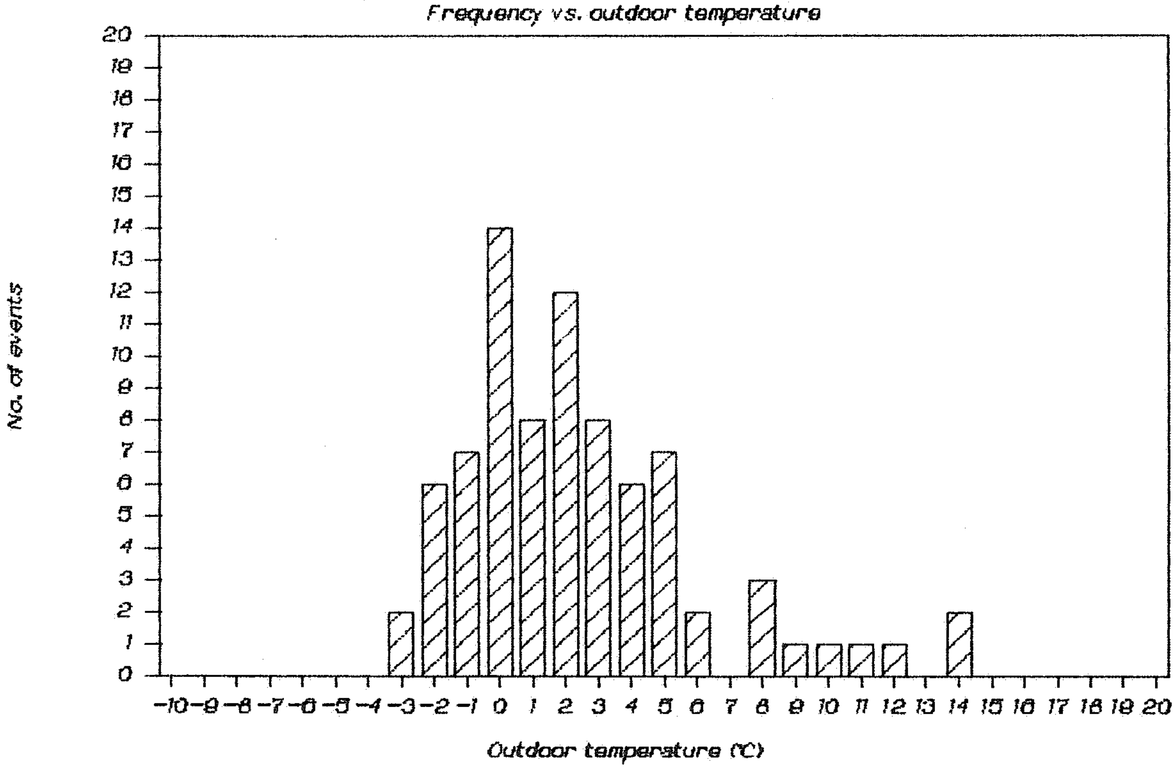


Fig.9 – Condensation events

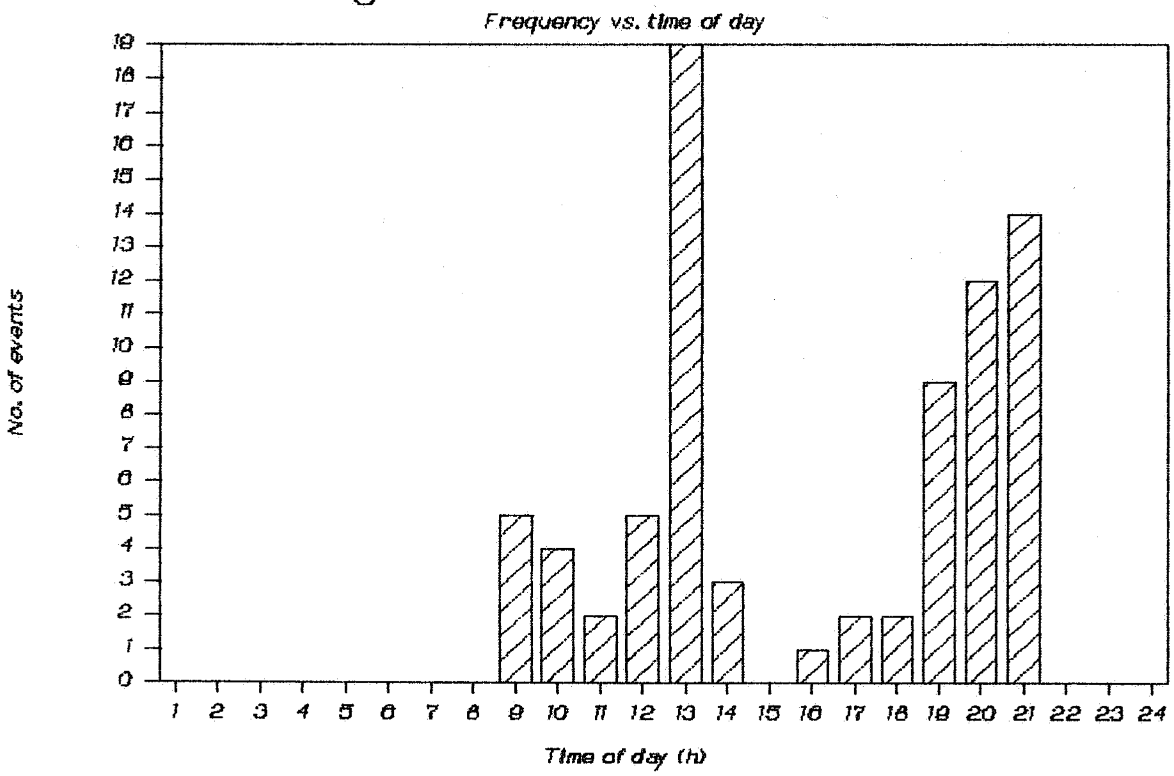


Fig. 10 – Calculated air flows

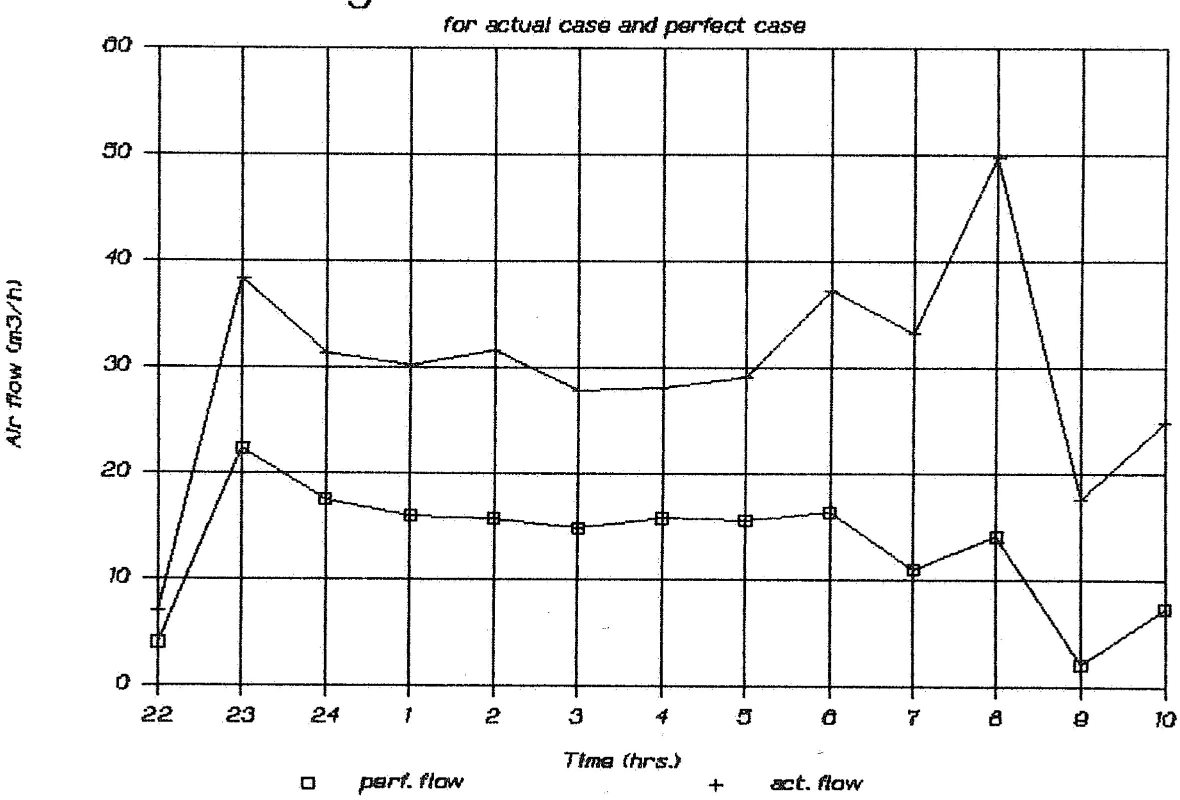


Fig. 11 – Calculated air flows

for actual case

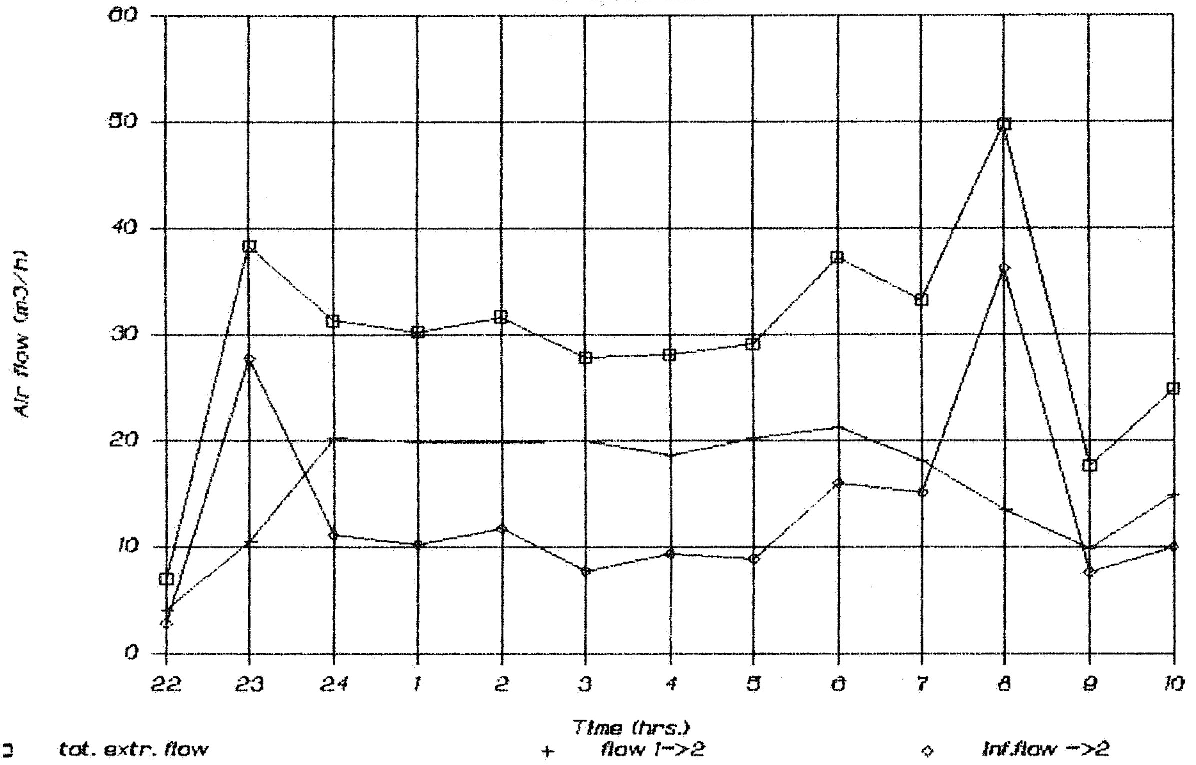
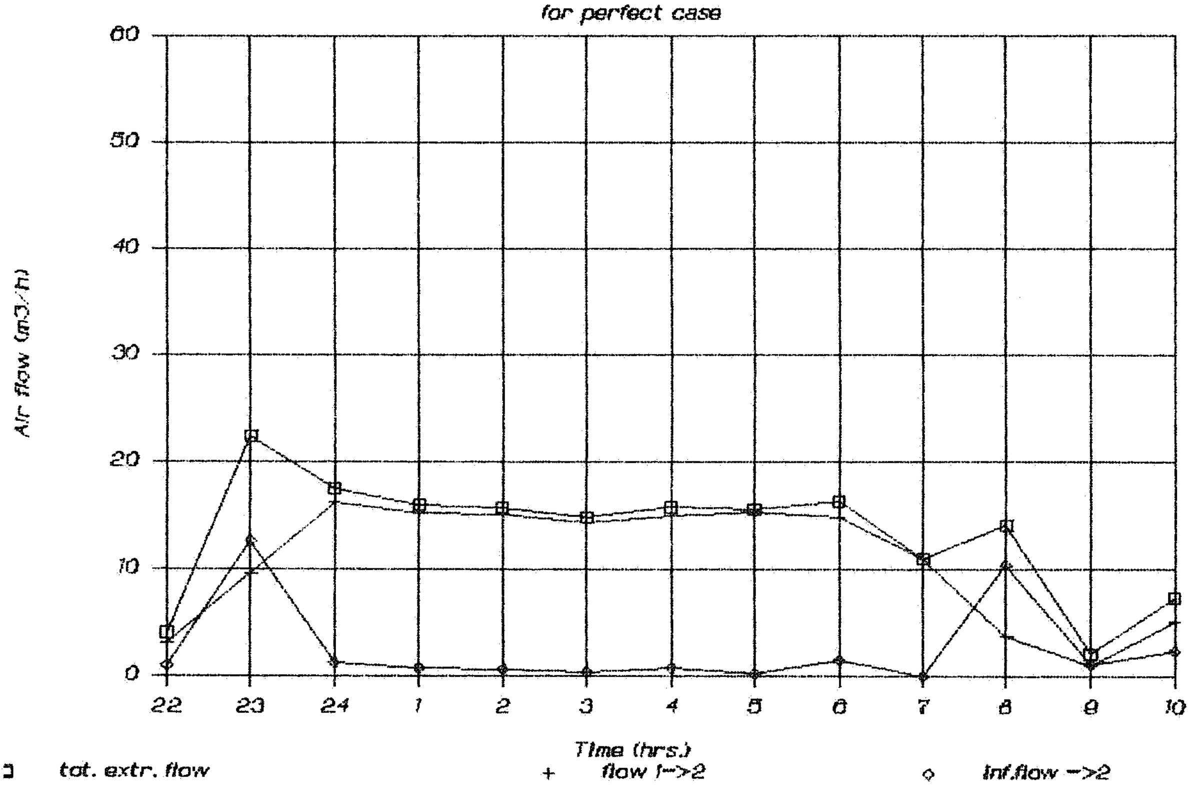


Fig. 12 – Calculated air flows

for perfect case



Discussion

Paper 29

D Bienfait (CSTB, France)

Calculating the air flow rates and humidity levels as you did seems to be the proper way to assess the performance of humidity controlled systems. However, the moisture transfer in furnishings affects dramatically the RH level. Did you take that into account?

G Fracastoro (Politecnico di Torino, Italy)

No. There are other simplifying assumptions in this first approach model. The best thing will be to measure the air flows in a further stage of this study.

VENTILATIOAN SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 30

IEA - ANNEX 14 "CONDENSATION AND ENERGY": ZOLDER CASE STUDY.

Presentation of the final report with special emphasis on the case studies.

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ABSTRACT

The paper summarises the IEA, Executive Committee on Energy Conservation in Buildings and Community Systems, Annex 14 work on Condensation and Energy, a joint research effort of the Federal Republic of Germany, Italy, the United Kingdom, the Netherlands and Belgium, finished end of march 1990.

First the complex relations between mould+ surface condensation, the outside climate, the building fabric, inhabitants behaviour and energy conservation are discussed. Then follows a short overview of the Annex achievements with mayor emphasis on the guidelines and practice results.

These are illustrated by the Zolder case, an example of a problem estate. The causes of the complaints found there, seem typical: social houses with a restricted living space, intensively used, ruinous thermal quality and poor ventilation possibilities. On 3 houses, different curing measures could be evaluated in a systematic way: loft space insulation, inside insulation, double glazing, outside insulation, natural ventilation, demand controlled ventilation. This paper describes the results for 1 of the houses.

0. INTRODUCTION

The IEA- Annex 14 work on mould, surface condensation and energy, generated from a widespread feeling in the 5 countries involved, that too straight forward energy conservation actions during the seventies and early eighties had increased the number of moderate to severe mould cases in the existing housing stock. It was feared that this could enhance any further energy conservation policy. The spread of the problem in the low income housing sector was also quoted as no longer acceptable.

1. A THEORETICAL APPROACH TO MOULD AND SURFACE CONDENSATION

1.1 Condition for mould growth and surface condensation

Mould growth becomes possible when the long lasting mean relative humidity (= the water activity) on a surface remains higher than a threshold value a. This condition can be written as:

$$p \geq a.p' \quad (a \leq 1) \qquad \text{(eq 1)}$$

with p the vapour pressure against and p' the saturation pressure on the surface.

Surface condensation starts each time the relative humidity (RH) on the surface reaches 100%, t.m., each time the vapour pressure p against equals or becomes higher then the saturation pressure p' on the surface:

$$p \geq p' \qquad \text{(eq 2)}$$

1.2 The saturation pressure p'on a surface

The saturation pressure p'in a point on a surface is determined by the local surface temperature Θ_s, Θ_s being given by:

$$\Theta_s = \Theta_e + \tau_{hi} \cdot (\Theta_i - \Theta_e) \qquad \text{(eq 3)}$$

with τ the local temperature factor, coupled to a surface film coefficient h_i, Θ_i the inside reference temperature and Θ_e the outside <sol- air> temperature.

For a flat wall in steady state thermal conditions (= mean thermal situati-on), τ_{hi} becomes an areal property, given by:

$$\tau_{hi} = 1 - U/h_i \qquad \text{(eq 4)}$$

with U the thermal transmittance of the wall and h_i the inside surface film coefficient. For 2D- or 3D- envelope parts in steady state conditions, τ_{hi} is a linear or punctual property, dependant of the specific geometry of the part, the materials combination and the in- and outside surface film coefficients (h_i, h_e). In non steady state conditions, τ_{hi} turns to be time dependant, as shown in fig.1.

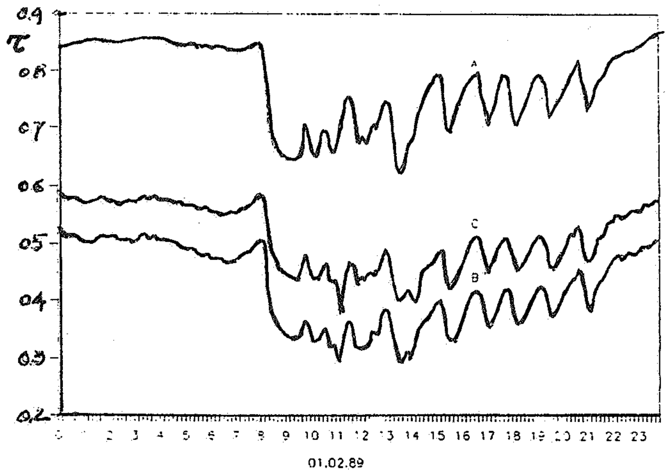

fig 1

the temperature factor of a cavity wall in non steady state

a:middle of the wall

c:corner

b:behind a cupboard

Calculating the temperature factor means: solving in a very detailed way the thermal balance of a building on the level of each thermal zone (heating+ solar gains+ free gains+ enthalpee flow+ convective exchanges/ radiation) and each envelope part (= 1D-,2D- or 3D conduction),with a coupling by the convection+ radiation+ conduction- balance at the inside surfaces (= $\{h_i, \Theta_{ref}\}$).

1.3 The vapour pressure p against a surface.

The vapour pressure p against a surface follows from the hygric balance in each building zone. If one assumes ideal mixing of the zonal air, the balance for zone i becomes:

$$\Sigma(G_{aji} \cdot p_j/RT_i) - p_i \cdot (\Sigma G_{aij})/RT_i + \Sigma\beta_{ki} \cdot A_{ki} \cdot (p'_{ki} - p_i) + \Sigma\beta_{li} \cdot A_{li} \cdot (p_{li} - p_i) + G_{pi} = V_i \cdot dp_i/dt \qquad \text{(eq.5)}$$

with G_{aji} the air flow from zone j to zone i, p_j the vapour pressure in zone j (j= e for the outside), G_{aij} the air flow from zone i to zone j, p_i the vapour pressure in zone i, A_{ki} the surfaces in zone i, where condensation takes place, p'_{ki} the saturation pressure on these surfaces, A_{li} the hygros-copic surfaces in zone i, p_{li} the vapour pressure against these hygroscopic surfaces, β_{ki} and β_{li} the vapour surface film coefficient against the surfaces A_{ki} and A_{li}, G_{pi} the vapour production in zone i and Vi the zonal volume. R is the gass constant of vapour (462 J/(kgK))and T_i the zonal temperature in K.

In this hygric balance, the air flows follow from an interzonal air exchange calculation.

The local vapour pressure p_{li} against each hygroscopic surface is linked to the vapour transport in the surface material by the surface vapour balance:

$$\beta_{li}.(p_{li}-p_i)= [\delta.\ grad(p)]_{sl} \qquad \text{(eq.6)}$$

and the mass balance in the material:

$$div[\delta.\ grad(p)]_{sl}= \delta w_{Hs}/\delta t \qquad \text{(eq.7)}$$

In eq.6 and 7, δ is the vapour conductivity and w_{Hs} the hygroscopic moisture content of the surface layer.
w_{Hs} is given by the suction isotherm.

If steady state or long lasting mean conditions are looked for, then the time derivates in eq. 5 and 7 equal 0 and the vapour pressure against each hygroscopic surface p_{li} becomes the zonal vapour pressure p_i. If further only outside air ventilation plays and G_{aei} is written as $n.V_i$ with n the outside air ventilation rate in zone i, the hygric balance simplifies to:

$$p_i= p_e + \frac{462.T_i.G_p-\Sigma\beta_{ki}.A_{ki}.(p_i-p'_{ki})}{n.V_i} \qquad \text{(eq.8)}$$

If nor surface condensation nor surface drying are present, (eq.8) reduces to the very simple expression:

$$p_i= p_e + \frac{462.T_i.G_{pi}}{n.V_i} \qquad \text{(eq.9)}$$

saying that the mean inside vapour pressure has as lowest value the mean outside vapour pressure and that the difference between both increases when:
- more vapour is produced in the zone (G_{pi} <);
- less ventilation is present (n >);
- the zonal volume is smaller (V >).

Eq.8 adds that, if surface condensation or surface drying are present, the influences of vapour production and ventilation are weakened.
See fig.2.

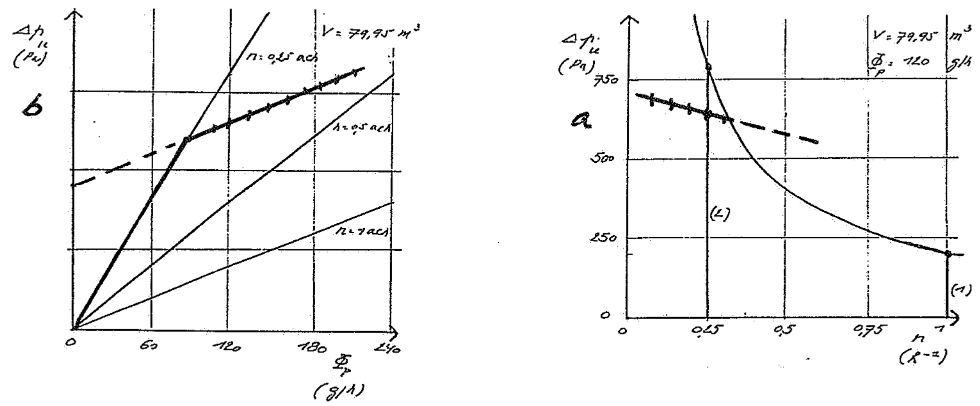

fig 2 the diff. in outside - inside vapour pressure as a function of the mean ventilation rate and vapour production :

— : no cond.
+++: cond. on 7.1 m² of double glazing
-- : drying (Zolder living room)

1.4 Conclusions

The formulas 1 and 2 show in a nutshell that the chance on mould/ surface condensation increases with lower surface temperature Θ_s and higher inside vapour pressure p_i.
The further theory learns that both depend of:

1.- the outside climate

. the temperature Θ_e and the vapour pressure p_e: the lower Θ_e and the higher p_e, the more probable mould and surface condensation. A low temperature and a high vapour pressure however are in conflict: they cannot occur together;

. the wind velocity: the lower, the lower the ventilation rate n and the more probable mould and surface condensation...

2.- the building fabric

. the volume: the smaller V, the more probable mould and surface condensation;

. the thermal quality: the lower the temperature factor, the more probable mould and surface condensation. A low temperature factor implies high U-values, thermal bridging and low surface film coefficients h_i;

. the airtightness: the basic ventilation rate n_b is a direct result of the airtightness of the fabric.The lower n_b, the more probable mould and surface condensation.

. the h_i-value: h_i depends of convection and radiation.The last is to a significantly influenced by the overall thermal quality of the fabric and the 'outside wall area-total wall area'-ratio, in the sense that the worser the thermal quality and the higher the defined ratio, the lower radiation and the more probable mould and surface condensation;

. The inside temperature: Θ_i is fabric coupled, in the sense that,if the dwelling is badly insulated, maintaining a sufficient high temperature reveils too energy consuming and expensive for the inhabitants. The lower the inside temperature, the more probable mould and surface condensation!;

. <u>the vapour production</u>: a high vapour production G_p may be a consequence of other building fabric coupled moisture problems. The higher G_p, the more probable mould and surface condensation.

. <u>the internal finishing</u>: some materials,paints,wall papers are more mould sensitive then others or, the threshold relative humidity 'a' may be lowered by the choice of the finishing solutions (as far as clean)

- **the inhabitants behaviour**

. <u>the inside temperature</u>: depends also of the heating habits.The less heating, the lower Θ_i and the more probable mould and surface condensation;

. <u>the ventilation rate</u>: the lower n, the more probable mould and surface condensation. Inhabitants have a substantial effect on excess ventilation

. <u>the moisture production</u>: the higher G_p, the more probable mould and surface condensation. Living in and using a dwelling inevitably means vapour production. Nevertheless, using it in an unadapted way, may result in too much

These three sets of influencing parameters are interrelated.

2. LINKS WITH ENERGY USE FOR HEATING

The parameter check makes clear that mould and surface condensation are most probable in badly insulated dwellings, t.m. houses with a high basic energy demand. More, avoiding mould in these, asks for a substantial ventilation rate, especially when the dwellings are intensively used. This means: a still higher demand. Economising by lowering the mean inside tempera-ture Θ_i also is counteracted by an increasing ventilation need, if one wants to avoid mould. The result is a total loss of energy demand elasticity.

Insulated houses give complaints as far as problematic thermal bridges are left. These have a net energetical impact, multiplying in negative cases the conductive heat losses with a factor 1.3. To avoid mould on these, also a substantial ventilation is needed, pushing the energy demand further up.

This makes clear that avoiding mould complaints, realising low energy demands and having a good comfort and acceptable IAQ, all point in the same direction:

the construction of well insulated, problematic thermal bridges free, correctly ventilated buildings.

The mould reality adds: 'provided with a heating possibility in each thermal zone'

3. ACHIEVEMENTS OF THE ANNEX

3.1 In general

The main achievement are 4 reports, the first entitled the source book (1), the second being a Catalogue of Material Properties (2), the third handling all Case Studies (3) and the fourth summerising Guidelines and Practice (4).

The source book contains 6 chapters:

1. Material properties (B)
2. Mould (U.K.)
3. Modelling: thermal aspects (I, FRG, NL, B)
4. Modelling: hygric aspects (B)
5. Modelling: heat-air-moisture transfer (NL)
6. Boundary conditions (FRG)

The first gives extended information on the hygrothermal material properties and introduces the catalogue of material properties. The second describes mould and all related aspects. The thirth, fourth and fifth concern the physics involved. They develope the theoretical framework for modelling. The sixth gives practical information on outside climate, vapour production, ventilation, building use...For more information, see (5).
The source book forms the reference for the guidelines report, the translation to practice of the annex work.

3.2 Guidelines and Practice

These have been developed, starting from 4 question:

1.- Condition for mould germination on a surface? (=asking for the value of the threshold-RH a)

2.- Handling of the inside climate in view of mould problem checks: reference temperature, climate charts, climate classes?

3.- Value of the inside surface film coefficient h_i?

4.- Temperature factor τ: exact definition, value?

The answers to these questions are found implicitely throughout the 6 chapters of the source book. Their explicite formulation, given in the Guidelines report, sounds:

Question 1:
Mould germination on a surface becomes very probable, if the monthly mean RH on the surface exceeds 80%. Or:

a= 0.8 on monthly mean basis

Question 2:
As inside reference temperature Θ_i is taken

the AIR TEMPERATURE at 1.7 m height in the center of the zone

The climate chart to be used in judging mould problems, is a 3D-one, with on the x- axis the outside temperature, on the y- axis the inside reference temperature and on the z- axis the difference in inside- outside vapour pressure Δp_{ie}. See fig. 3. This difference stays for the ratio between the vapour production and the ventilation rate (see eq.9)

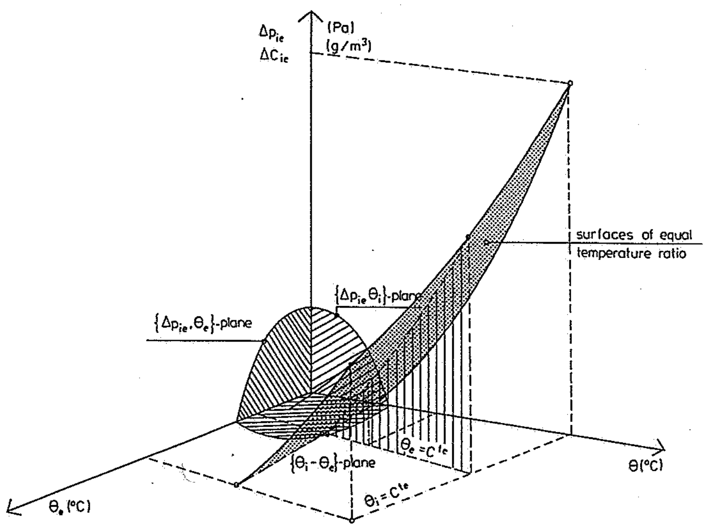

fig 3

the 3-D climate chart with a surface of equal temperature factor or ratio

Question 3:

A clear methodology has been developed to calculate the local inside surface film coefficient h_i, combining heat transfer by convection and radiation and linked to the inside reference temperature Θ_i. If detailed information on the zone is known (thermal quality of all parts, heating system), the methodology allows to calculate $\{h_i,\Theta_i\}$. If not, a set of h_i- values is offered:

upper edges and corners	: 4	W/(m²K)
vertical edges at mid height	: 4	W/(m²K)
lower edges and corners	: 2.9	W/(m²K)
glazing	: 6.7	W/(m²K)
shielded surfaces	: 2	W/(m²K)

all being lower than the value for vertical walls, used in energy and power demand calculations: 8 W/(m^{2}K)!

Question 4:

The temperature factor τ is defined as:

$$\tau = (\Theta_s - \Theta_e)/(\Theta_i - \Theta_e)$$

with Θ_i the inside reference temperature, Θ_s the local surface temperature and Θ_e the outside <sol air> temperature, the three taken on monthly mean basis. τ is used to introduce temperature factor- classes in the climate chart: see fig.3. This is done by calculating the surfaces of equal τ_{hi}- value with the formulas ($x = \Theta_e$, $y = \Theta_i$, $z = p_i - p_e$):

$$\Theta_{si} = x + \tau.(y-x)$$
$$p'_s = p'(\Theta_s)$$
$$z = p'_s - p_e$$

The final performance criterium to avoid mould in new buildings exists in implementing a τ- class. The more severe this class, the more freedom exists in building use and living habits: vapour production, ventilation, heating.
The less severe the τ- class performance, the more restrictions are introduced in the building use: less vapour production, more ventilation, more heating! A save way to come to a τ- class- value is by handling a high but still normal vapour production, the minimal mean ventilation rate for IAQ- requirements and "normal" heating. This leads to:

$$0.65 \le \tau \le 0.7$$

Implementing this performance criterium results, in designing new buildings or curing problem cases, in 4 practice rules

1.A sufficient overall thermal quality: $R \ge 1.5$ to 2 m^2K/W
2.No unacceptable thermal bridges: $\tau \ge 0.65$ to 0.7, U_l as low as possible
3.Implementing a ventilation system, that guarantees $n_{mean} \ge 0.5$ ach
4.Heating possibility in each zone

4. THE ZOLDER CASE STUDY

The Zolder estate has functioned, throughout the annex, as practice laboratory, to judge the problem causes and evaluate retrofitting actions in accordance with the 4 design rules given.

4.1 Generalities

The Lindeman estate, consisting of some 140 dwellings, was build shortly after world war 2 by the "Kempische steenkoolmijnen NV". The 2 story- houses, containing cellar and loft space, were erected with maconry cavity walls, a tiled roof and single glazing in metallic frames. They were coal-fire heated, with a chimney in each room.
In the early eighties, intensive renovating actions were done: a reorganisation of the ground floor, joining the kitchen and little living in 1 larger living room, transforming the washing room in kitchen and adding a bathroom. All chimneys, except one, were demolished and the coal fires removed for a gass fired central heating with radiators in each room. The single glazed, leaky metallic windows were replaced by airtight PVC- windows, double glazed, with an openable part in all rooms, except for the streetside window in the living room. The kitchen was equipped with a hood above the gass fire.
No ventilation system was build in (grids in windows and doors, vertical vents in bathroom and kitchen) nor was any further thermal upgrading, other than double glazing, done (no loft floor insulation, no cavity filling....)
See also fig.4..

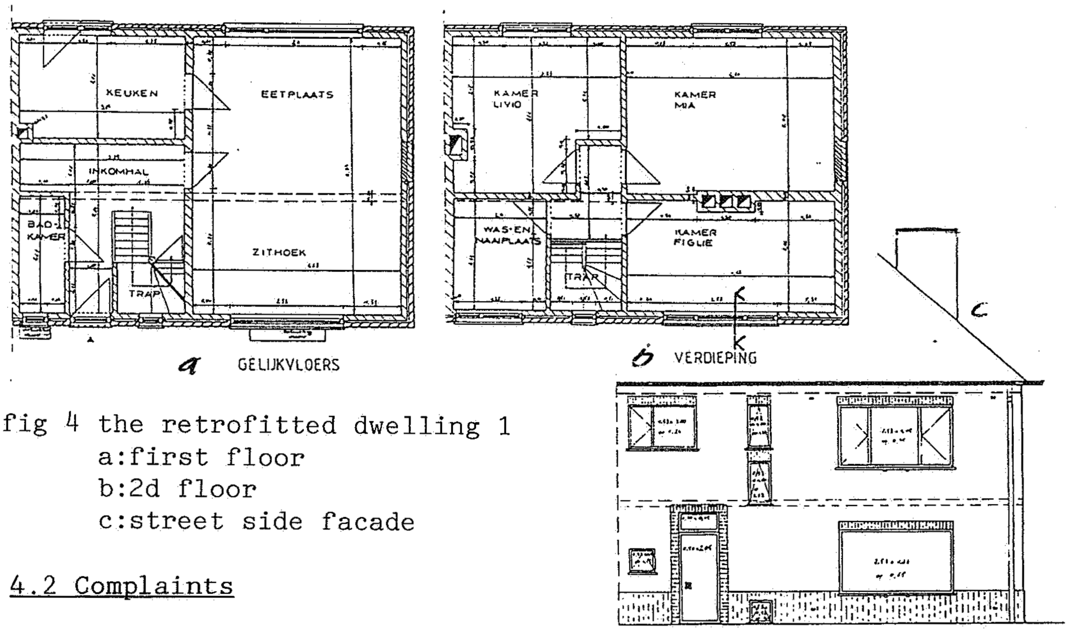

fig 4 the retrofitted dwelling 1
a:first floor
b:2d floor
c:street side facade

4.2 Complaints

Shortly after the retrofit, the first rumours on bad smell, mould and moisture started. An enquiry in 110 houses of the estate, organised in 1987 at the beginning of the Annex, revealed all houses having wet cellar walls, while 75 showed more or less severe mould damage:

Mould in/on:

in	on	number	%
bedrooms	lintels,entire ceiling, corners	64	85
kitchen	ext.wall/ceiling	26	35
bathroom	ext.wall/ceiling	27	36
living room	ext.wall/corners	22	29
hall	ceiling/walls	25	33

Health problems, attributed by the inhabitants to the bad state of their dwelling, were noted in 26 cases out of the 75 with mould.

4.3 Mould species

These were analysed in 3 dwellings. As most important species were found:

species	in ... of the 10 samples
Ulocladium Consortiale	7
Aspergillus Fumigatus	6
Cladesporium Cladesporoides	6
Penicillium Cyclopium	4
Mucor sp.	4

The species on the samples differed from one dwelling to another, from one room to another and in the same room between finishing layers.

4.4 Causes of the problem

2 dwellings were taken as reference, the first- an end of the row house- retrofitted, inhabited by 2 pensioners and having all of the above problems, the second- also an end of the row house- not yet retrofitted, inhabited by a family with 3 little childrens and showing mould in the bedrooms and the bathroom, proving that also before the retrofit, at least some problems existed.
As anti-mould measure, implemented by the housing society, the double glazing in the first dwelling was exchanged for single glass. Curing effects: NONE.

- INSIDE CLIMATE

This was measured in all rooms of dwelling 1:

room	$\Theta_i=f(\Theta_e)$ °C	$p_i-p_e=f(\Theta_e)$ Pa
living room	$17.8-0.08.\Theta_e$ (r^2=0.17)	$562-4.7.\Theta_e$ (r^2=0.25)
kitchen	$19.2-0.09.\Theta_e$ (r^2=0.19)	$506-3.8.\Theta_e$ (r^2=0.14)
bathroom	$13.6-0.01.\Theta_e$ (r^2=0.00)	$478-4.0.\Theta_e$ (r^2=0.20)
hall down	$13.7+0.16.\Theta_e$ (r^2=0.39)	$327-6.3.\Theta_e$ (r^2=0.29)
hall up	$12.2+0.15.\Theta_e$ (r^2=0.10)	$374-4.2.\Theta_e$ (r^2=0.25)
bedroom 1	$11.5+0.16.\Theta_e$ (r^2=0.32)	$356-9.7.\Theta_e$ (r^2=0.35)
bedroom 2	$10.2+0.29.\Theta_e$ (r^2=0.68)	$401-1.0.\Theta_e$ (r^2=0.10)
bedroom 3	$10.4+0.42.\Theta_e$ (r^2=0.41)	$373-3.3.\Theta_e$ (r^2=0.43)
bedroom 4	$8.1+0.38.\Theta_e$ (r^2=0.70)	$399-8.6.\Theta_e$ (r^2=0.39)
cellar	$10.3+0.31.\Theta_e$ (r^2=0.55)	$217-15.\Theta_e$ (r^2=0.29)

If one implements these results on the climate chart, and compares with the reference lines (dayzone: Θ_i= 20°C, nightzone: Θ_i= 15°C/ dayzone: p_i-p_e= $688-21.\Theta_e$) and the lines of equal τ- value, than the conclusions are (fig.5):

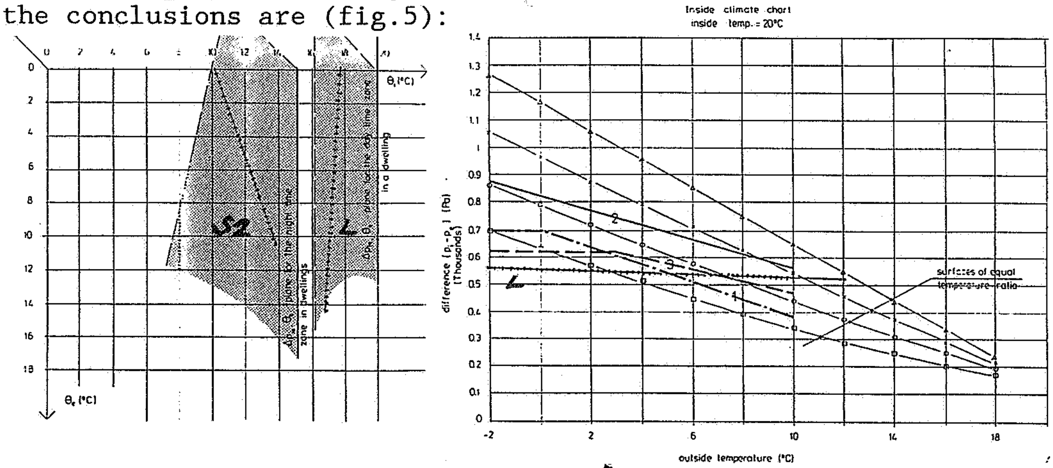

fig 5 the Zolder results against the Δp_{ie} and Θ,-references : colder/high Δp_{ie} (L:living room, S2: sleeping room 2)

- the dwelling is rather poorly heated.This may be caused by a bad insulation quality, making full heating too expensive!;
- the difference in inside- outside vapour pressure is high. This

may be caused by or poor ventilation, or by an important moisture production, or by vapour transfer from the wet cellar to the living space. The last could be excluded after a detailed measuring campaign.The second is rather impossible, because of 'inhabited by only two retired people'.

- to avoid mould problems,a temperature factor ≥ 0.78 is needed in the dayzone. In the nightzone, the RH in the air already exceeds 80%, mould problems being unavoidable!!

- THERMAL QUALITY

The thermal quality was evaluated by calculations and measurements.

<u>Calculated results:</u>

Mean U-value:	dwelling 1:	Compacity	: 1.34 m
		Mean U- value	: U_m= 2.0 W/(m^2K)
	dwelling 2:	Compacity	: 1.32 m
		Mean U- value	: U_m= 2.1 W/(m^2K)

The actual legislation in the Flemish country asks for a mean U-value 0.77- 0.78 W/(m^2K) for social housing with a compacity 1.32-1.34 m, t.m. 3 times lower than present here: the dwellings of the Lindeman estate are in fact of a very poor insulation quality...

<u>Measured results:</u>

. Thermal resistance, U- value, Temperature factor τ, surface film coeff. h_i

dwelling 1		R m²K/W	U W/(m²K)	τ -	h_i W/(m²K)
Living room,	cavity wall	0.47	1.47±0.1	0.75	6.3
	id.,behind cupboard			0.36	
	id.,lower corner			0.45	
	id.,upper corner			0.60	
Sleeping room,	cavity wall	0.66	1.10±0.4	0.72	3.9
	ceiling	0.26	2.23±0.05	0.67	5.8

These results sharply underline the poor thermal quality, 0.36 being the lowest temperature factor. With the measured inside climate, a safety value ≥ 0.78 was needed in the living room!! One also finds surface film coefficients, much lower than 8 W/(m^2K). Remarkable is the U-value of the cavity wall, being lower in the sleeping than in the living room. Explanation: the living room is heated, the sleeping room not. The cavity air, warmed at groundlevel, raises by stack effect to the first floor and diminuishes there the heat flow through the inside leaf. This is translated in an fictitious higher thermal resistance. Applying this to the measuring results, gave a constant thermal resistance 0.27 m^2K/W for the inner leaf and a mean cavity temperature Θ_c= 3.4+ 0.72.Θ_e (Θ_e≤ 10°C).

. Mould and surface condensation

Mould is visibly present on all spots with a temperature factor lower than 0.7. Where the temperature factor is lower than 0.5, surface condensation regularly appears.

- VENTILATION

The ventilation was checked by calculations and measurements.

Calculated results:

Calculations have been done for dwelling 1, situation before the retrofit,coal fire burning and leaky metallic windows:

dwelling 1	wind speed m/s	n h^{-1}
Living room	0	0.3
	4	0.85
	8	2.0
	16	4.9

Measured results:

Measurements were performed on dwelling 1, situation after the retrofit.

Pressuration test

dwelling 1	n_{50} h^{-1}
overall	5.1
hood in the kitchen airtighted	4.8
cellar airtighted	3.9
entry to the loft space airtighted	3.2
joints between doors and floors airtight	2.1
windows airtighted	1.0

These results prove that after the retrofit, the airtightness is so high, that no divice directed natural ventilation cannot be effici-ent!!

Decrease in vapour pressure

From the exponential decrease in vapour pressure in the living room, a ventilation rate 0.17 h^{-1} was deduced. This in fact is a very low value, proving that indeed poor ventilation and not a too high vapour production is the cause of the high difference in inside- outside vapour pressure.
However, because of the overall airtightness and the impossibility to have cross ventilation in the daytime zone, a better ventilation is not possible.

CONCLUSION

The causes of the mould problems in the Lindeman estate are:
- the very poor thermal quality of the houses;
- lack of ventilation, especially in the retrofitted dwellings.

Precisely the last has aggravated the situation after the renovation of the early eighties.

4.5 Remedial measures

The remedial measures in dwelling 1 and 2 focussed on improving the thermal quality and increasing the mean ventilation rate. Only the measures and results for dwelling 1 are discussed, the evaluation of the retrofitting actions on dwelling 2 still going on (loft space insulation, new windows, double glazing, outside insulation, a devise coupled natural ventilation).

Measure 1:
improving the loft floor insulation with 12 cm thick mineral wool slabs.
Effects
1. Mean U-value :decreased from 2.0 to 1.46 W/(m²K) (calculated)
2. R- value ceiling:increased from 0.26 to 3.3 W/(m²K)(measured)
3. τ- value ceiling:increased from 0.67 to 0.85 (measured)
4. h_i-value ceiling:decreased from 5.8 to 1.6 W/(m²K) (measured)
5. air temperatures:INCREASED

	before	after
living room	$17.8-0.08.\Theta_e$ (r^2=0.17)	$17.7+0.09.\Theta_e$ (r^2=0.14)
kitchen	$19.2-0.09.\Theta_e$ (r^2=0.19)	$18.7+0.00.\Theta_e$ (r^2=0.01)
bathroom	$13.6-0.01.\Theta_e$ (r^2=0.00)	$16.1-0.07.\Theta_e$ (r^2=0.07)
hall down	$13.7+0.16.\Theta_e$ (r^2=0.39)	$14.0+0.05.\Theta_e$ (r^2=0.16)
bedroom 1	$11.5+0.16.\Theta_e$ (r^2=0.32)	$13.2+0.12.\Theta_e$ (r^2=0.68)
bedroom 2	$10.2+0.29.\Theta_e$ (r^2=0.68)	$13.3+0.16.\Theta_e$ (r^2=0.69)
bedroom 3	$10.4+0.42.\Theta_e$ (r^2=0.41)	$13.0+0.36.\Theta_e$ (r^2=0.70)
bedroom 4	$8.1+0.38.\Theta_e$ (r^2=0.70)	$13.3+0.07.\Theta_e$ (r^2=0.50)

Through the increase in air temperature, the RH in all rooms dropped (in bedroom 2 from 79 to 68%). The temperature factor of the ceiling in the bedrooms also turned out high enough to avoid further mould germination, althrough the lower h_i-value counteracted to some extend the positive influence of the loft floor insulation.

Measure 2:
replacing single by double glazing.
Effects
1. Mean U-value :decrease from 1.46 to 1.23 W/(m²K) (calculated)
2. air temperature:INCREASED

	before	after
living room(*)	$17.7+0.09.\Theta_e$ (r^2=0.14)	$18.5+0.01.\Theta_e$ (r^2=0.00)
kitchen	$18.7+0.00.\Theta_e$ (r^2=0.01)	$19.1+0.07.\Theta_e$ (r^2=0.14)
bathroom	$16.1-0.07.\Theta_e$ (r^2=0.07)	$18.8+0.37.\Theta_e$ (r^2=0.87)
hall down	$14.0+0.05.\Theta_e$ (r^2=0.16)	$14.2+0.08.\Theta_e$ (r^2=0.50)
bedroom 1	$13.2+0.12.\Theta_e$ (r^2=0.68)	$14.7+0.11.\Theta_e$ (r^2=0.82)
bedroom 2	$13.3+0.16.\Theta_e$ (r^2=0.69)	$14.9+0.07.\Theta_e$ (r^2=0.73)
bedroom 3	$13.0+0.36.\Theta_e$ (r^2=0.70)	$14.6+0.23.\Theta_e$ (r^2=0.76)
bedroom 4	$13.3+0.07.\Theta_e$ (r^2=0.50)	$14.5+0.23.\Theta_e$ (r^2=0.23)

(*): included the effect of inside insulation: see measure 3
3. Difference in inside- outside vapour pressure : no increase,except in the bedroom in use

	before		after	
bedroom 2	$401-1.0.\Theta_e$	(r^2=0.10)	$505-4.2.\Theta_e$	(r^2=0.17)

Through the increase in air temperature, the RH in all rooms,except bedroom 2, stabilised or further dropped. In bedroom 2, there was a slight increase. Reason: the higher difference in inside- outside vapour pressure, caused by the loss of the air drying effects by surface condensation on single glass.

Measure 3:
<u>Internal insulation in the living room: 3 cm PS, with a gypsum board internal lining.</u>
Effects
1. Mean U-value :decrease from 1.23 to 1.15 W/(m^2K) (calculated)
2. R- value wall:increased from 0.43 to 1.23 W/(m^2K) (measured)
3. τ- value wall:increased from 0.75 to 0.82 (measured)
4. h_i-value wall:decreased from 6.3 to 3.6 W/(m^2K) (measured)
5. τ- value adjacent 2 and 3-dimensional parts: from no influence to a slight decrease.
6. air temperature in the living room: see double glazing
7. Difference in inside- outside vapour pressure : no change

Applying inside insulation was not a convincing retrofit: the situa-tion ameliorates on the insulated parts but deteriorates on the adjacent non insulated parts!

Measure 4:
<u>Installing a natural ventilation system: grids in windows and doors, dimensionned in accordance with the Dutch standards NEN 1087 and NPR 1088</u>
Effects
1. air temperature:DECREASED, except in living room and kitchen

	before		after	
living room	$18.5+0.01.\Theta_e$	(r^2=0.00)	$18.7-0.19.\Theta_e$	(r^2=0.21)
kitchen	$19.1+0.07.\Theta_e$	(r^2=0.14)	$21.0-0.30.\Theta_e$	(r^2=0.41)
bathroom	$18.8+0.37.\Theta_e$	(r^2=0.87)	$14.4+0.38.\Theta_e$	(r^2=0.38)
hall down	$14.2+0.08.\Theta_e$	(r^2=0.50)	$13.1+0.33.\Theta_e$	(r^2=0.17)
bedroom 1	$14.7+0.11.\Theta_e$	(r^2=0.82)	$11.1+0.45.\Theta_e$	(r^2=0.48)
bedroom 2	$14.9+0.07.\Theta_e$	(r^2=0.73)	$12.4+0.33.\Theta_e$	(r^2=0.39)
bedroom 3	$14.6+0.23.\Theta_e$	(r^2=0.76)	$13.5+0.22.\Theta_e$	(r^2=0.63)
bedroom 4	$14.5+0.23.\Theta_e$	(r^2=0.23)	$12.8+0.34.\Theta_e$	(r^2=0.17)

3. Diff. in inside- outside vapour pressure :DECREASED (fig 6)

	before		after	
living room	$562-4.7.\Theta_e$	(r^2=0.25)	$525-35.\Theta_e$	(r^2=0.21)
kitchen	$506-3.8.\Theta_e$	(r^2=0.14)	$539-41.\Theta_e$	(r^2=0.36)
bathroom	$488-4.0.\Theta_e$	(r^2=0.20)	$279+5.0.\Theta_e$	(r^2=0.07)
hall down	$327-6.3.\Theta_e$	(r^2=0.29)	$142+1.0.\Theta_e$	(r^2=0.23)
bedroom 1	$356-9.7.\Theta_e$	(r^2=0.35)	$118+16.\Theta_e$	(r^2=0.13)
bedroom 2	$401-1.0.\Theta_e$	(r^2=0.10)	$322+17.\Theta_e$	(r^2=0.17)
bedroom 3	$373-3.3.\Theta_e$	(r^2=0.43)	$187-5.0.\Theta_e$	(r^2=0.30)
bedroom 4	$399-8.6.\Theta_e$	(r^2=0.39)	$131+16.\Theta_e$	(r^2=0.02)

This decrease caused a significant fall in inside RH, finally lowering the chance on persisting mould problems to practically 0. In fact, a control showed that in all rooms with the ventilation system functioning, the surface RH on a spot with τ- value 0.7, dropped definitively under 80%.
The fact that in 5 of the 8 rooms, the $\Delta p_{ie}(\Theta_e)$-line shows, contrary to all theoretical predictions of the necessity of a negative slope, a positive one, could be explained by the hygroscopic moisture release, the first weeks after the implementation of a better ventilation.
A multivariational analysis on all measuring results confirmed that in all rooms, outside air ventilation became, compared to the inter-zonal airflow, more important after than before the system came in use.
Tests in a sleeping room on only peak ventilation by opening the window, proved the deficiency of that option, showing that after closing,the inside vapour pressure quickly returned to his preopening level. Cause: hygroscopic inertia (fig.7)!

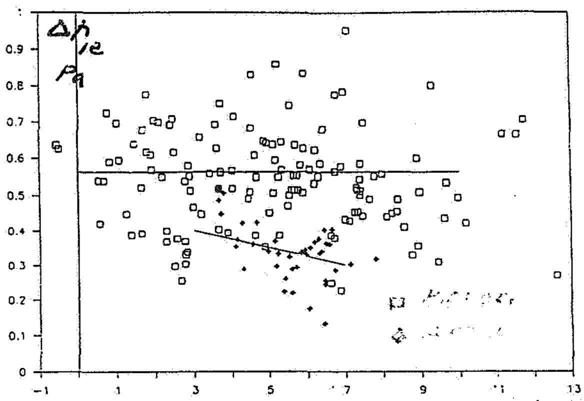

fig 7 living room
Δp_{ie} (Θ_e) before and after the ventilation improvement

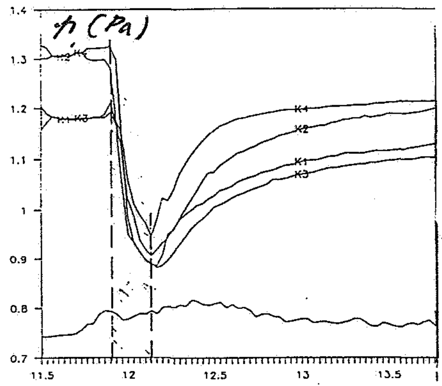

fig 8 effect of peak ventilation on p_i in the 4 sleeping rooms

CONCLUSION
The retrofitting actions on dwelling 1 learned that especially efficient were:
- insulation of the loft space floor. This measure is a must in all dwellings of the estate;
- replacing single by double glazing;
- ameliorating the ventilation.

The importance of a good insulation was stressed by the only partly heating of the house,giving a good insulation the extra impetus of increasing the inside temperature in the non heated and through that mould sensitive rooms. This partly and intermittently heating was the only way for the 2 pensioners to kept the heating bill payable..

5.GENERAL CONCLUSION

The diagnosis and retrofitting work in the Zolder case study proved the correctness of the 4 practice rules to avoid mould problems:

1.A good overall thermal quality: $R \geq 1.5$ à 2 m^2K/W

2.No unacceptable thermal bridges: $\tau \geq 0.65$ to 0.7, U_1 as low as possible

3.Implementing a ventilation solution, that guarantees ≥0.5 ach

4.Heating possibility in each zone

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Discussion
Paper 30

P.Wouters (Belgian Building Res. Inst., Belgium)

Comment

The sensor for the relative humidity is in a part where indoor air is flowing. But at this place this indoor air has not the temperature of the indoor air in the centre of the room, so there are still more problems. But this system is better than to have no system for supply air.

K-J Albers (University of Dortmund, Germany)

Comment

A problem of this and the French system is that they measure the relative humidity of the outside air more than of the indoor air.

J Axley (USA)

Regarding comment solving moisture problems by considering moisture sources, I don't believe any of the morning presenters directly discussed moisture sources.

H Hens (Leuven, Belgium)

Yes and No. Yes, they weren't mentioned. No, because the performance criteria, formulated in the Annex XIV report goes back to a moisture production assumption. In the source book Chapter 5 is devoted to that subject.

W Raatschen (Dornier GmbH, Germany)

In houses where you do not have the possibility of installing better insulation, and where you want to install a humidity driven ventilation system, what practical recommendation would you give to get the proper setpoint for a humidity sensor?

H Hens (Leuven, Belgium)

In cases where bad insulation is combined with no heating, more ventilation, humidity driven or not, may not solve the problems. A good humidity driven system should measure the temperature on a cold, mould sensitive spot, the air temperature and the air humidity. These 3 quantities allow us to calculate the RH next to the surface (= a processor).

C-A Roulet, LESO, Switzerland

In many countries it is shown that air inlets are often taped or closed by the occupants. Is it not time to take this fact into account and act in order to avoid this behaviour? (e.g. hide the inlets, avoid the drafts, inform inhabitants, etc).

H Hens, (Leuven, Belgium)

The air inlets should be placed so that drafts are avoided (at the top of a window, etc.). On the other hand, the possibility always remains that inhabitants seal the inlets. Fear of a too high energy consumption is one of the motives. The only weapon to avoid that behaviour is informing them about the function of the inlets and the necessity for ventilation.

VENTILATION SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 31

The influence of different ventilation devices on the occupants behaviour in dwelling

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Abstract

On basis of several case studies into the ventilation behaviour in dwellings in the Netherlands, it is possible to answer the question whether the type of ventilation device influences the behaviour of the occupants during mild winter periods (5°C).
The dwellings are discerned in three types:

- those with natural ventilation through passive stacks only;
- those with mechanical exhaust ventilation provisions;
- those with balanced mechanical ventilation provisions.

By means of questionaires in each of these dwellings characteristics of the households and the ventilation behaviour has been assessed.
Consequences with respect to the air change rates will be discussed too.

The mean conclusions are that:

- the ventilation by behaviour is related to the type of ventilation device installed in the dwellings;
- there seems to be a 'subjectively preferred' amount of total ventilation from mechanical devices and behaviour taken together;
- in naturally ventilated dwellings and controlled for non occupancy a risk group of 20% is estimated: occupants who tend to ventilate too little: controlled for smoking this is 10%;
- the type of windows or grilles available determine the ventilation more by behaviour than the occupancy of the dwellings in time and number.

Introduction

In the Netherlands much work has been done to study the inhabitant's behaviour with respect to ventilation in occupied dwellings and apartments.
Objectives of these studies are to get insight into the behaviour to calculate air flows and loss of heating energy, to estimate the consequencies for the indoor air quality, to know the circumstances during indoor climate and indoor air measurements and to advise the occupants as well as the manufacturers of the ventilation provision and building architects.
Results of these studies have already been published in (1, 2, 3, 4, 5).

Due to the increased air tightness of the dwellings, the planning of 'open kitchens' and windowless bathrooms and toilets and the exposition of high outdoor noise levels, in nearly all newly built or renovated Dutch dwellings, mechanical exhaust ventilation provisions have been installed to attain the minimal desired ventilation

demand for dwellings of 150 m^3/hs (42 dm^3/s) or 25 m^3/h (7 dm^3/s) per person in the living-room.
Apart from this more and more dwellings are provided with balanced mechanical ventilation provisions in order to save of heating energy, the use of warm air heating systems and to prevent or to decrease problems with moisture and mould.

The question can be posed whether the type of ventilation devices influences the ventilation by behaviour by the occupants of dwellings. On basis of the already available data and an additional field study this has been investigated with respect to a mild **winter situation** with outdoor temperatures of about 5°C and a wind speed lower than 5 m/s.
The resulting air change rates and the possible consequences with respect to subjective experienced health problems will also be discussed.
Three types of ventilation devices in single-family and apartment dwellings are discerned:

- those with natural ventilation through passive stacks only (N=137);
- those with continuously operating mechanical exhaust ventilation provisions (N=133);
- those with balanced mechanical ventilation provisions (N=145).

Also characteristics of the type of households have been introduced as factors: the number of involved persons and hours present in the home.

The use of ventilation provisions by behaviour is twofold: inhabitants 'ventilate' and 'air'. These two types of inhabitants behaviour will be distinguished in this paper.
To **ventilate** by behaviour is defined as providing continuously a certain rate of fresh indoor air by using ventilation grilles and vent lights or by casement windows or pivot windows set ajar. To keep an internal room door opened can also be viewed as a way to ventilate.
It has been calculated (and measured) that at a wind speed lower than 5 m/s and an indoor-outdoor temperature difference of 15°C the ventilation air flow will

vary between 2-10 m^3/h or .5-3 dm^3/s (grilles) to 50 m^3/h or 14 dm^3/s (totally opened vent light) approximately (6)*).

An open internal door (with an open surface of 1.5 m^2) can generate air flows even up to 600 m^3/h if the temperature difference between the hall and the room is 4°C.

Airing is the opening of pivot or casement windows,more than ajar during a certain period. During a mild winter period (+5°C) and a low wind speed this results in air flows varying between 35 m^3/h (or 10 dm^3/s) to 400 m^3/h (110 dm^3/s) depending on the width of opening of the windows.

Figure 1 illustrates the different windows or ventilation provision mainly applied in Dutch dwellings, as well as the indications of the air flows through these provisions during the given mild winter period.

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* The mathematical formula of the air flow stream (in dm3/s) is as follows:

$$q_v = \frac{a}{2} x \sqrt{\frac{1}{100} + 0.001 \text{ x } V_w^2 + 0.0035 \text{ H} + \Delta t}$$

Where a = measured open window surface (in m^2)

$\frac{1}{100}$ = air turbulance

V_w = meteorologic wind speed (m/s)

H = height of open window

Δt = mean temp. difference inside/outside (°C)

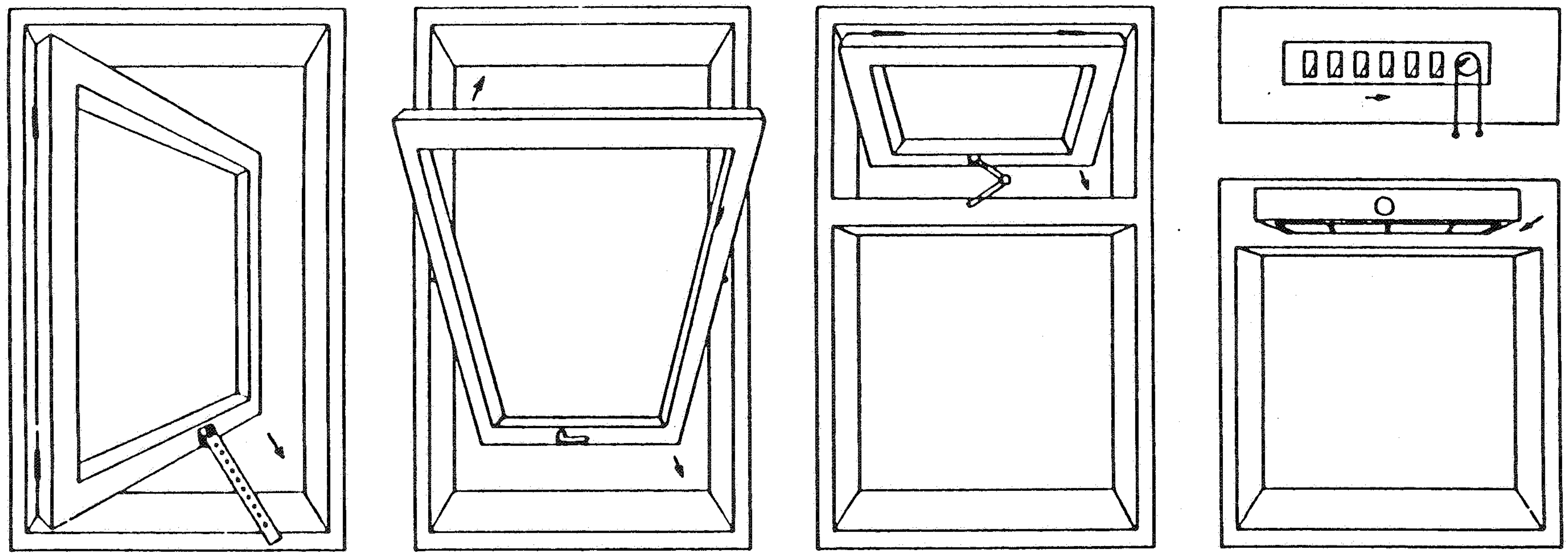

Figure 1 Types of windows or ventilation provisions applied in the different locations of studies

Results

In the framework of this study, per type of ventilation provision the length of time during which the windows and grilles were opened to ventilate or to air has been assessed in three rooms: the living-room, the main bedroom and the kitchen. If more provision are used per room at the same time (e.g. a grill and a window ajar) the provision which is opened longest has been used as yard-stick with the exceptions that if two or more provisions in the same room are opened 0-1 hour or at least 8 hours opened, this has been recoded to one provision which is opened 1-4 hours and 24 hours respectively.
There was also consideration given to the combined situation of airing and ventilation.
The use of the **extractor hood** as a special device above the cooking range has been excluded from the tables. From (7) it appeared that in 61% of the Dutch households this extractor hood is present. Their use is actually restricted to the cooking hours, because the ventilatior noise level is very high.

The tabels 1 to 4 and figures 2 and 3 show the main results discerned on basis of the type of ventilation devices.

hours open	living-room only nat.vent. %(n)	kitchen only nat.vent. %(n)	living/kitch. contin. mech.vent. %(n)	living/kitch. balanced vent. %(n)
0	44 (47)	9 (10)	15 (20)	60 (87)
0-1 (0,5)*	22 (23)	27 (29)	9 (12)	14 (21)
1-4 (2,5)	9 (10)	31 (33)	21 (28)	16 (23)
4-8 (6)	4 (4)	5 (5)	11 (15)	1 (2)
> 8 (12)	10 (11)	15 (16)	9 (12)	2 (3)
24 (24)	10 (11)	12 (13)	35 (46)	6 (9)
mean hours ventilation	4,3	6,0	10,6	2,3
mean hours open if ventilated	7,7	6,6	12,5	5,7
open inward doors (to hall/stair well)	12,9	19,6	7,9	2,7**
ventilator on high speed	-	1	2,7	1,7

* between brackets: hours used for calculation of means
** closed in 89% of the dwellings

Table 1. **Ventilation** by behaviour through windows or grilles in **living-room** and **kitchen.**

hours open	only nat.vent. %(n)	contin. mech.vent. %(n)	balanced vent. %(n)
0	13 (18)	16 (21)	26 (38)
0-1 (0,5)	4 (5)	5 (7)	8 (12)
1-4 (2,5)	3 (4)	9 (12)	6 (9)
4-8 (6)	4 (5)	8 (11)	2 (3)
> 8 (12)	18 (25)	21 (28)	19 (28)
24 (24)	58 (80)	41 (54)	38 (55)
mean hours ventilation	16,5	13,0	11,7
mean hours open if ventilated	19,0	15,5	15,9
open inward doors	8,2(N=52)*	12,4	14,8 (N=133)*

* not known from all studies

Table 2. **Ventilation** by behaviour through windows or grilles in the **main bedroom**

hours open	living-room only nat.vent. %(n)	kitchen only nat.vent. %(n)	living/kitch. contin. mech.vent. %(n)	living/kitch. balanced vent. %(n)
0	40 (42)	64 (68)	52 (69)	74 (108)
0-1 (0,5)	36 (38)	25 (27)	36 (48)	22 (32)
1-2 (1,5)	7 (7)	8 (9)	7 (9)	1 (2)
2-4 (3)	8 (9)	0 (0)	3 (4)	1 (1)
4-8 (6)	6 (6)	2 (2)	2 (2)	1 (1)
>8 (12)	4 (4)	0 (0)	1 (1)	1 (1)
mean hours airing	1,3	0,4	0,6	0,3
mean hours open if airing	2,2	1,0	1,1	1,1

Table 3. **Airing** through windows in **living-room and kitchen.**

hours open	only nat.vent. %(n)	contin. mech.vent. %(n)	balanced vent. %(n)
0	28 (39)	55 (73)	32 (46)
0-1 (0,5)	20 (28)	20 (26)	28 (40)
1-2 (1,5)	9 (13)	2 (2)	9 (13)
2-4 (3)	11 (15)	7 (9)	6 (9)
4-8 (6)	7 (10)	7 (9)	8 (12)
>8 (12)	23 (32)	11 (14)	17 (25)
mean hours airing	3,8	2,0	3,0
mean hours open if airing	5,3	4,4	4,4

Table 4. **Airing** through windows in the **main bedroom.**

Summarized the results in the tables can be formulated as follows:
With respect to what has been defined as **ventilation** it was found that in the dwellings with a continuously operating mechanical ventilation system the **living-open plan kitchen rooms** are ventilated longest, especially where grilles were installed in

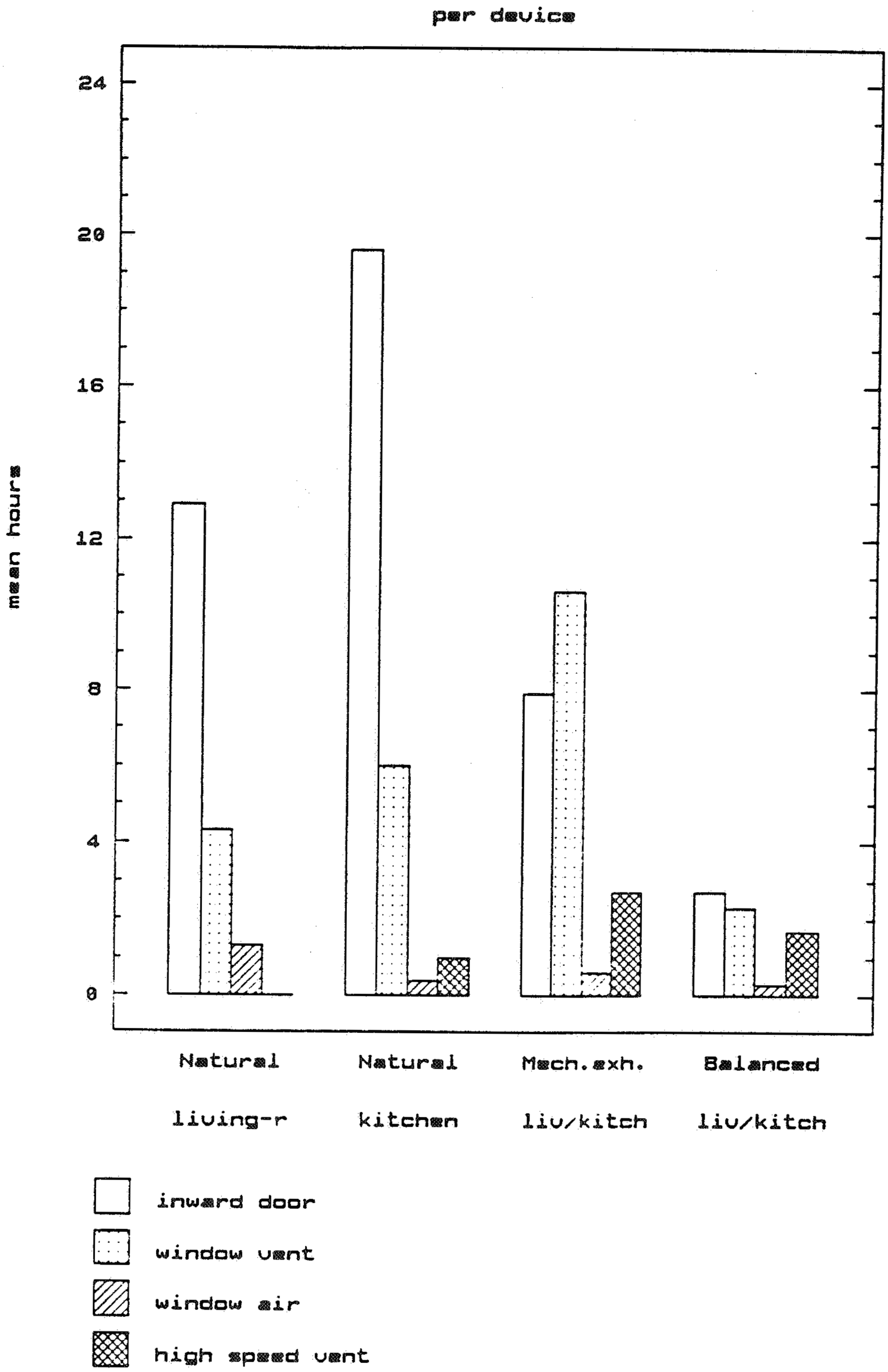

Figure 2 Use of provisions in living-room/kitchen

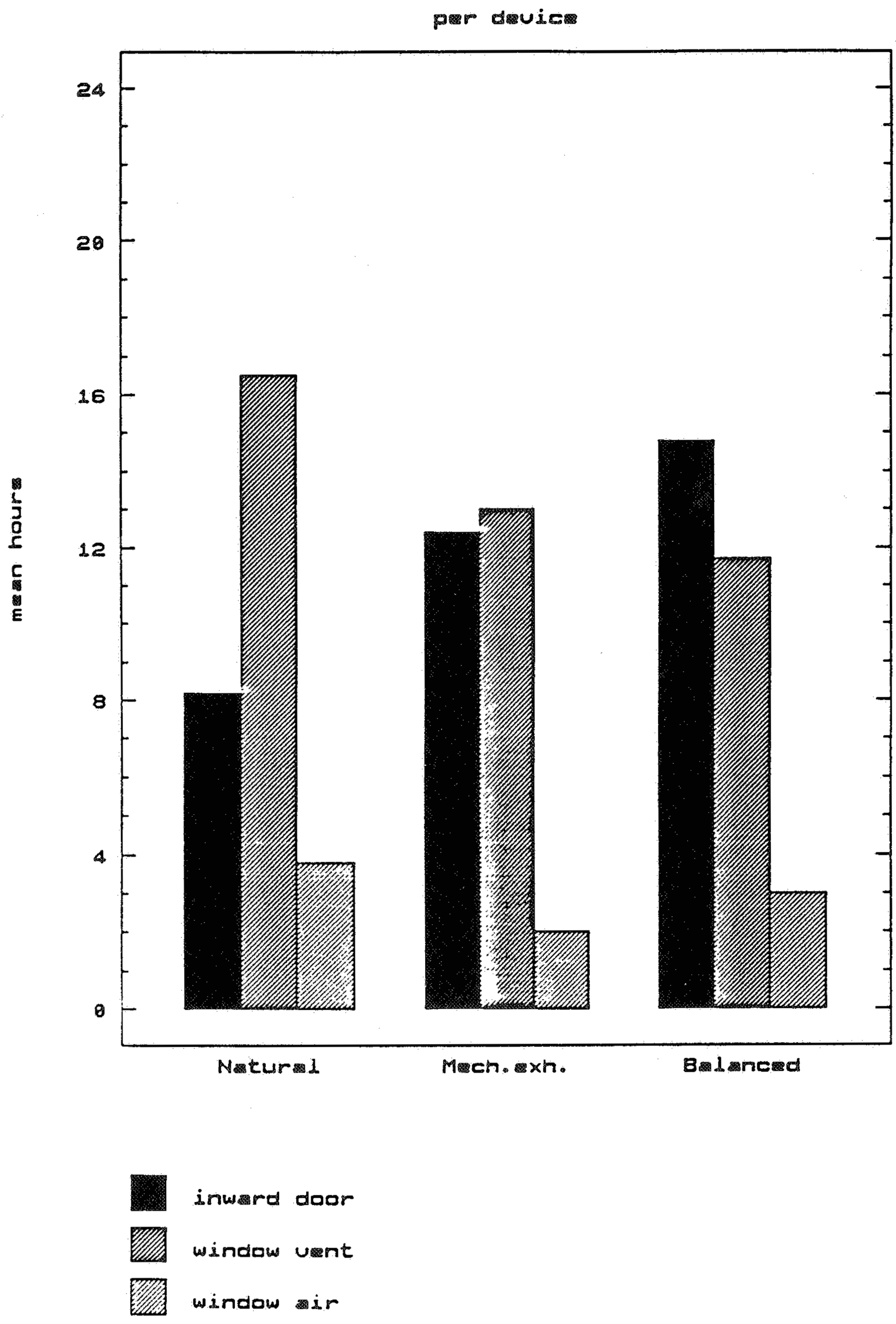

Figure 3 Use of provisions in main bedroom

the dwellings. If grilles are installed in these dwellings they are used by 85% of the households and if opened this is during 12.5 hours on the average.

In the dwellings with a balanced ventilation system, as wished and advised by the developers, the living/kitchen rooms are ventilated least through the window provisions (by 40%) and, if this is done, shortest: 5,7 hours on the average.

The inward doors to the living-rooms were predominantly kept closed too especially. This is in contract with the use of these doors in the only natural ventilated living-rooms, which are opened about five times longer.

The high speed of the ventilators are only used 1 to 2,7 hours per day on the average, mainly at cooking or bathing time. The noise annoyance was the main reason not to use it for a long time. From (8) it is known that an equivalent sound level of 30 dB(A) immitted from a ventilator into the living-room already gives rise to about 25% or sometimes annoyed occupants.

In dwellings (n=19) with a balanced mechanical ventilation system, provided with a Relative Humidity control at 60% RH, it appeared that during winter time this RH high is rarely reached.

In the **main bedrooms** the influence of the type of ventilation device appears to be less, although in the dwellings with a balanced system the use of windows to ventilate is lowest. Furthermore it can be seen that the use of the inward doors clearly negatively correlates with the use of the windows to ventilate.

With respect to what has been defined as **airing** it is found that in living-rooms with natural ventilation the windows are used more to air than those with mechanical ventilation provisions. This is also the case in the bedrooms.

In the dwellings with balanced mechanical ventilation systems the use of windows to air was mostly restricted in the living-rooms, as was the case with the ventilation by behaviour mentioned before. However this was not the case in the bedrooms, which might be explained by the fact that a number of occupants (17%) experienced the air from the system not as "hot fresh" **and** as "too dry".

Evaluating the results it was remarkable remarkably found that the total air flow created by the occupants behaviour appears to depend on the three discerned ventilation types but in such a way that the **total** amount of **air flows** created by dwelling characteristics (airtightness), ventilation systems and ventilation behaviour is more or less of a fairly **constant level**.

The same finding has already been shown in (1, 2, 3) with respect to the different types of windows in the living-room.

If the air flows shown in table 5 are assumed it is calculated that for the living-rooms and the main bedrooms this amount of 'subjectivity preferred' air exchange is in the range of 45-60 dm^3/s or 160-210 m^3/h each during a 24 hours day (see figure 4). Only in the living-room of the balanced ventilated dwellings the ventilation by behaviour was less; 35 dm^3/s or 126 m^3/h, but this was influenced by the

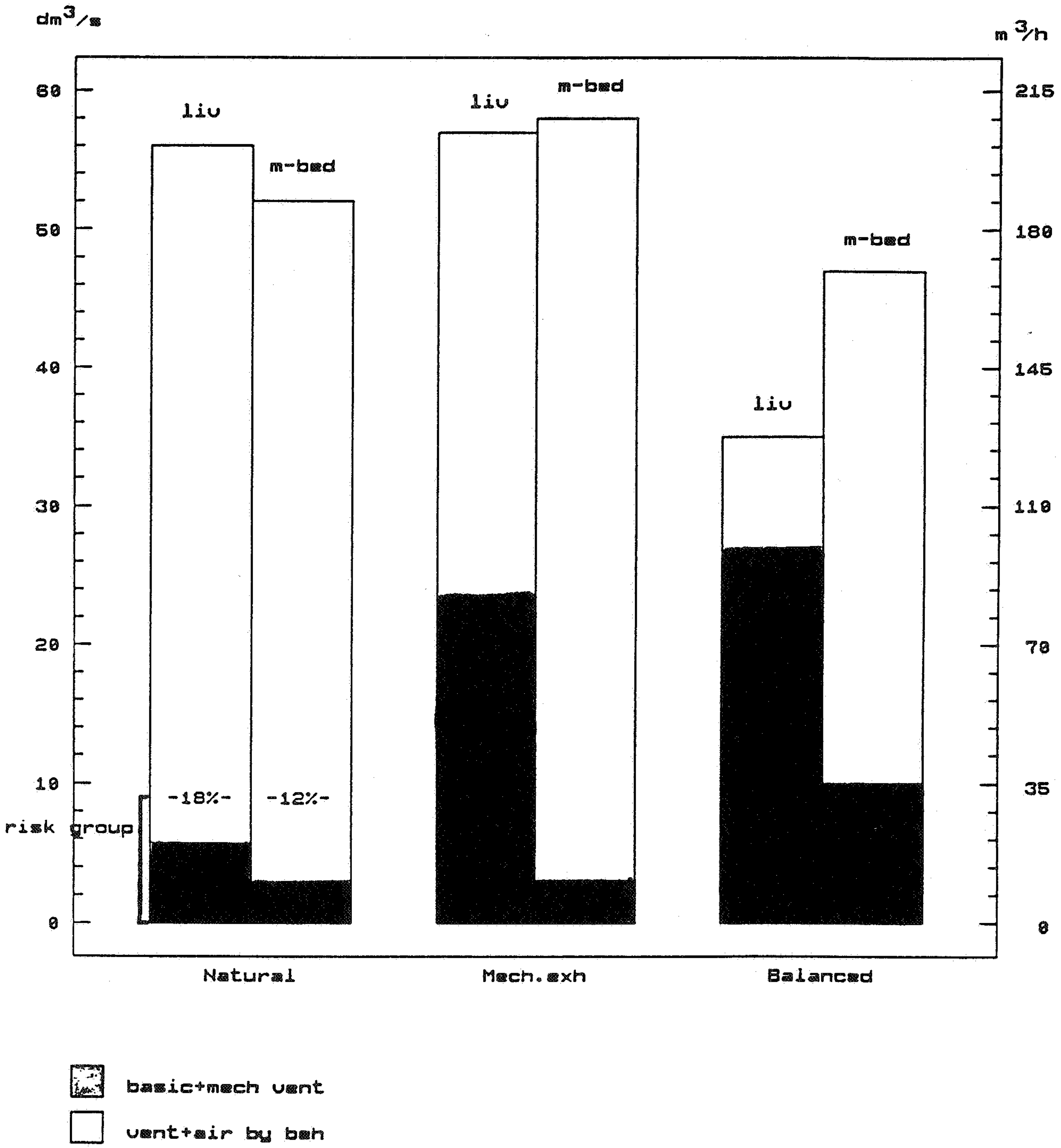

Figure 4 Calculated air flow per type of device

instruction to minimize this behaviour since this would disturb the balanced system.
This suggests that independent of the type of ventilation device there seems to be a certain constant total amount of air exchange which the occupants actively try to obtain by the use of windows or the other provisions in order to feel comfortable.

		m^3/h	dm^3/s
- window opened more than ajar		300	83
- ventlight or window		40	11
- grill opened		10	3
- inward door open and temp. diff.		300	83
- inward door open and few temp. diff.		150	42
- ventilator on low (standard) speed		60	17
- ventilator on high speed (living-room)		100	28
- balanced systeem	living-room	75	21
	bedroom	25	7
- leaks in walls of living-room		20	6
- leaks in walls of bedrooms		10	3

Table 5. Assumed air flows per source

Combined use

In practice the ventilation provisions to air or to ventilate, are used in combination with each other.
To get insight into the proportion of dwellings in which the ventilation by behaviour is low, those dwellings are selected where per day in the different rooms

- a window is opened more than ajar < 1 hour or
- a ventlight or window is opened ajar < 4 hours or
- a grill is opened < 8 hours or
- a ventilator is on high speed < 1 hour or
- an inward door is opened < 1 hour
  (if temperature differences or "fresh air flows" exist) or
- an inward door is opened < 4 hours (if temperature differences or "fresh air flows" are very low).

	nat.vent. %(n)	mech.vent. %(n)	balanced vent. % (n)
living-room	18 (19)	13 (1)*	-
kitchen (separate)	20 (21)	88 (7)*	-
living/open kitchen	19 (6)	7 (9)	60 (67)
main bedroom	12 (17)	13 (17)	5 (7)

* Small number of cases (N=8)

Table 6. Percentages low ventilating by behaviour

Especially in the natural ventilated dwellings these figures (controlled for non-occupancy of the dwellings during the daytime for more than 20 hours per week) give an indication of the population at risk: at first in about 20% of the living-rooms and kitchens the ventilation seems to be too restricted, e.g. in the order of less than 3 dm^3/s or 10 m^3/h on the average during a 24 hours day. In practice in the living-room of this group the air exchange rate might be about 5 dm^3/s higher due to building related air leakages and in the kitchen about 10 dm^3/s due to a ventilation duct and an extractor hood which is excluded here.
In 10(=40%) of these 25 natural ventilated living-rooms more than 5 cigarettes per day are smoked. So if smokers are excluded, the risk group is reduced to nearly 10%.
The results in table 6 also show that in the dwellings with a balanced mechanical ventilation system more than half of the occupants (as instructed) ventilate their living-room and open planned kitchen in a very restricted way.
However this did not appear to be the case in the main bedroom at night.

Health problems

In the case study in Den Bosch with balanced ventilated dwellings, experienced health complaints (extra fatigue, tickling nose and eyes, sneezing, sore throat, bad taste in mouth and, more especially where is smoked, headache and dry skin) as well as the feeling that the indoor air from the inflow valves is "not fresh" are mentioned as reasons for additional ventilation by behaviour (5).

A conclusion which should be stated here, is that in spite of the fact that mechanical ventilation devices are developed to provide a seemingly sufficient amount of air, based on a chosen hygienic standard of 7 dm^3/s per person or 42 dm^3/s per dwelling for instance, in practice this appeared to be unsufficient, since this air is not experienced as fresh enough, or is still to much polluted by indoor air pollutans like ETS, formaldehyde and dust. So additional use of windows and grills to ventilate or to air must be kept possible to allow occupants to react gradually (on a stepwise way) to get rid of their experienced problems.

Influencing factors

In general the finding in (3) is confirmed that the **number of persons** in a household **does not** influence the length of ventilation or airing by means of windows or grills in the kitchens and living-rooms (see Table 7). However a positive relation has been found between the use of the **high speed** of the ventilator (if this provision is present) and the number of occupants per dwelling. The high speed is used by about 25% of the 1-2 person households, 35% of the 3 person households and

50% of the 4 or more person households. In the main bedrooms (which are always used for sleeping) the use of the provisions to ventilate and to air is independent of the number of persons too, but as a matter of course in total the use of bedroom windows per dwelling will rise if more bedrooms are used for sleeping. On the average a window in a childrens bedroom is opened about half as long as in a parents bedroom.

CLIMATOLOGICAL FACTORS:

* less ventilation when:
  - the outdoor temperature is lower
  - windows are less oriented to the sun
  - windows are more oriented to the prevailing wind direction

HUMAN FACTORS:

* daily patterns in household activities
* less ventilation when:
  - the inside temperature is higher (as preferred)
  - the occupants are less energy conscious
  - the occupants prefer less fresh air
  - no respiratory diseases occur

ENVIRONMENTAL FACTORS

* less ventilation when:
  - condensation or mold growth are not perceived
  - the moisture production in the dwelling is lower
  - noise or odor annoyance has been experienced from outside

ARCHITECTURAL FACTORS:

* type of rooms
* type of ventilation provisions and heating systems
* less ventilation when:
  - the basic natural ventilation is higher through cracks
  - the volume of rooms is smaller
  - the occupants are conscious of an installed mechanical ventilation (which is not noisy)

Table 7 List of factors influencing airing and ventilation of dwellings by behaviour (from (3))

The hypothesis that **non occupancy** of the dwellings during the daytime of more than 20 hours per week influences the rate of **ventilation** by behaviour has not been confirmed in general. The type of ventilation provisions and the possibility

of burglary are the dominant factors here: where dwellings are provided with grills, especially in the bedrooms these grills opened even longer when no one is st home. Only in the living rooms of the dwellings with a balanced mechanical ventilation system has a significant different use of the provisions to ventilate been found in relation to the presence at home.
Also with respect to **airing** the relation with absence appeared weak in general. Difference in behaviour only exists in the living room, but not in de bedrooms: where dwellings are often unoccupied during daytime, less than one hour per day is aired in 95% of the dwellings and is never aired in 70% of the dwellings. In more or less continuous occupied dwellings these percentages are 80% and 50% respectively.

Conclusions:

Summarized, the data presented here lead to a number of conclusions:

- With respect to **ventilation by behaviour** a strong relation has been found with respect to the type of ventilation provisions installed in the dwellings and the use of windows or grilles in terms of hours of opening. With respect to **airing** this relation is weak, especially in the bedrooms.
- From calculations based on the occupants' use of windows and grilles to ventilate and air it appeared that, independent of the type of ventilation provisions and air tightness of the dwellings, there seems to to be a certain **constant total** amount of air exchange, induced by behaviour **and** by mechanical provisions together, which the majority of the occupants try to obtain in order to feel comfortable:

  For the living-rooms this average amount of 'subjectively preferred' air exchange is about 175 m^3/h or 50 dm^3/s.
- However in about 20% of the **only natural by ventilated** dwellings and controled by non-occupancy the ventilation by behaviour appeared to be restricted to about 3 dm^3/s on the average during a 24 hours day in the living-rooms and kitchen. This flow can be increased by 5 dm^3/s due to air leakages in the living-room and with about 10 dm^3/s in the kitchen due to a ventilation duct or extractor hood, but even in those cases and, in view of the indoor air quality, this exchange rate is too low in most cases, especially in the living-room where people smoke, which is the case in 40% of them. The non smoking risk group is nearly 10%.
- In dwellings with a balanced mechanical ventilation system which in principle garantees an air flow of 42 dm^3/s (or 150 m^3/h) through the dwelling as a whole, a ventilation amount which in the Dutch standard is viewed as suffi-

cient, health and comfort problems can be expected due to indoor air sources, if additional use of windows and grills to ventilate by the occupants themselves is not possible.

- From the results in the case studies it appeared that the **number** of occupants **does not**, and **non occupancy** of the dwellings during daytime periods longer than 4 hours per day **only slightly** does influence the ventilation by behaviour. More determining than non occupancy is the type of window or grilles which are available: its user-friendliness, its possibility for stepwise regulation and its burglary preventing quality.

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Discussion
Paper 31

J. Van der Maas (LESO, Switzerland)

Can you say something more about the effect on the occupants behaviour of the instructions the occupants received (e.g. Balanced system).

W. de Gids (TNO, Netherlands)

The first few days after instruction people seemed to change their behaviour. After a few weeks they seemed to stick on their old behaviour again. Which means no change at all. Only a small group, namely the energy concious people had a slight change in the first few days.

VENTILATION SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 32

Numerical and Experimental Study on Flow and Diffusion Field in Room.

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SYNOPSIS

Turbulent flow fields of velocity and diffusion in several types of mechanically ventilated rooms are precisely analyzed both by model experiment and by numerical simulation based on the k-ε two-equation turbulence model. The detailed analyses of contaminant diffusion by simulation make it possible to comprehend clearly the structures of velocity and diffusion fields in rooms.

The flow fields in such rooms, as analyzed here, are mainly characterized by the inflow jet and the rising streams around it. The combination of one jet and rising streams forms a 'flow unit.' The total velocity field and the resulting diffusion field of contaminant in a room are well modeled as serial combinations of these 'flow units.'

Room air distribution is greatly affected by the arrangement of supply openings and, possibly, exhaust openings. The influence of those arrangements on the flow fields is studied. When the number of supply openings is decreased, the flow units corresponding to the eliminated supply openings vanish and the remaining flow units expand. A change in arrangement or in the number of exhaust openings hardly affects the entire flow field; however, such changes often have a large influence on the contaminant diffusion field.

Apparatus placed in a room has a great influence on the flow field. The air flow distribution and the contaminant diffusion field in the room with flow obstacles in various arrangements are also analyzed. On the whole, the influence of flow obstacles on the entire flow field is limited to a certain degree, however, the influence on the 'flow unit' is significant.

NOMENCLATURE

C_μ, C_1, C_2 = empirical constants in the k-ε turbulence model (cf. Table 2)
C = mean contaminant concentration
C_o = representative concentration defined by that of exhaust opening
E = empirical constant in log law, 9.0 in case of smooth wall
h = interval of finite difference
h_1 = length from the solid wall surface to the center of the near wall fluid cell
k = turbulence kinetic energy
l = length scale of turbulence
L_0 = representative length defined by width of supply opening
P = mean pressure
q = contaminant generation rate
Q = air exchange volume
SVE = scale for ventilation efficiency
SVE1, spatial average concentration
SVE2, mean radius of diffusion
SVE3, concentration in case of uniform contaminant generation throughout the room
U_i, U_j = components of mean velocity vector
U_o = representative velocity defined by inflow jet velocity
ε = turbulence dissipation rate

κ = von Karman constant, 0.4

ρ = fluid density

ν = molecular kinematic viscosity

ν_t = eddy kinematic viscosity

$\sigma_1, \sigma_2, \sigma_3$ = turbulence Prandtl/Schmidt number of k, ε , C (cf. Table 2)

1. INTRODUCTION

The airflow pattern in a room is mainly determined by the shape of the room and the number of supply openings. Therefore, in order to accurately design the airflow for such a room, one should analyze each room independently. However, it is also well-known that the flow fields of such rooms share many common characteristics, especially when the supply openings are set on the ceiling. In this study, the flow fields and resulting diffusion fields of contaminant in rooms, where supply openings are located on the ceiling, are precisely analyzed.

The distribution of contaminant diffusion is a very useful means by which to comprehend the diffusion field. On the other hand, the diffusion field alone cannot give effective information for evaluating ventilation efficiency because, when given two patterns of contaminant diffusion, it is often difficult to judge which one is better. For this purpose, we need a simple index that can express the characteristics of the diffusion pattern as a quantitative value. Kato and Murakami (1988) proposed the new concept of ventilation efficiency for the diffusion fields of contaminant and presented a method by which to estimate the different distributions of contaminant concentration as a whole and to evaluate the difference of ventilation efficiency. We will here briefly summarize the new concept of ventilation efficiency and apply it to the diffusion fields in the rooms under discussion.

As stated above, the airflow pattern in a room is mainly determined by the shape of the room and by the number of supply openings. In this study, the influence of the arrangement and number of supply and exhaust openings on the flow and diffusion fields in rooms and the effects of the flow obstacle are analyzed from the viewpoint of flow structure and the ventilation efficiency.

Numerical simulation of turbulent air flow allows us to analyze the flow and diffusion fields in a room precisely (Murakami et al. 1987). It is confirmed that the correspondence between experiment and numerical simulation is fairly good for both velocity vectors and contaminant concentration. Analysis by numerical method is very powerful in parametric study is. The influences of various flow conditions on the flow and diffusion fields are analyzed parametrically here. Thus, in this paper, flow fields and contaminant diffusion fields are examined mainly by means of numerical simulation.

2. MODEL ROOMS ANALYZED

Eight types of rooms are used for analysis in this study. In table 1, the specifications of these rooms are presented. These rooms may be regarded as the models of conventional flow type clean rooms. Length and velocity scales are non-dimentionalized by dividing by the representative values, the width of supply opening L_0, and the supply air velocity V_0, respectively. Figure 1 shows the plans and sections of these 8 types. The source points of contaminant are located under the supply opening, near the wall, and at the center of the room, respectively. Their height from the floor is set equally at 1.25 in dimensionless value. Another source point of contaminant is located in front of the exhaust opening, where its height from the floor is 0.5. Since the contaminant in this study is assumed to be of passive scalar quantity, and thus of no effect on momentum equations, its transportation or diffusion is fully controlled by the air flow. Flow fields and resulting diffusion fields are assumed to be in steady states. The contaminant generation rate is also assumed to be constant.

Table 1. Specifications of model rooms used

Types of Model Clean Room	Dimension of Plan (*1)	Height of Ceiling (*1)	Number of Supply openings	Number of Exhaust openings	Supply Air Velocity (*2)	Remarks
TYPE 1	5 X 5	4.5	1	4	1.0	basic type:the smallest room
TYPE 2	8 X 8	4.5	4	4	1.0	basic type
TYPE 3	8 X 8	4.5	4	4	1.0	2 exhaust openings are closed
TYPE 4	11X11	4.5	9	4	1.0	basic type:the largest room
TYPE 5	11X11	4.5	6	4	1.0	3 supply openings are closed
TYPE 6	11X11	4.5	5	4	1.0	4 supply openings are closed
TYPE 7	11X11	4.5	4	4	1.0	5 supply openings are closed
TYPE 8	11X11	4.5	1	4	1.0	8 supply openings are closed

*1 : dimensionless value (divided by the width of supply opening L_0 (0.6m))
*2 : dimensionless value (divided by the supply air velocity V_0 (1.0m/s))

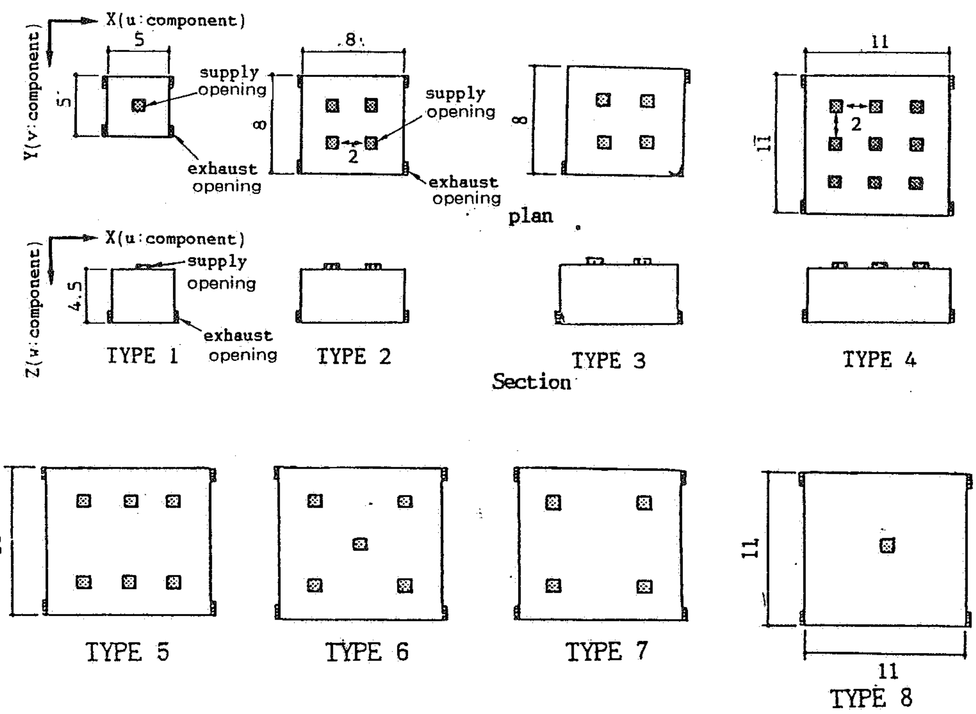


Figure 1. Plans and sections of model rooms (Length scale in this figure is non-dimensionalized by the width of supply outlet Lo)

3. MODEL EXPERIMENTS

Model experiments were conducted using scale models. The representative length, the width of supply opening L_0, was set as 0.1 m in all room models. The velocity of the jet from supply opening U_0 was set at 6 m/s. The Reynolds number of the inflow jet $U_0 L_0 / \nu$ is 4.2×10^4.

Air velocity is measured by means of tandem type, parallel hot-wire anemometer, which can discern the vector components of turbulent flow (Murakami et al.1980).

The distribution of contaminant concentration is investigated by means of a tracer gas diffusion experiment. Since ethylene (C_2H_4),whose density is nearly the same as that of air, is used as the tracer, the buoyancy effect of the tracer can be disregarded. Its concentration is measured by means of F.I.D. gas chromatography.

Table 2. Two-Equation Model (Three-Dimensional)

$\frac{\partial U_i}{\partial X_i} = 0$	(1) Continuity equation
$\frac{\partial U_i}{\partial t} + \frac{\partial U_i U_j}{\partial X_j} = -\frac{\partial}{\partial X_i}\left(\frac{P}{\rho} + \frac{2}{3}k\right) + \frac{\partial}{\partial X_j}\left(\nu_t\left(\frac{\partial U_i}{\partial X_j} + \frac{\partial U_j}{\partial X_i}\right)\right)$	(2) Momentum equation
$\frac{\partial k}{\partial t} + \frac{\partial k U_j}{\partial X_j} = \frac{\partial}{\partial X_j}\left(\frac{\nu_t}{\sigma_1}\frac{\partial k}{\partial X_j}\right) + \nu_t S - \varepsilon$	(3) Transport equation for k
$\frac{\partial \varepsilon}{\partial t} + \frac{\partial \varepsilon U_j}{\partial X_j} = \frac{\partial}{\partial X_j}\left(\frac{\nu_t}{\sigma_2}\frac{\partial \varepsilon}{\partial X_j}\right) + C_1\frac{\varepsilon}{k}\nu_t S - C_2\frac{\varepsilon^2}{k}$	(4) Transport equation for ε
$\nu_t = k^{1/2} l = \left(C_\mu \frac{k^2}{\varepsilon}\right)$	(5) Equation for deciding ν_t
$\frac{\partial C}{\partial t} + \frac{\partial C U_j}{\partial X_j} = \frac{\partial}{\partial X_j}\left(\frac{\nu_t}{\sigma_3}\frac{\partial C}{\partial X_j}\right)$	(6) Concentration equation
here $S = \left(\frac{\partial U_i}{\partial X_j} + \frac{\partial U_j}{\partial X_i}\right)\frac{\partial U_i}{\partial X_j}$, $C_\mu = 0.09$, $\sigma_1 = 1.0$, $\sigma_2 = 1.3$, $\sigma_3 = 1.0$, $C_1 = 1.44$, $C_2 = 1.92$	

Table 3. Boundary Conditions for Numerical Simulation

(1) Supply Outlet boundary	:	$U_t = 0.0$, $U_n = U_{out}$, $k = 0.005$, $l = 0.33$, $C = 0.0$ suffix t: tangential component, n : normal component U_{out} : Supply opening velocity, $U_{out} = 1.0$
(2) Exhaust Inlet boundary	:	$U_t = 0.0$, $U_n = U_{in}$, $\partial k/\partial Z = 0.0$, $\partial \varepsilon/\partial Z = 0.0$, $\partial C/\partial Z = 0.0$ U_{in}: Exhaust opening velocity, $U_{in} = 2.25$ in case of TYPE 4
(3) Wall boundary	:	$\partial U/\partial Z_{z=0} = m U_{t\,z=h} / h$, $U_n = 0.0$, $\partial k/\partial Z = 0.0$, $\partial C/\partial Z = 0.0$ ε term in k equation : $\varepsilon_{z=h} = [C_\mu k^{3/2}_{z=h}]/[C_\mu^{1/4}\kappa h] \cdot \ln(E h (C_\mu^{1/2} k)^{1/2}/\nu)$ ε equation : $\varepsilon_{z=h} = [C_\mu k^{3/2}_{z=h}]/[C_\mu^{1/4}\kappa h]$ h : Length from the wall surface to the center of the adjacent cell m : 1/7, Power law of profile $U_t \propto Z^m$ is assumed here. E : 9.0, a function of the wall roughness (for a smooth wall) ν : 1/Re, Kinematic viscosity κ : 0.4, von Karman constant
(4) Finite difference Scheme	:	Time marching : Adams Bashforth Scheme (second order) Convective terms of Ui, k, ε, and C : Quick Scheme (second order)

(Values are expressed in non-dimensional form)

4. NUMERICAL SIMULATION METHOD

Model equations (3-D k- ε two-equation turbulence model) are given in Table 2. The boundary conditions are tabulated in Table 3. Various types of boundary conditions at the solid wall have been devised in various problems of engineering applications of the numerical method. Some boundary conditions were derived using the concept of log-law (Launder et al.1974; Chieng et al.1980; Rodi 1984). The solid wall boundary condition derived from the power law of velocity profile is used (authors cf. Table 3). The latter is very simple and has given successful results (Murakami et al.1987; Murakami et al.1988).The difference in the simulation results between these log-law types and the power law type has been examined and confirmed to be negligibly small(Kato et al.1988b; Nagano et al.1988). In this context,the solid wall boundary condition derived from the power law of velocity profile is used here.

The flow fields in rooms divided into the mesh systems shown in Figure 2 are solved by the finite difference method. The numerical simulation method follows that given in Murakami et al.(1987). After the room flow fields are obtained, the contaminant diffusion fields are calculated using such flow field properties as the distribution of velocity vectors and eddy viscosity. The simulated flow fields are not entirely steady and symmetrical due to numerical instability. However, asymmetry of flow fields is very slight and can be disregarded. The calculated contaminant diffusion fields are thus also slightly asymmetric in accord with the flow fields.

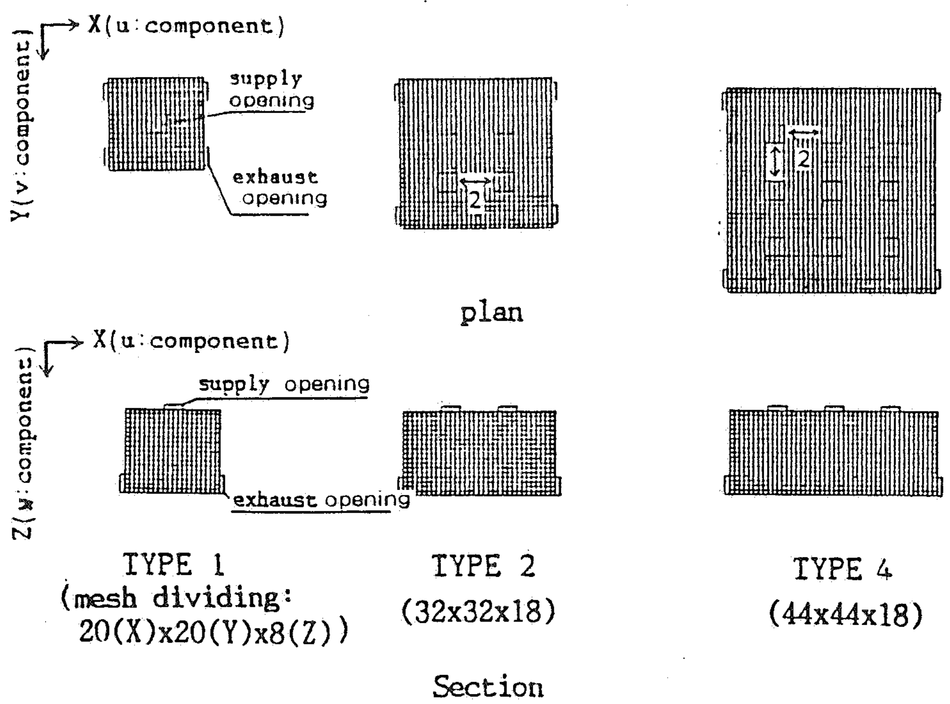
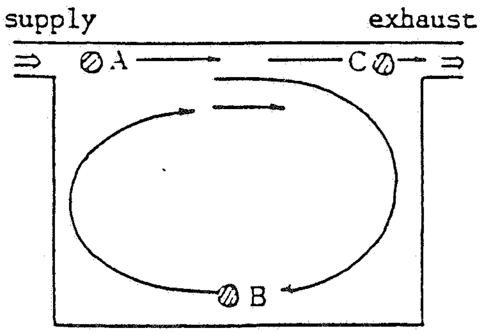


Figure 2. Mesh systems

5. EXPRESSION METHODS OF CONTAMINANT DIFFUSION FIELD AND DEFINITION OF SVE1,2,3

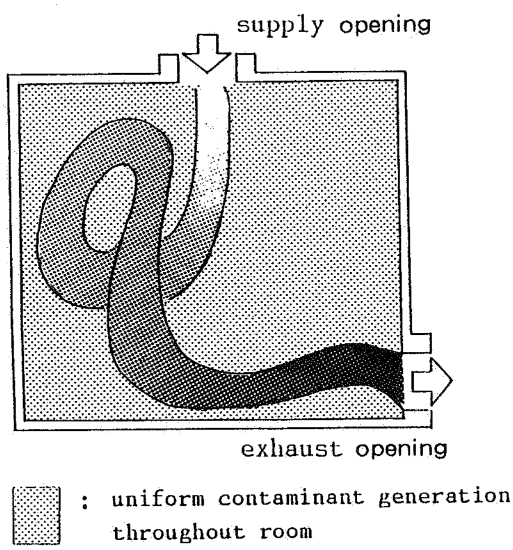
In this study, contaminant diffusion fields are expressed by four methods:

1. Distribution of contaminant concentration in case of point source: this distribution allows intuitive comprehension of the contaminant diffusion field in a room.
2. Spatial average concentration: the first Scale of Ventilation Efficiency (SVE1). This value is proportional to the average time the



- Contaminant source point A
  mean radius of diffusion (SEV2) : large
  spatial average concentration (SEV1)
  ≈ concentration in exhaust
- Contaminant source point B
  mean radius of diffusion : large
  spatial average concentration
  > concentration in exhaust
- Contaminant source point C
  mean radius of diffusion : very small
  spatial average concentration
  ≪ concentration in exhaust

Figure 3. Change of diffusion field according to the change of source position



- SEV3 is defined as the value of concentration at each point.
- It is an index of the travelling time for an air mass blown from supply openings to arrive at each point.

Figure 4. SVE3 : Diffusion field in case of contaminant generated uniformly throughout the room

contaminant is present in the room and indicates how quickly the contaminant generated in the room is exhausted by the flow field. This condition may be easily explained as follows. When the generated contaminant takes more time to be convected to the exhaust opening, it is certain that there exists more contaminant within the room in spite of the constant generation and constant exhaust of contaminant. Figure 3 shows the change of the value of SVE1 according to position of the source. If the contaminants are generated near the exhaust, at point C, the contaminants are smoothly exhausted and the value of SVE1 is expected surely to be very small. However, if the contaminants are generated within the recirculating flow, at point B, the contaminants are likely to stay longer in the room and the value of SVE1 will increase.

3. Mean radius of diffusion: the second Scale of Ventilation Efficiency (SVE2). This value represents the average spatial diffusion. Figure 3 shows the change of the value of SVE2 according to position of the source. If the contaminants are generated near exhaust, at point C, the contaminants are exhausted without diffusion. In this case, the value of SVE2 is expected surely to be very small. However, if the contaminants are generated at the supply opening, at point A, the contaminants spread and diffuse throughout the room. The value of SVE2 is expected to be the largest.

4. Concentration in case of uniform contaminant generation throughout the room: the third Scale of Ventilation Efficiency (SVE3). At a given point in a room this value is proportional to the mean travelling time of the supply air to that point. High value of this scale indicates a high possibility of air contamination, because the air mass must have travelled a long way from the supply opening to that point. This situation is illustrated in Figure 4.

The details of these scales are described by Kato et al.(1988a)

6. DIMENSIONLESS STUDY OF CONCENTRATION (MODELS 1 AND 2)

In this study, concentration is made dimensionless by dividing by representative value of C_0, the mean contaminant concentration averaged over all exhaust openings. The value of C_0 is necessarily equal to the ratio of the contaminant generation rate to the supply air volume rate. The value of C_0 may be changed according to the room type in which the ratio of the generation rate to the supply air volume rate may be changed. Thus, two kinds (Models 1 and 2) of dimensionless concentration are used. Model 1 is defined as the concentration non-dimensionalized by the individual C_0 of the each room type. In Model 2, the value of C_0 of a basic type is commonly used for non-dimensionalization. Both of Model 1 and Model 2 are important for the various analysis of diffusion field.

The values of SVE1 is generally calculated using the non-dimensional data of Model 1, given by the numerical simulation, since this scale expresses the individual properties of each diffusion field.

However Model 1 is not convenient for comparing the different diffusion fields because the representative value of C_0 is not common. When we want to compare two dimensionless concentration distribution fields, the value of C_0 must be held in common. If the representative concentration used for non-dimensionalizing the concentration is identical, the two dimensionless concentration fields can be compared directly. For this purpose, the value of C_0 for the basic type is often used as the common representative concentration. This dimensionless concentration is defined as Model 2.

7. CORRESPONDENCE BETWEEN EXPERIMENT AND SIMULATION

7.1 Flow Field

As is shown in Figure 5 (a) and (c), Figure 7 (a) and (c),and also Figure 15(a) and Figure 16(a), the results of the simulation of the flow field correspond well with those of the experiment. Figure 5(a) and (c) show a comparison of the simulated results with the experimental results for the distribution of the velocity vectors for Type 1 room. Figure 7 (a) and (c) show a comparison in the case of Type 2. Detailed comparisons are given in Murakami et al.(1988)

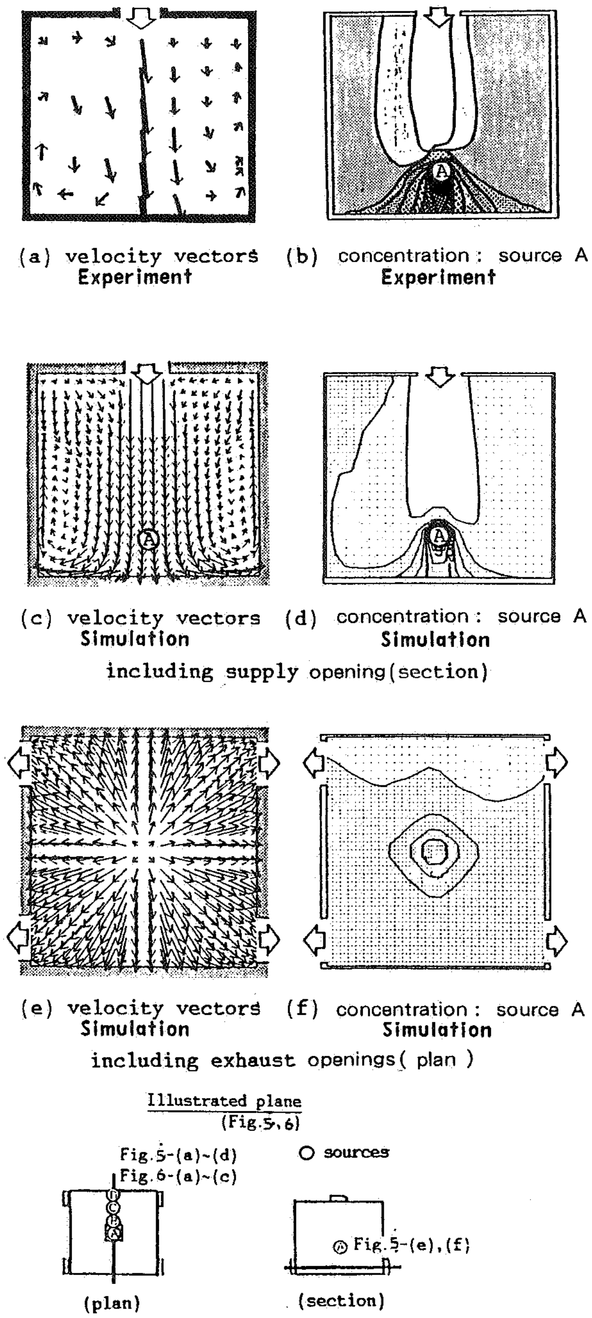
7.2 Diffusion Field

As is shown in Figure 5 (b) and (d), Figure 7 (b) and (d), and Figure 15 (b) and Figure 17 (a), the results of the simulation of the contaminant diffusion field, correspond well to those of the experiment. Although the contour lines of concentration are not exactly the same, the main characteristics of the contaminant diffusion are well reproduced, that is, the shape of the high concentration region, the low concentration region under the supply outlet, and so on. However, the result of the simulation tends to be more diffusive than those obtained by the experiment, thus the values of the contaminant concentration tends to be smaller than those given by the experiments for areas where the concentration is high and to be larger for areas where the concentration is low.

8 FLOW AND DIFFUSION FIELD FOR TYPE 1 (ONE SUPPLY OPENING, Figures 5,6)

8.1 In the Case of Contaminant Generated under Supply Opening

The flow field is shown in Figure 5 (c) and (e). The jet from the supply opening attacks the floor and diverges toward the wall. The diverged streams reach the sidewalls and turn up toward the ceiling. The distribution of concentration in the case where the contaminant is generated in the supply jet is shown in Figure 5 (d) and (f). The contaminant source point is marked as A. The concentration is very high in the area between the source and floor. However, the value of the concentration is rather uniform throughout the room and is more than 0.5 (non-dimensional value), except for the area just beneath the supply opening where it is very clean (Figure 5 (d)). The spatial average concentration, SVE1, is 0.9 and the mean radius of diffusion, SVE2, is 2.8 (non-dimensionalized by L_0), which is 29% of the relevant length of the room, 8.4. The relevant length of the room is defined as the square root of the sum of the square of each of the three dimensions of the room.



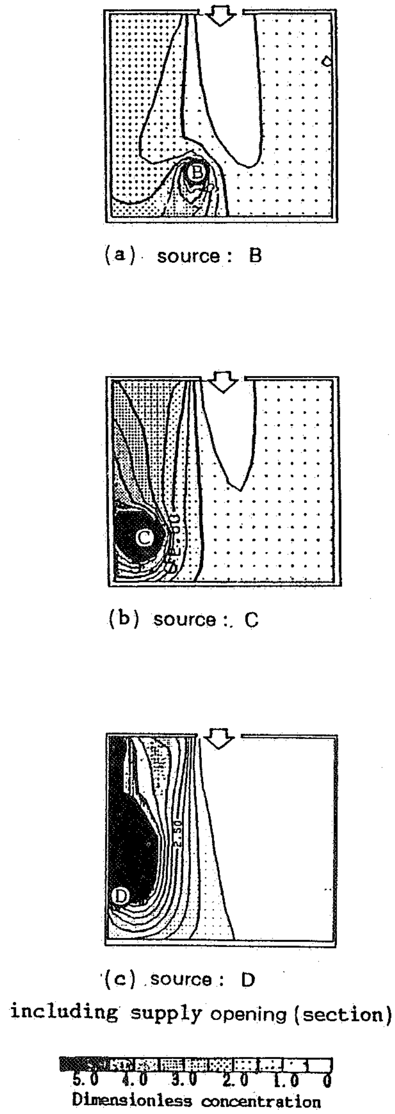


Figure 5. Velocity vectors and contaminant distribution in TYPE 1 room model (1 supply & 4 exhausts, source: point A)

Figure 6. Comparison of diffusion field for various source points in TYPE 1 (simulation, source: point B~D)

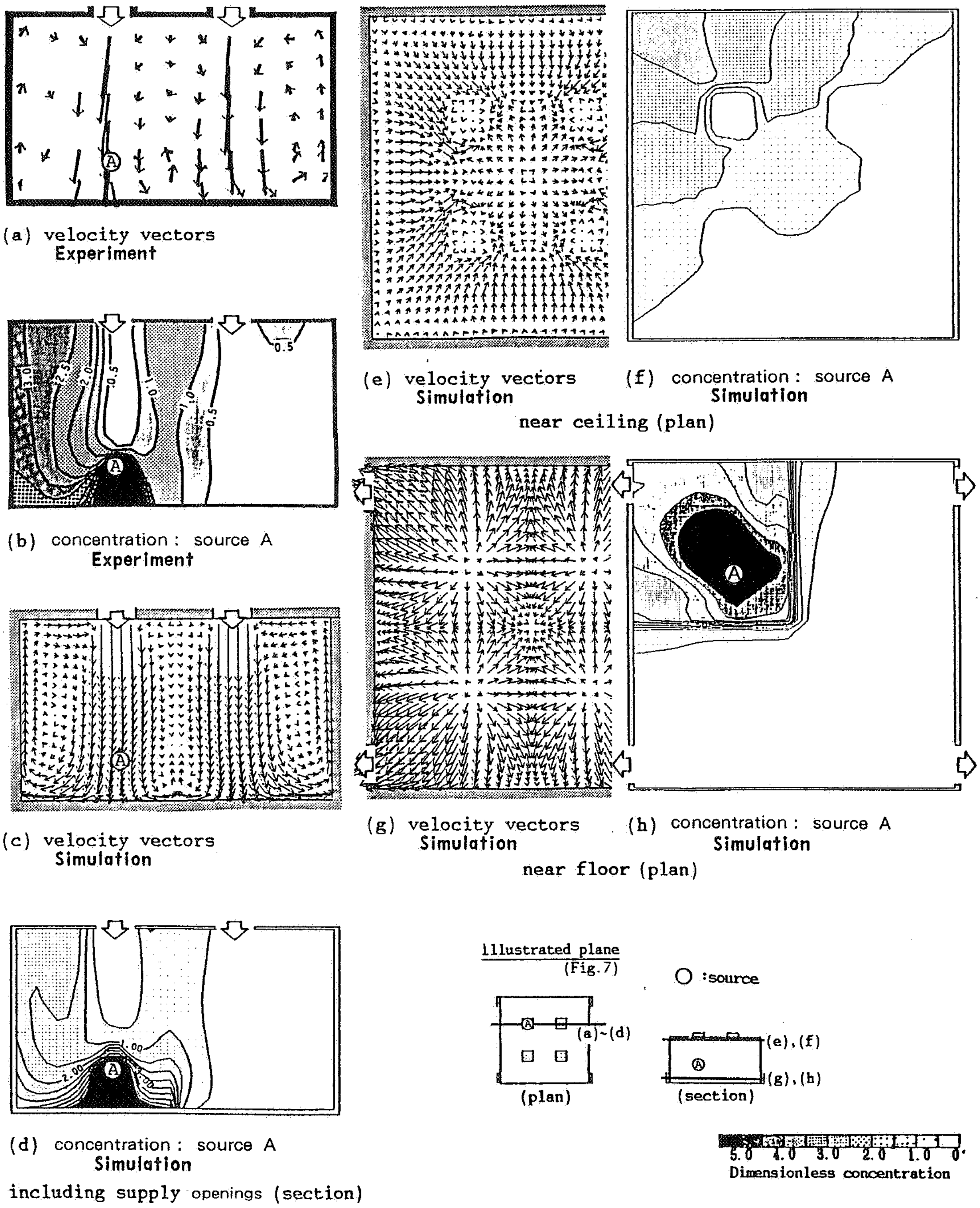


Figure 7. Velocity vectors and contaminant distribution in TYPE 2 room model (4 supplies & 4 exhausts, source: point A)

8.2 In the Case of Contaminant Generated between Supply Jet and Wall

Figure 6 shows the distributions of concentration in the case contaminant being generated between the supply jet and the wall at points B, C, and D, respectively. The generated contaminant is convected and diffused by the diverged flow near the floor and by the rising stream along the wall (Figure 6 (a), (b), and (c)). When the air velocity is relatively weak at the source of the contaminant, it diffuses in all directions (Figure 6 (b)).The spatial average concentrations, SVE1, are 1.0 (in the case of point B), 1.3 (in the case of point C), 1.6 (in the case of point D). These values become larger as the source points are located closer to the wall. The mean radii of diffusion, SVE2, are 2.4 (in the case of point B), 2.3 (in the case of point C), and 2.1 (in the case of point D). These values become smaller as the sources are located nearer to the wall.

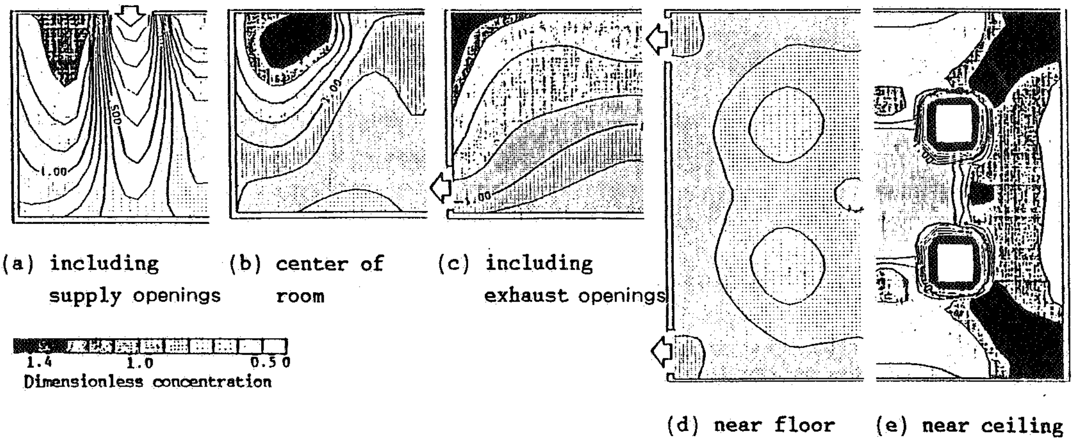


Figure 8. Contaminant distribution (SVE3) in TYPE 2 (source: uniform generation throughout the room)

9. FLOW AND DIFFUSION FIELD FOR TYPE 2 (FOUR SUPPLY OPENINGS, Figures 7,8)

9.1 Characteristics of Flow Field

The distributions of velocity vectors in the several sectional planes are shown in Figure 7. Many characteristics of the flow pattern of Type 1 often appear in Type 2. It may be reasonable modeling to regard the flow pattern of Type 2 as a combination of four flow patterns of Type 1. The flow pattern of Type 1, which is characterized by a vertical down jet from the supply opening and the rising streams around it, might be called a 'flow unit,' each of which occupies a quarter space of Type 2.

9.2 In the Case of Contaminant Generated under Supply Opening

The supply jet hits the floor and diverges in all directions. Rising streams are formed between the area of supply openings and the area near the side walls (Figure 7(c)). The contaminant that is generated in the supply jet spreads in accordance with this flow field. The concentration is the highest in the area from just below the source to the floor (Figure 5 (d)). The value of concentration is more than 0.5 only in the quarter part of the room that corresponds to the single 'flow unit' in which the contaminant is generated (Figure 7 (h)). In the remaining space of the room, concentration is very low (Figure 7 (d) and (h)). The spatial average concentration, SVE1, is 0.8 and is less than the value in the same case of Type 1. The mean radius of diffusion, SVE2, is 3, which is 25% of the relevant length of the room, 12.1, and is relatively less than the value in the same case for Type 1. These results are caused by the fact that the spreading area of the contaminant is confined to one 'flow unit'.

9.3 In the Case of Contaminant Generated Uniformly throughout Room, SVE3

Figure 8 shows the distribution of concentration in the case of the contaminant being generated uniformly throughout the room. The concentration takes its higher value around the supply openings, and at the corner of the ceiling, as shown in Figure 8. Thus, in terms of air mass movement, because the area around the supply opening is the farthest from the supply opening and the air mass takes the longest way to reach the area around the supply opening, the probability of the air around the supply opening being contaminated is the highest.

10. FLOW AND DIFFUSION FIELD FOR TYPE 4 (NINE SUPPLY OPENINGS, Figure 9,10)

10.1 In the Case of Contaminant Generated as Point Source

The flow field of Type 4 is shown in Figure 9-1(a),(b). As with Type 2 it is logical to regard the pattern of Type 4 as a serial combination of 'flow unit', in this case nine units. When the contaminant is generated in the flow unit that faces the exhaust opening (source point F, Figure 9-1(c) and (d)), the contaminant hardly diffuses into the other flow units, although the concentration is very high in that single flow unit. The spatial average concentration, SVE1, in this case is only 0.3 and the mean radius of diffusion, SVE2, is 2.3, 14% of the relevant room length of 16, showing very small value.

When the contaminant is generated in the center flow unit adjacent to the wall, (source point B, Figure 9-1(e) and (f)), the contaminant spreads, not only within that center flow unit, but also into the adjacent flow units that are located on the way to the exhaust opening. That one-third of the room is contaminated, but the remaining two-thirds of the room is very clean. The spatial average concentration, SVE1, is 1.2, and the mean radius of diffusion, SVE2, is 3.3. The latter value is considerably greater than that of the contaminant being generated at point F.

When the contaminant is generated in the center of the room at point E, all of the space is contaminated. Because this flow unit in which the contaminant is generated does not face the exhaust opening but is adjacent to all the other flow units, the contaminant is convected by the flow toward the exhaust through all the other flow units. The spatial average

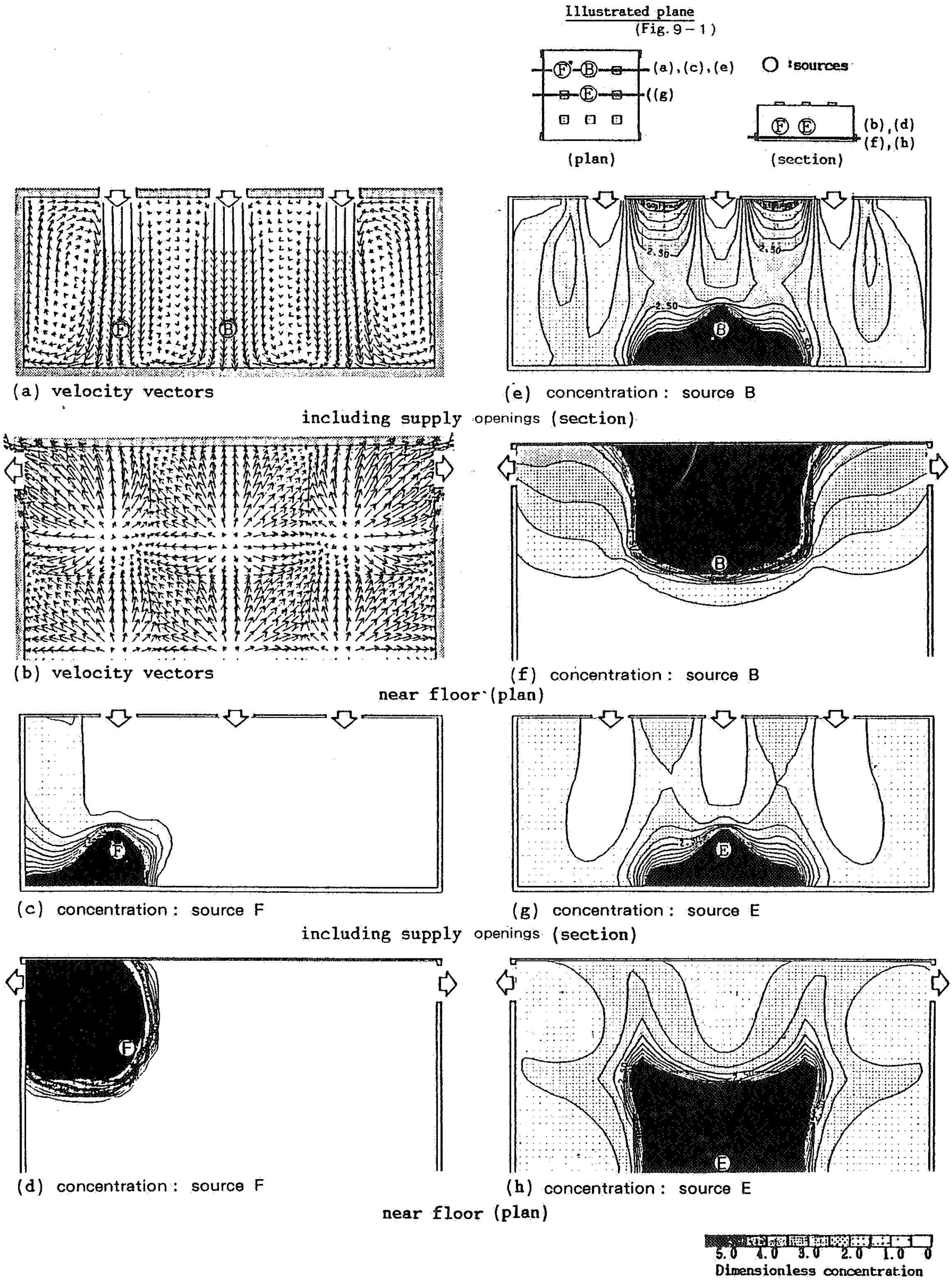


Figure 9-1. Velocity vectors and contaminant distribution in TYPE 4 room model
(9 supplies & 4 exhausts, source: point B,E,F)

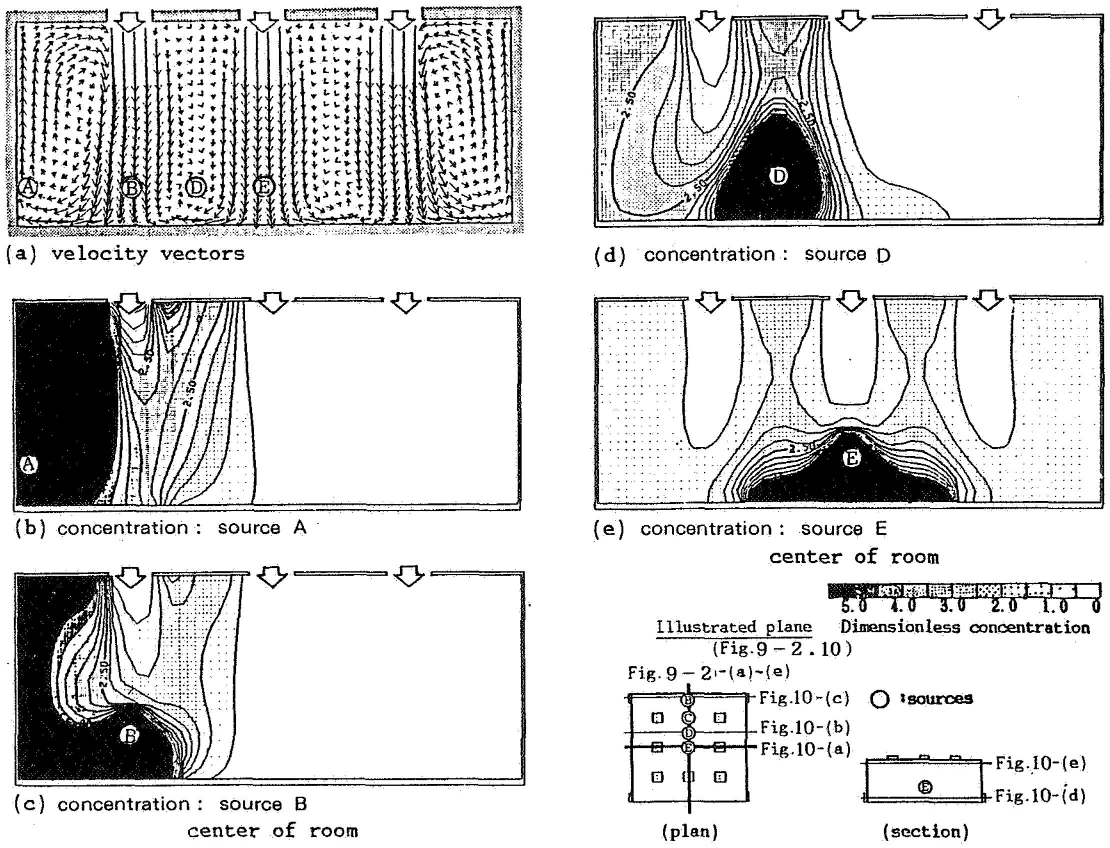


Figure 9-2. Comparison of contaminant distribution for various source points in TYPE 4 (source: point A~E)

concentration, SVE1, is 1.4, and the mean radius of diffusion, SVE2, is 4.3, 26% of the relevant length of the room.

Figure 9-2 shows the diffusion field when the contaminant is generated at point A, B, D, and E. These source points move from the area neighboring the wall to the center of the room. The spatial average concentrations, SVE1, are 1.7, 1.3, 1.4, and 1.4, respectively. The mean radii of diffusion, SVE2, are 3.1, 3.3, 3.6, and 4.3. Therefore, SVE2 becomes greater as the contaminant source is placed farther from the wall.

10.2 In the Case of Contaminant Generated Uniformly throughout Room, SVE3

Figure 10 shows the distribution of concentration of SVE3. The major characteristics of the concentration distribution pattern are almost the same as in the cases of Type 2. The highest value is observed near the ceiling around the supply opening and at the corners of the ceiling.

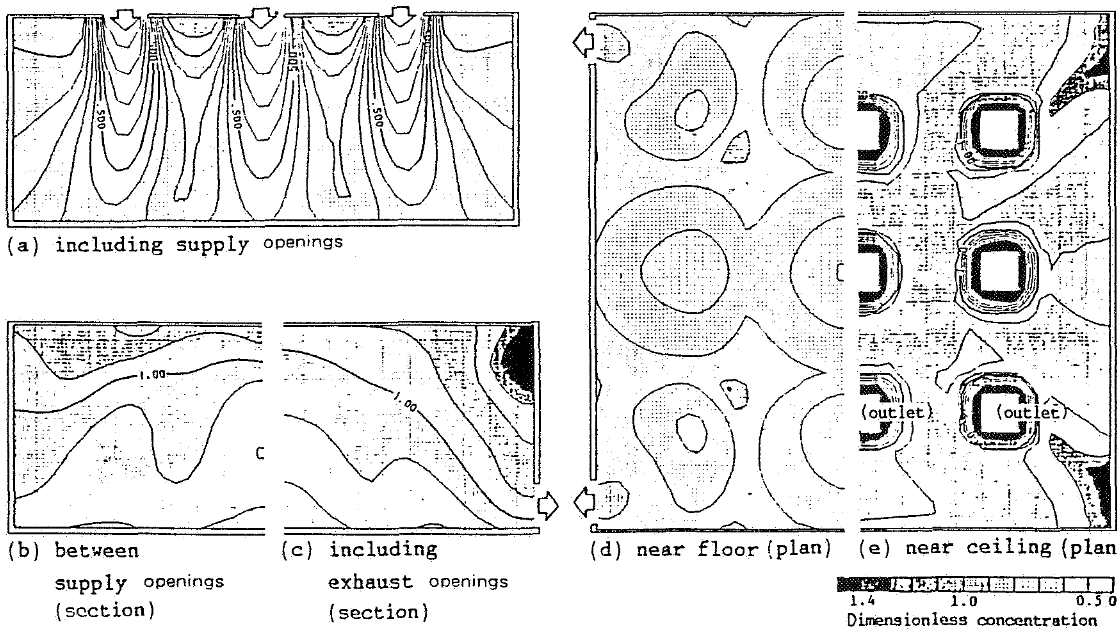


Figure 10. Contaminant distribution (SVE3) in TYPE 4 (source: uniform generation throughout the room)

11. CONCEPT OF 'FLOW UNIT'

From the results of the simulations, it may be concluded that the mean flow structure in the room that has supply openings in the ceiling is composed of series of flow units that consist of one supply jet and the rising streams around it. Such a flow unit is useful in comprehending the complicated flow pattern in rooms in which the supply openings are set in the ceiling. Furthermore, this concept of a flow unit seems to be helpful in comprehending the contaminant diffusion in rooms. It is a well-known fact that the exhaust flow has small influence on the whole flow pattern. Therefore, it is not defective that this model of a flow unit does not include the function of the exhaust opening.

When the contaminant is generated in a flow unit, the contaminant diffusion at its first stage is confined within that unit. If the flow unit faces the exhaust opening, the contaminant is not convected to the other flow units and only a small amount of the contaminant spreads to them by turbulent diffusion. If the contaminant is generated in the flow unit that does not face any exhaust openings, the contaminant is convected to the flow units that face the exhaust opening and the remaining flow units are not contaminated. Even if they do become contaminated, it is only to a small degree, because such contamination is caused solely by turbulent diffusion, which has much less ability to transport the contaminant than does mean flow convection. The turbulent Reynolds number (Peclet number), $U_0 L_0/\nu$, in these cases is on the order of 100, which means in general that the ability to transport the contaminant by convection is a hundred times greater than that of turbulence diffusion.

When the contaminant is generated at the boundary of two flow units, where strong rising streams are usually formed, the contaminant diffuses into both and passes through the other flow units that are located on the path of the flow to the exhaust opening.

The qualitative characteristics of the structure of the diffusion field described above are quantitatively assessed very well by means of the new scales of ventilation efficiency SVE1,2 and 3.

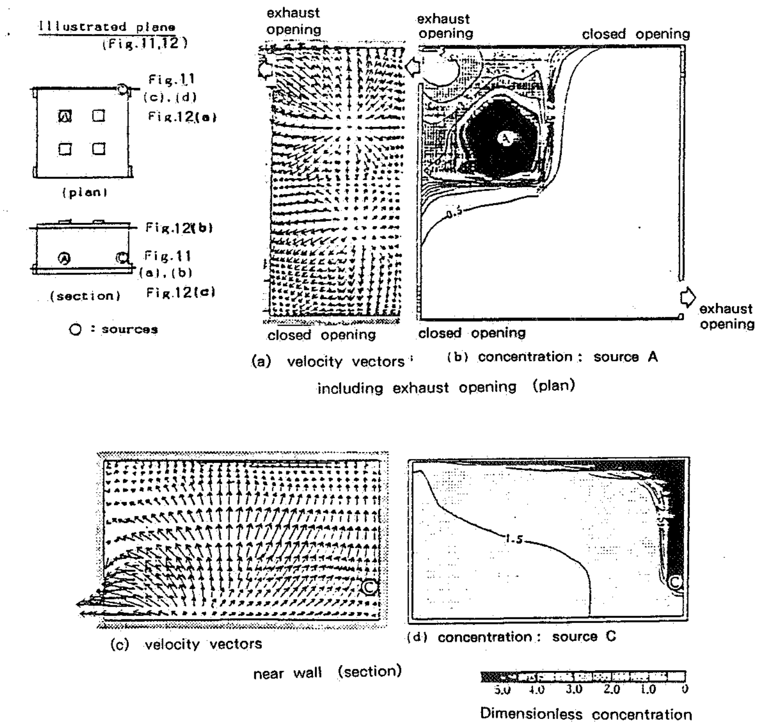


Figure 11. Velocity vectors and contaminant distribution in TYPE 3 (4 supplies & 2 exhausts)

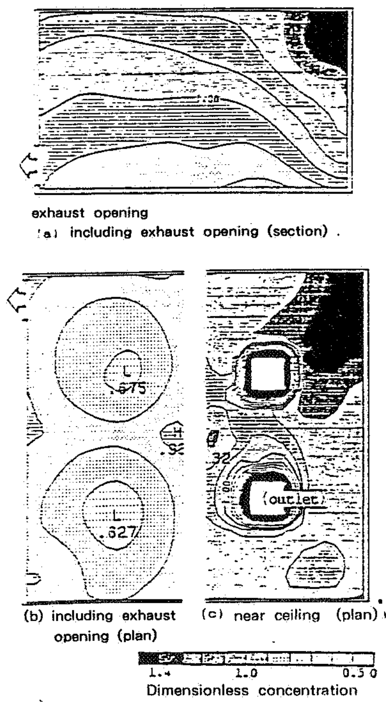


Figure 12. Contaminant distribution (SVE3) in TYPE 3 (4 supplies & 2 exhausts, source: uniform generation throughout the room)

12. <u>INFLUENCE OF ARRANGEMENT OF EXHAUST OPENINGS (TYPE 2,3, Figures 7,11,12)</u>

Type 3, in which two exhaust openings are located diagonally, is a room model derived from Type 2 and is regarded as a case of decreasing exhaust openings. In this room model, only two exhaust openings are located at diagonal corners (the other two exhaust openings are eliminated). As shown in Figure 11(a), there are four 'flow units' in this type as well as in Type 2 (Figure 7). In the case of contaminant generation at point A, which is in the 'flow units' adjacent to the exhaust opening and is under the supply opening, although the contaminated space is nearly the same as with Type 2 (cf.Figure 7(h) and 11(b)) and the value of the mean radius of diffusion, SVE2, is the same, the value of the spatial average concentration, SVE1, for Type 3 is smaller than that for Type 2, which means that the contaminant is exhausted effectively by the stronger flow toward the exhaust opening.

At the corner of the eliminated exhaust openings, strong rising streams along the wall appear. When the contaminant is generated in this position (point C), the contaminant spreads upward along the wall and the large area along the ceiling becomes highly contaminated (cf. Figure 11(c) and 11(d)). In this case, the values of the spatial average concentration, SVE1, and the mean radius of diffusion, SVE2, are higher than for all other cases in this room model (1.6 and 3.4 respectively). The distribution of the concentration in the case of uniform contaminant generation throughout the room is shown in Figure 12. At the corner near the ceiling of the upper position of the closed exhaust openings, the concentration becomes very high.

13. INFLUENCE OF SYSTEMATIC CHANGE OF ARRANGEMENT OF SUPPLY OPENING (TYPE 4,5,6,7,8, Figures 13,14)

In this section, flow and contaminant diffusion fields in the same room with different arrangements of supply openings (Type 4,5,6,7,8) are compared. These supply arrangements, in which the numbers are progressively decreased, are modeled on the basis that the air exchange rate is decreased according to the elimination of supply openings rather than by decreasing the supply air volume of each opening.

Figure 13 shows the flow field and contaminant diffusion field in the case where the contaminant is generated at the center of the room (Point E). The outline of the structure of the flow units is illustrated in each superimposed figure using broken circles.

13.1 Flow fields and Contaminant Diffusion Fields in Case of Contaminant Generated at point E

(1) Type 4 (Figure 13(a)-(c)).

There are nine flow units in the room model, and rising streams appear at the boundary of each flow unit. The rising streams in the space between the two closest jets do not reach the ceiling. Since the contaminant is generated in a supply jet, the highly contaminated region spreads under source point E. The flow unit that includes the contaminant source is highly contaminated.

(2) Type 5 (Figure 13(d)-(f)).

Six flow units comprise the total flow field. At the centerline of the room, where three supply openings are closed, strong rising streams appear toward the ceiling. This centerline corresponds to the boundary of the expanded flow units. Since the contaminant is generated in this rising stream, the highly contaminated region spreads upward from source point E. The whole room is filled with highly contaminated air.

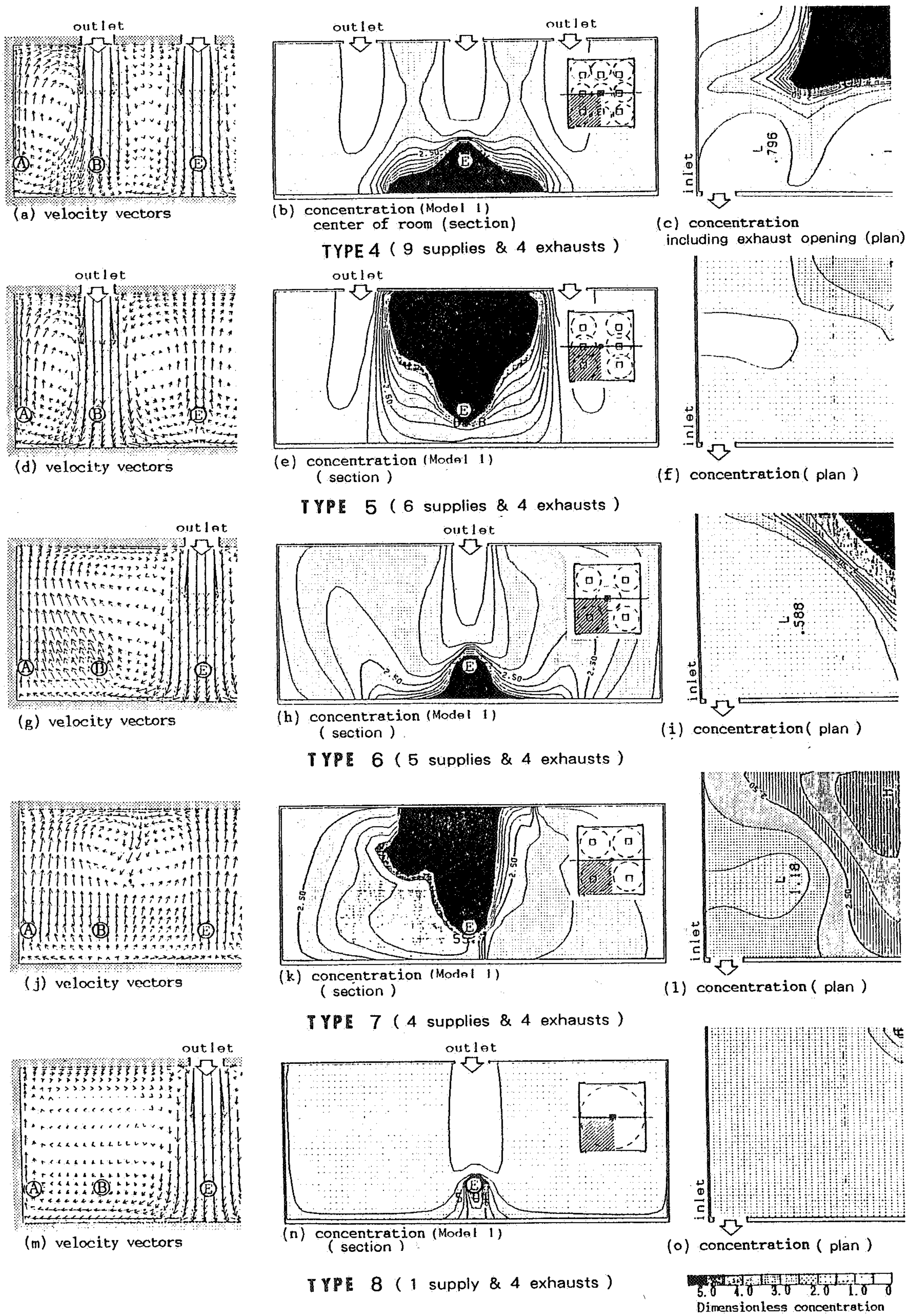


Figure 13. Effect of decreasing supply openings on diffusion field
(TYPE 4-8, source: point E)

(3) Type 6 (Figure 13(g)-(i)).

The five checkered flow units comprise the total flow field. The rising streams surrounding the center flow unit spread toward the upper position of the walls. Since the contaminants are generated in the supply jet in the center flow unit, the highly contaminated region appears under source point E. This contaminated air is transported by the rising flow toward the upper position of the walls and most of the space becomes contaminated. The concentration becomes more than 1.0 in most of the space.

(4) Type 7 (Figure 13(j)-(l)).

Four large flow units comprise the total flow field. At the center of the room, a narrow rising stream appears toward the ceiling. Since the contaminant is generated in the rising stream, the highly contaminated region spreads above source point E, and most of the room is filled with contaminated air. In this simulation, the contaminant diffusion field is slightly asymmetric because of the asymmetry of the flow field due to numerical instability.

(5) Type 8 (Figure 13(m)-(o)).

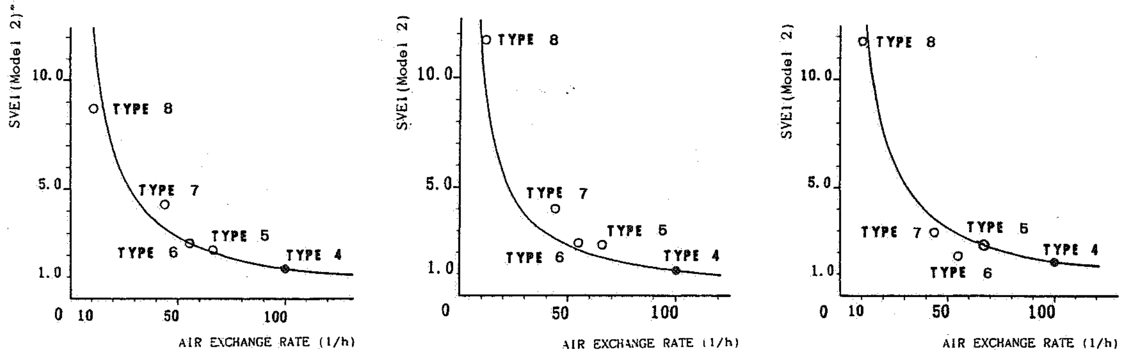
There is only one flow unit in the room model. Since the contaminant is generated in the supply jet, the highly contaminated region spreads under source point E. The concentration is more than 1.5 throughout the whole room except for the area around the clean supply jet.

13.2 Comparison of Location of Supply Openings Concerning Ventilation Effectiveness (Source Point E)

In Table 4 two kinds of averaged spatial concentration (Model 1 and 2) and the mean radius of diffusion are tabulated for each type. Model 1 is a dimensionless concentration which is normalized by the mean concentration, Co, for each type. Model 2 is also a dimensionless concentration which is normalized by the mean concentration of Type 4 commonly.

The supply air velocity is the same for all types. Therefore, the air exchange rate is naturally different for each type. In Model 1, the representative concentration (C_o) for non-dimensionalization is not the same. But in Model 2, the representative concentration (C_o) of Type 4 is used in common for making the dimensionless value.

Figure 14(a) shows the spatial average concentration of each type for source point E. In this figure, the dimensionless concentration of Model 2 is shown. The hyperbolic curve expresses the dimensionless average spatial concentration of Type 4 in which the air exchange rate is gradually decreased under the condition of a constant generation rate. We can thus comprehend the ventilation effectiveness of the different arrangements of the supply openings. Using this figure, if the plotted point of average spatial concentration for a type is below the hyperbolic curve, the ventilation effectiveness of that type is superior to that of Type 4 under



(a) Spatial averaged concentration with respect to air exchange rate for source point E

(b) for source point B

(c) for source point A

*1 SVE1 : spatial averaged concentration.
*2 Contaminant generation rate for each type is same as in TYPE 4.

Figure 14. Comparison of ventilation efficiency based on SVE1 by decreasing supply openings

Table4. Values of spatial averaged concentration (SVE 1) and Mean radius of Diffusion (SVE 2) for TYPE 4~8

sources	point E (center of room)			point B (between center and wall)			point A (near the wall)		
Scales of Ventilation Efficeincy	SVE 1 (Model 1)	SVE 1 (Model 2)*1	SVE 2 (*2)	SVE 1 (Model 1)	SVE 1 (Model 2)	SVE 2	SVE 1 (Model 1)	SVE 1 (Model 2)	SVE 2
TYPE 4	1.4	1.4	4.2	1.3	1.3	3.3	1.7	1.7	3.1
TYPE 5	1.5	2.3	4.0	1.6	2.3	4.1	1.7	2.5	4.0
TYPE 6	1.4	2.5	4.3	1.4	2.5	3.8	1.0	1.9	4.0
TYPE 7	1.9	4.3	3.8	1.8	4.1	3.8	1.3	2.9	4.0
TYPE 8	1.0	8.7	3.9	1.3	11.8	4.3	1.3	11.6	4.3

*1 : Model 2 is dimensionless concentration in which the mean concentration of TYPE4 is commonly used as the representations value for non-dimensionalization.

*2 : Dimensionless length : these values are made dimensionless by dividing by the width of the supply opening (0.6m).

the same air exchange rate. This corresponds to the comparison based on Model 1, since the comparison based on Model 1 assumes the same air exchange rate and the same contaminant generation rate.

As shown in Figure 14(a), for the contaminant source point E, ventilation effectiveness among these different supply opening arrangements may be judged in the following order: Type 8 > Type 4 = Type 6 = Type 5≫ Type 7 (cf. comparison of SVE1 based on Model 1 in Table 4).

For the mean radius of diffusion Type 4 and Type 6 have rather high values (cf. Table 4). The contaminant source point in all these cases is located in the supply jet.

13.3 Comparison of Location of Supply Openings Concerning Ventilation Effectiveness
(Source Point B)

For the case of source point B (cf. Figure 13), two kinds of average spatial concentration (Models 1 and 2) and the mean radius of diffusion are tabulated for each type in Table 4 and comparisons of ventilation effectiveness are shown in Figure 14 (b) in the same manner as before.

Since every plotted point of the average spatial concentration in Type 5 - Type 8 room model is above the hyperbolic curve, it may be concluded that the arrangement of the supply openings in Type 5 - Type 8 for this contaminant source is inferior to that of Type 4 under the same air exchange rate. For contaminant source point B, ventilation effectiveness among these different cases of arrangement of supply openings is estimated in the following order: Type 4 = Type 8 > Type 6 > Type 5 > Type 7.

For the values of the mean radius of diffusion, there seems to be small difference among them.

13.4 Comparison of Location of Supply Openings Concerning Ventilation Effectiveness
(Source Point A)

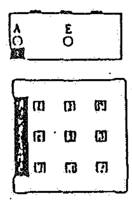
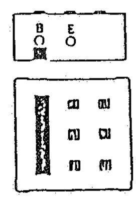
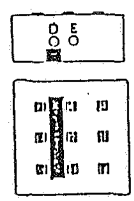
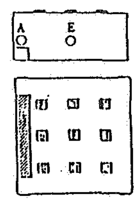
For the case of source point A (cf. Figure 13), two kinds of average spatial concentration and the mean radius of diffusion are tabulated for each type in Table 4; comparisons of ventilation effectiveness are shown in Figure 14 (c) as before.

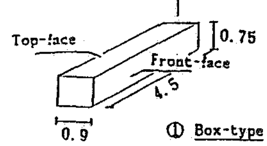
Since the plotted points of the averaged spatial concentration of Type 6 and Type 7 are below the hyperbolic curve, it may be concluded that for this contaminant source, the arrangements of the supply openings in Type 6 and Type 7 are superior to that of Type 4 under the same air exchange rate from the viewpoint of ventilation effectiveness. Note especially that Type 6, which has only five-ninths the air exchange rate of Type 4 , has almost equal ventilation effectiveness with Type 4. Thus, for contaminant source point A, ventilation effectiveness among these different arrangements of supply openings is estimated in the following order: Type 6 $\gg$ Type 7 = Type 8 > Type 4 = Type 5.

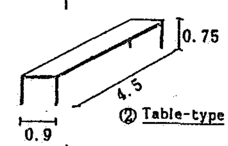
The values of the mean radius of diffusion for these room models, except for Type 4, are close to 4.0 and thus larger than Type 4 (3.1: cf. Table 4)

The room model used here is Type 4 which has 9 supply openings and 4 exhaust openings. Table 5 lists the four cases analyzed here and illustrates the various arrangements of flow obstacles and various source positions of contaminant.

Table 5. Specifications of Cases of Air Flow with Obstacles

Cases	Case 1	Case 2	Case 3	Case 4
Flow Obstacles	One Box-type Obstacle	One Box-type Obstacle	One Box-type Obstacle	One Table-type Obstacle
Arrangement of Flow Obstacles and Position of Contaminant Generation (shown by the circle in the section)	 (in contact with the wall)	 (under suply jets)	 (between supply jets)	 (in contact with the wall)
Position of Contaminant Generation	A:adjacent to the wall (on the top-face of obstacle) E:center of the room S:SVE3	B:under supply jet E:center of the room S:SVE3	D:between supply jets E:center of the room S:SVE3	A:adjacent to the wall E:center of the room S:SVE3





remarks 1 Numerical simulations were conducted for all cases.
2 Model experiment was conducted only for case 1.

14.1 Arranging a Box-Type Obstacle in Contact with the Side Wall (Case 1, Figures 9,10,15,16,17)

The velocity field and the contaminant diffusion field with a flow obstacle in contact with the side wall as given by numerical simulation are shown in Figures 16 and 17. The results of the experiment for this case are shown in Figure 15. As stated before, the correspondence between the simulation and the experiment is rather well. The standard case with no flow obstacle is shown in Figure 9. The designated names for each face of the obstacle used here are also illustrated in Table 5.

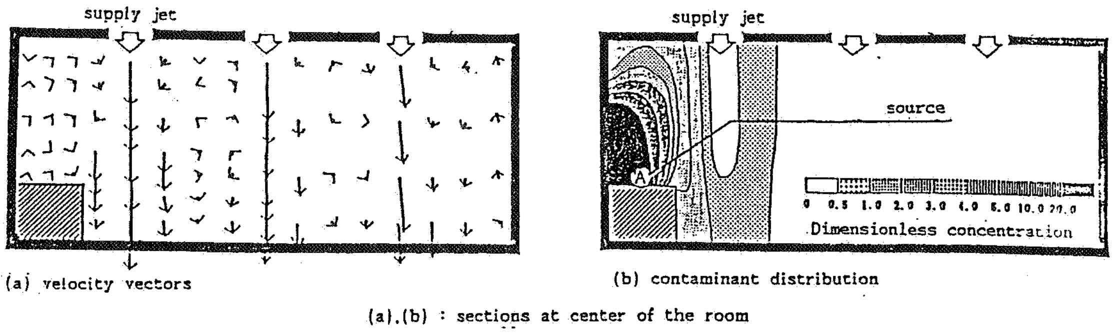


Figure 15. Velocity vectors and contaminant distribution in Case 1 (TYPE4) given by model experiment (Case 1: A box-type obstacle is placed in contact with the wall)

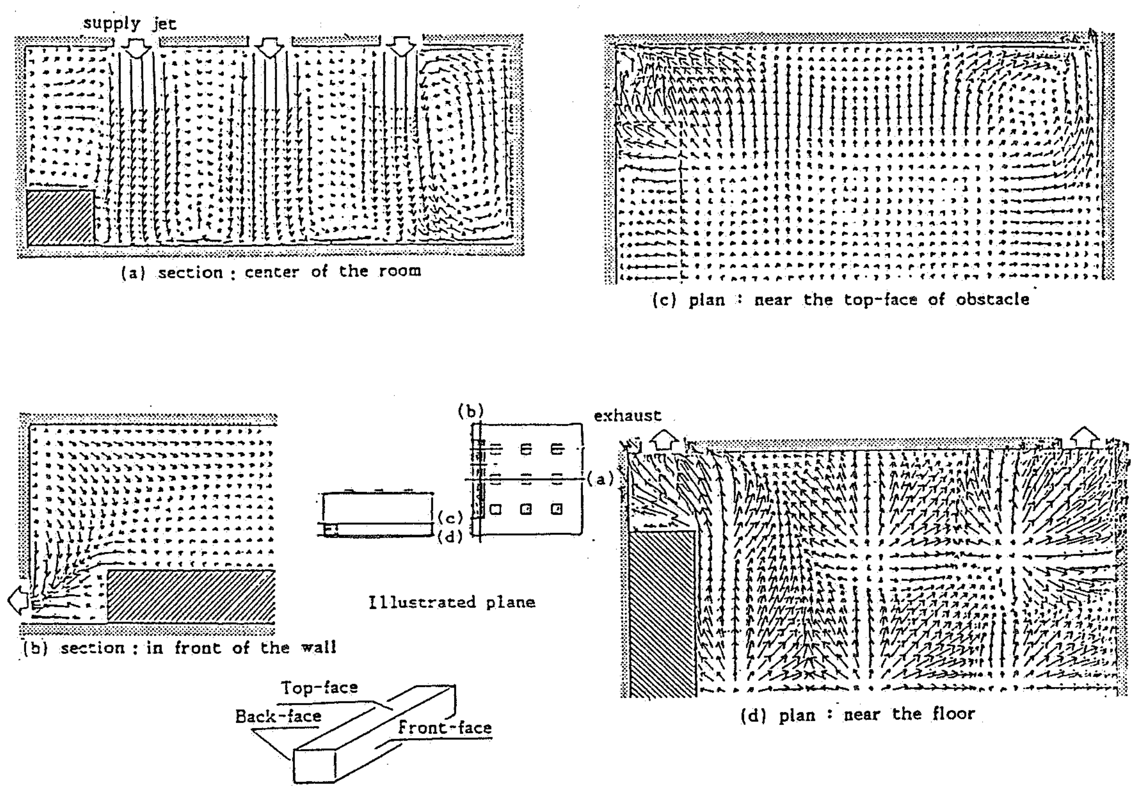


Figure 16. Velocity vectors in Case 1 (TYPE4) given by numerical simulation

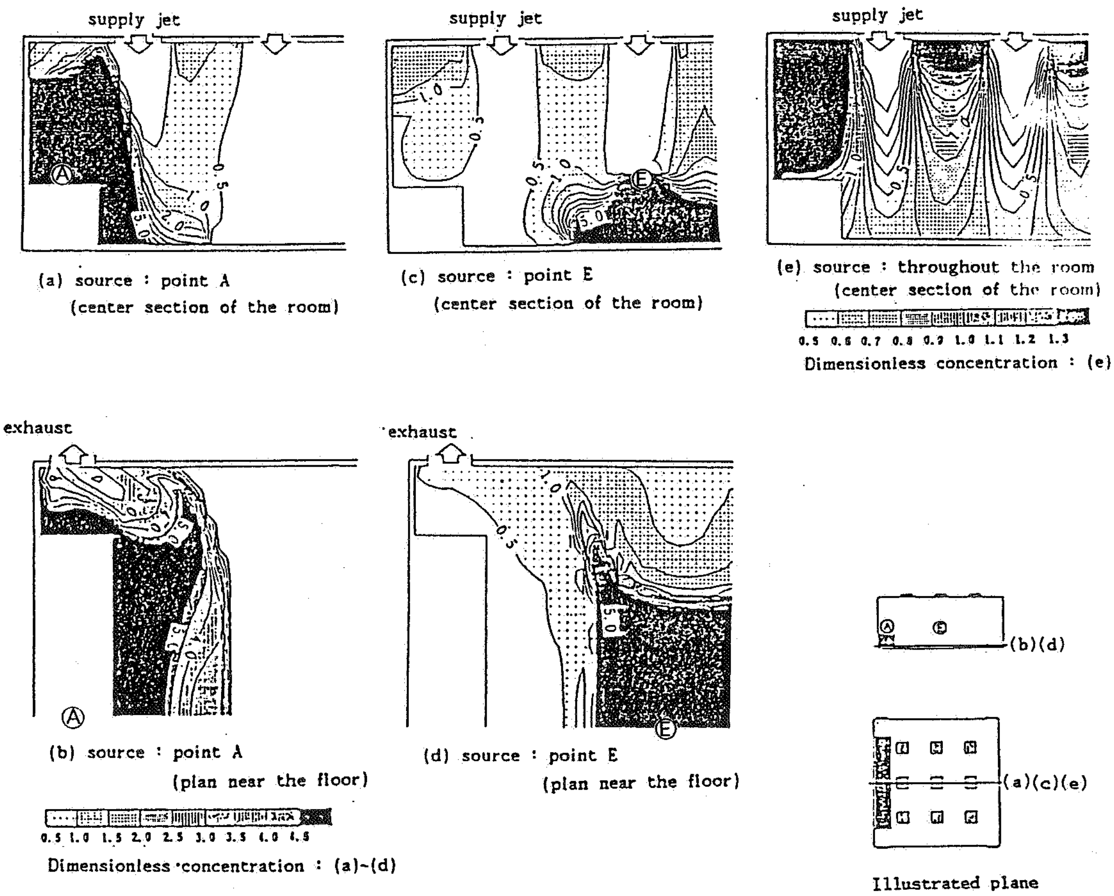


Figure 17. Contaminant distributions in Case 1 given by numerical simulation

(1) Velocity Field.

The flow pattern in front of the side wall is illustrated in Figure 16(b). As shown in Figure 16(a), a recirculating flow appears above the obstacle and some flows into the supply jet near the ceiling. The air above the top-face of the obstacle moves towards the side wall as shown in Figures 16(a),(c). In front of the obstacle, the air moves towards the exhaust opening along the front-face of the obstacle as shown in Figure 16(d).

The airflow pattern near the side wall with no obstacle is also shown in Figure 9-1. It differs greatly from that of Case 1. In the open area on the right side far from the obstacle, there is little difference between the two. The conspicuous effect of the flow obstacle is confined within the space around the obstacle, namely within the 'flow units' in which the obstacle exists.

(2) Contaminant Concentration Field.

When the contaminant is generated on the top-face of the obstacle (Point A, Figures 17(a),(b)), the contaminant diffuses into the left third of the room. This one-third diffusion pattern is similar to the result without obstacle. The concentration near the ceiling becomes rather low since the clean air is convected along the ceiling from the corner area. The value of SVE1 is 2.1 which is much larger than its value in the case without obstacle (1.7). The value of SVE2 is 2.5, smaller than in the case without obstacle (3.1). The values of SVE1 and 2 for all cases are tabulated in Table 6.

When the contaminant is generated in the center of the room (Point E, Figures 17 (c),(d)), it spreads throughout the whole room. But the space around the obstacle is not contaminated because air from the three supply jets near the side wall flows into this area. The value of SVE1 is 1.6 and hence larger than in the case without obstacle (1.4). Thus the ventilation efficiency for exhausting the contaminant decreased to some degree in Case 1. The value of SVE2 in Case 1 is 4.3, almost the same as the case without obstacle (4.2). Although the obstacle beside the wall has small effect on the velocity field around point E, the diffusion field for contaminant generation at Point E is influenced greatly whether the flow obstacle is present or not.

The value of SVE3 is compared in Figure 17(e) and Figure 10(a). The concentration above the obstacle in Case 1 is much higher than in the case without obstacle, thereby indicating that supplied clean air requires a long travelling time to reach this recirculating area around the obstacle.

14.2 Arranging a Box-type Obstacle Under Supply Jets (Case 2, Figures 18,19)

The velocity and diffusion fields when a flow obstacle is placed under the supply jets are shown in Figures 18 and 19.

(1) Velocity Field.

The supply jet attacks the top-face of the obstacle and diverges in all directions (Figures 18(a),(c)). A small rising stream appears above the top-face between the supply jets (Figure 18(b)). Recirculating flows exist in front of the back and front-faces of the obstacle(Figure 18(a)). In the open area on the right, the velocity field of Case 2 is the same as that of the case without obstacle, hence the significant effect of this arrangement of the flow obstacle is confined within a rather small area near the obstacle.

(2) Contaminant Diffusion Field.

When the contaminant is generated at the top-face of the obstacle (Point B), it is convected horizontally by the diverging flow at this area (Figure 19(a)). The high concentration spreads into the recirculating region along the side wall and also into the area in front of the back and front-faces (Figure 19a)). The contaminated area occupies the left half of the room (Figures 19(a),(b)). The value of SVE1 in Case 2 is 1.9 and much larger than in the case without obstacle (1.3). The value of SVE2 in Case 2 is 3.7 and also larger than in the case without obstacle (3.2).

When the contaminant is generated at the center of the room (Point E), it spreads into the open area on the right where no obstacle is arranged (Figures 10(c),(d)). The space to the left of the obstacle is clean since

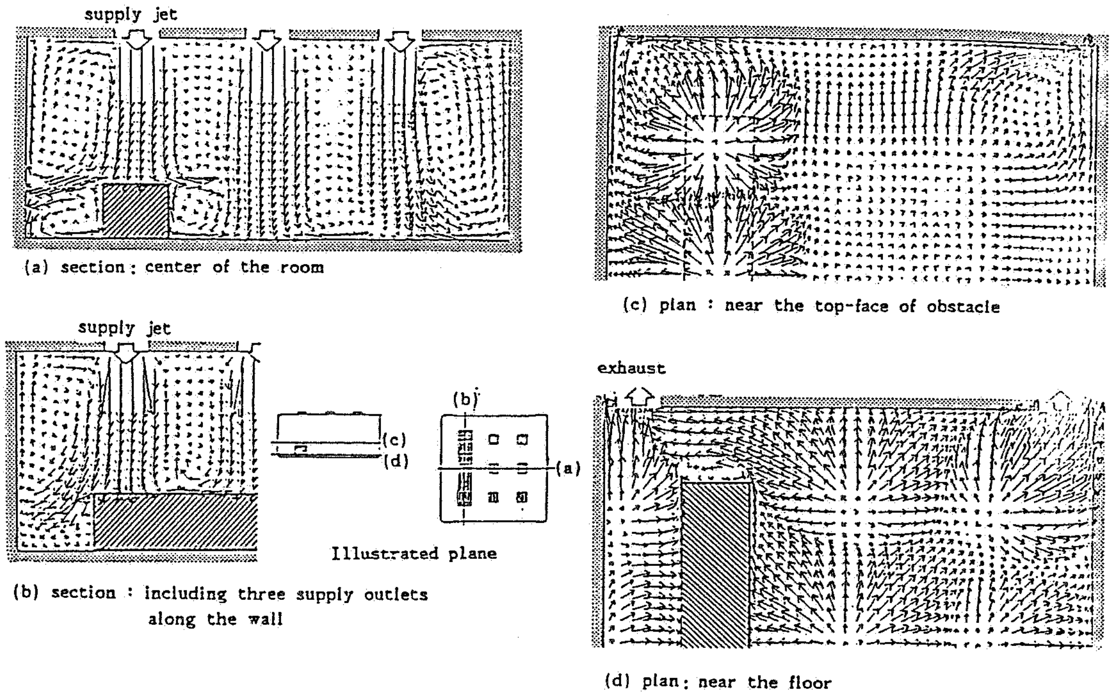


Figure 18. Velocity vectors in Case 2

(Case 2: A box-type obstacle is placed under supply jets)

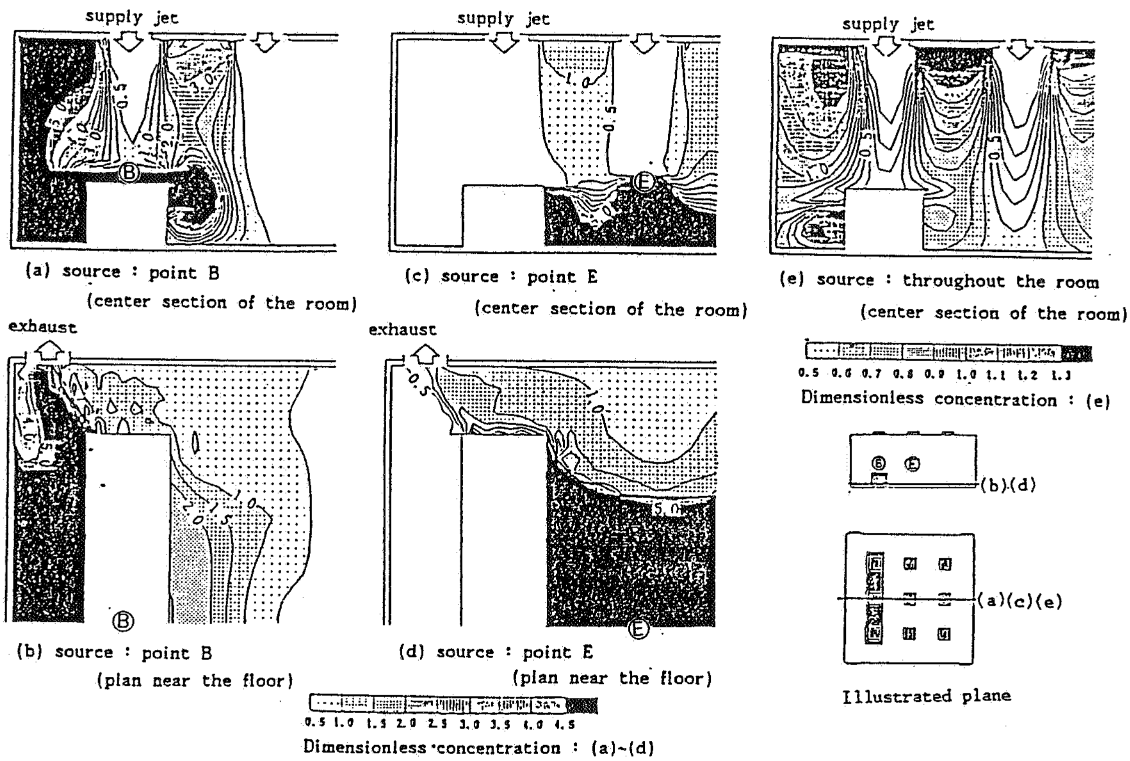


Figure 19. Contaminant distributions in Case 2 with various source position

the spread of the contaminant is blocked by the obstacle. The value of SVE1 is 1.6 and larger than in the case without obstacle (1.4). The value of SVE2 is 4.2, which is the same as in the case without obstacle (4.2).

The value of SVE3 is very low above the top-face of the obstacle because of the direct supply of clean air (Figure 19(e)).

14.3 Arranging a Box-Type Obstacle Between Supply Jets (Case 3, figures 20,21)

The flow and diffusion fields for Case 3 are illustrated in Figures 20 and 21, where a box-type obstacle is arranged between the supply jets.

(1) Velocity Field.

The velocity field at the top-face of the obstacle is horizontal and flows mainly towards the exhaust opening, as shown in Figures 20(a),(c). Rising streams appear at some points above the obstacle (Figure 20(b)). The supply jets at the center attack the floor and thus diverge towards the open area on the right because of blocking on the left side by the obstacle. The flow pattern in the open area on the right side is similar to that in the case without obstacle.

(2) Contaminant Diffusion Field.

When the contaminant is generated on the top-face of the obstacle (Point D, Figure 21(a)), it stays around the obstacle since the diffusion field is blocked by the rows of the supply jets on both sides of the obstacle (Figure 21(a)). The contaminated area is the left half of the room(Figures 21(a),(b)). The value of SVE1 is 1.7, which is larger than in the case without obstacle (1.5). The value of SVE2, on the other hand, is 3.2 and significantly smaller than in the case without obstacle (3.6).

When the contaminant is generated at the center of the room (Point E), it diffuses into the right half of the room, since diffusion towards the left is blocked by the obstacle (Figures 21(c),(d)). The top-face of the obstacle is clean since the supply jet attacks it. The value of SVE1 is 1.5 and a little larger than in the case without obstacle (1.4). The value of SVE2 (4.0) on the other hand is smaller than in the case without obstacle (4.2).

The distribution of SVE3 (Figure 21(e)) is similar to that in the Case without obstacle.

14.4 Arranging a Table-Type Obstacle (Case 4, Figures 22,23)

The flow and diffusion fields for Case 4 where a table-type obstacle is placed in contact with the side wall are illustrated in Figures 22 and 23.

(1) Velocity field.

A large recirculating flow appears above the obstacle (Figure 22(a)). The airflow pattern on the top-face is shown in Figure 22(c). The air under the top-face moves along the side wall and towards the exhaust opening (Figures 22(b),(d)). The flow pattern in the open area on the right side is the same as in the case without obstacle (Figure 22(a)). Thus the area affected by

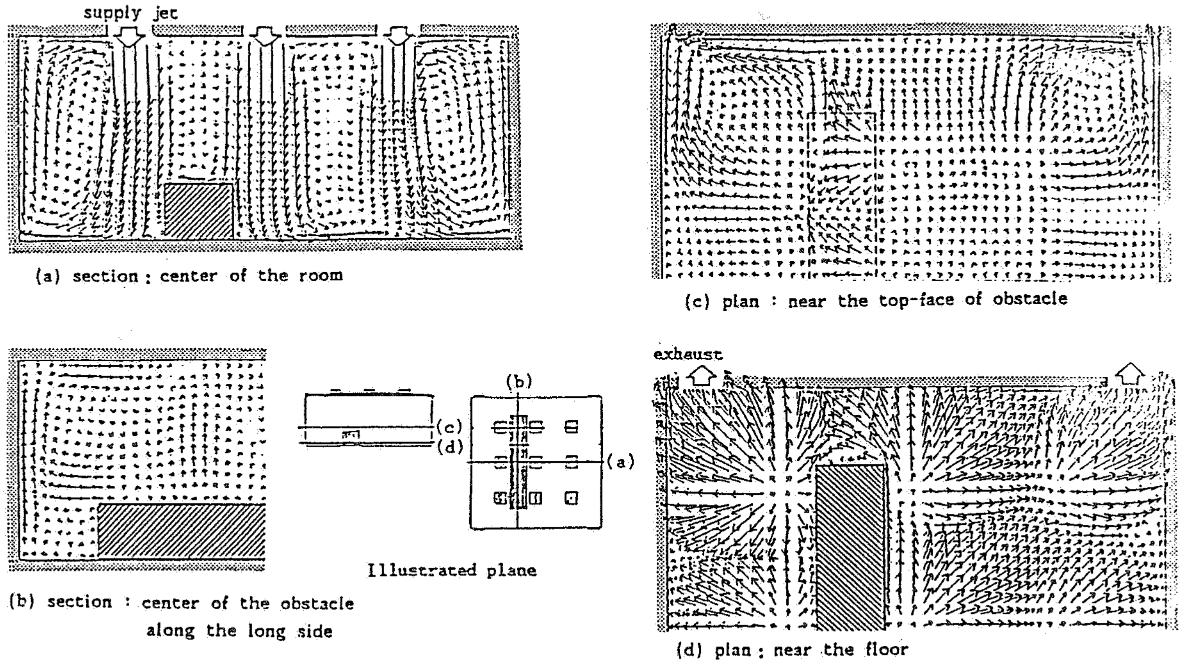


Figure 20. Velocity vectors in Case 3

(Case 3: A box-type obstacle is placed between supply jets)

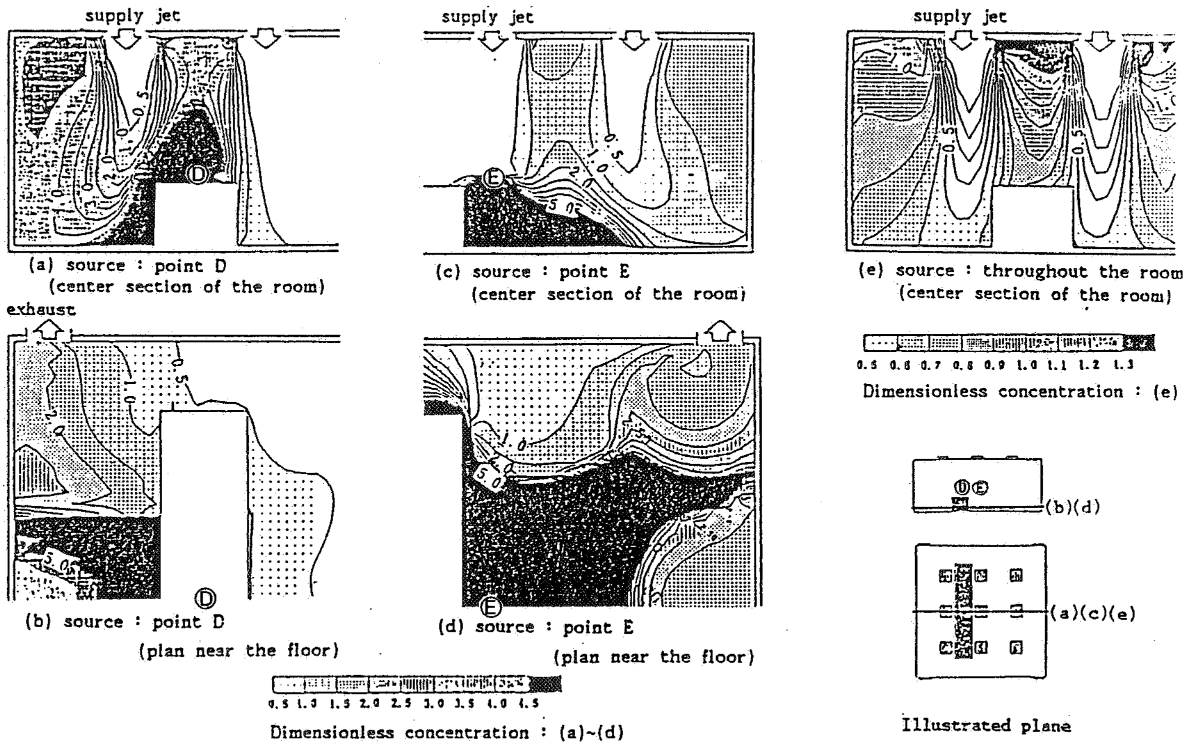


Figure 21. Contaminant distribution in Case 3 with various source position

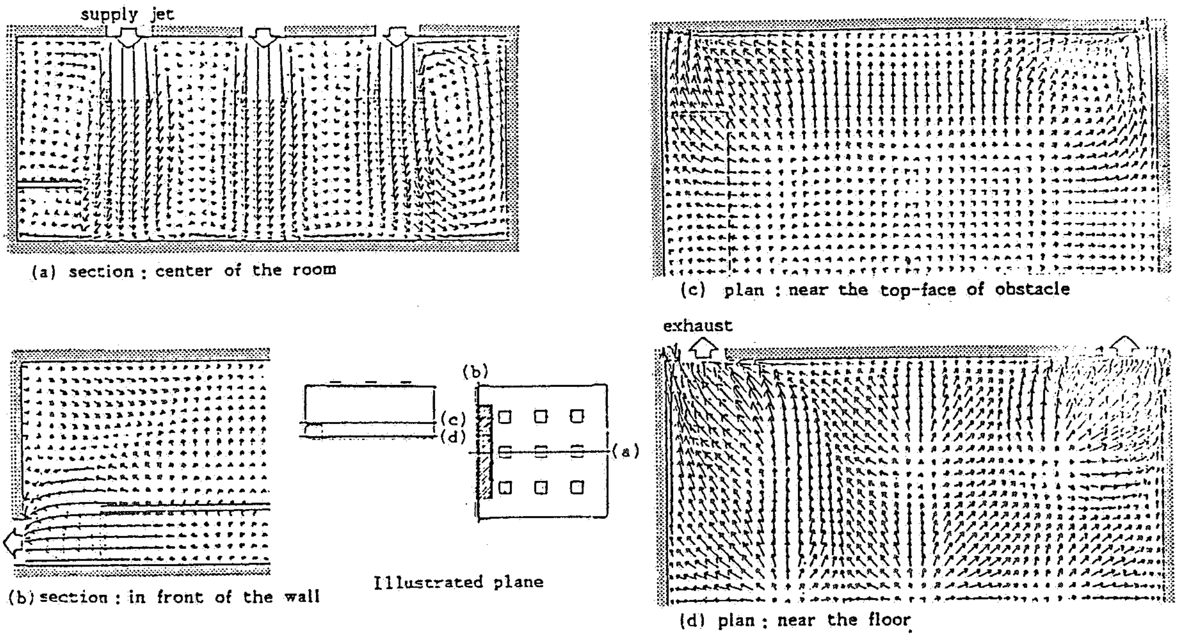


Figure 22. Velocity vectors in Case 4
(Case 4: A table-type obstacle is placed in contact with the wall)

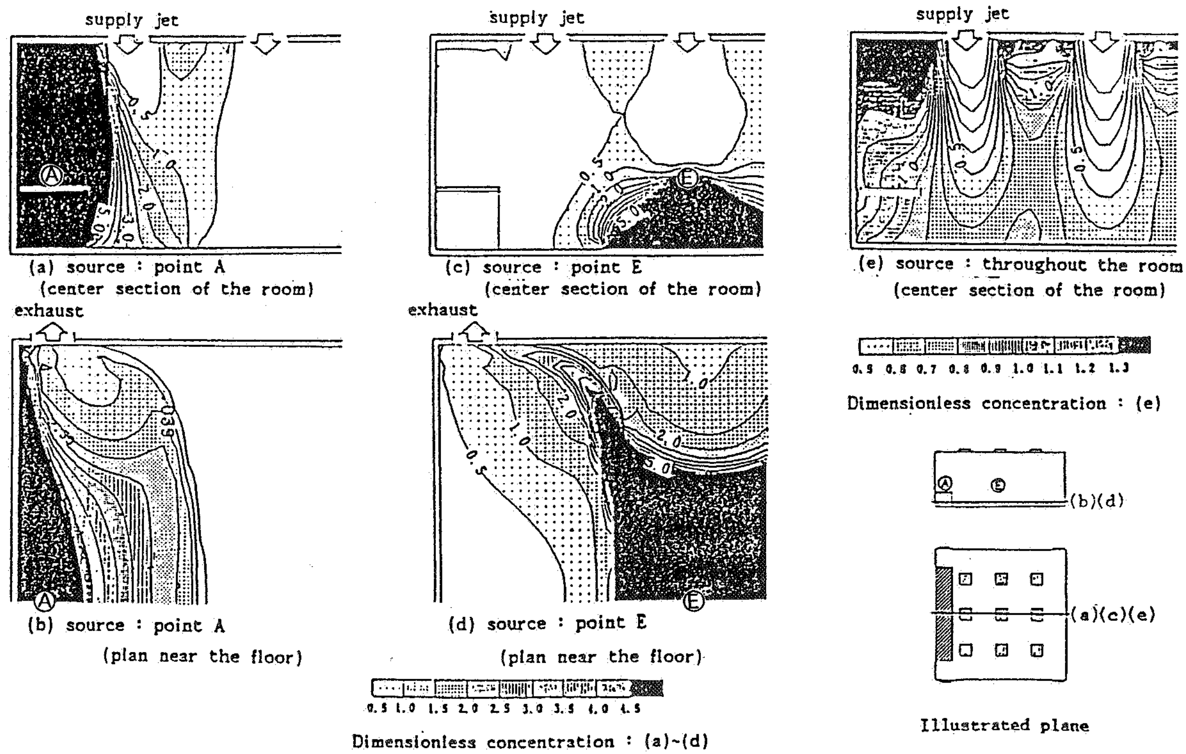


Figure 23. Contaminant distributions in Case 4 with various source position

the obstacle is rather small and is confined within the area around the table.

(2) Contaminant Diffusion Field.

When the contaminant is generated on the top-face of the table (Point A), the region from the floor to the ceiling is contaminated highly (Figures 23(a),(b)). But the contaminated area is limited to the left third of the room. The value of SVE1 in this case is 1.4 and much smaller than in Case 1 (2.1). Hence the flow field with a table-type obstacle is much more efficient to exhaust contaminant than that with a box-type obstacle. The value of SVE2 in this case is 2.5 and the same as in Case 1 (2.5).

When the contaminant is generated at the center of the room(Point E), it spreads around the center near the floor (Figures 23(c),(d)). The air around the table is very clean since the three supply jets along the side wall attack the table. The value of SVE1 is 1.3 and considerably smaller than in Case 1 (1.6). It can thus be concluded that a table-type obstacle is superior to a box-type obstacle from the view point of ventilation efficiency of SVE1. The value of SVE2 is 4.2 and the same as in Case 1 (4.3).

The value of SVE3 (Figure 23(e)) is high in the area above the obstacle, particularly near the ceiling, but it is smaller than in Case 1 (Figure 17(e)).

14.5 COMPARISON OF CONTAMINANT DIFFUSION FIELDS BY MEANS OF SVE1, 2, 3

(1) Study Based on SVE1

The values of SVE1 for all cases and for all contaminant generation points are given as the upper line in each space in Table 6. SVE1 shows larger value when the contaminant is generated near the wall.

(2) Study based on SVE2

The values of SVE2 are tabulated as the lower line in each space in Table 6. SVE2 shows smaller value when the contaminant is generated near the wall. It increases as the source point moves towards the center of the room.

(3) Study based on SVE3

A high value for SVE3 appears near the ceiling for all cases. When a recirculating flow is formed around the obstacle, SVE3 becomes higher in that region. The supplied clean air takes longer way to reach these areas, so there is much greater possibility that the air in this region will be contaminated.

15. CONCLUSIONS

It is confirmed that numerical simulation of the velocity and diffusion fields in a room is very useful in comprehending flow and diffusion patterns. The characteristics of the airflow and contaminant diffusion in a room with ceiling supply openings are summed up as follows.

(1) Mean flow structures of the airflow are modeled very well as serial combinations of flow units, which consist of one supply jet and the rising streams around it.

(2) The resulting diffusion field is mainly caused by the convection of the mean airflow. The structure of the diffusion fields also becomes very clear by introducing the concept of flow units.

(3) The supply openings have great influence on the flow fields and also a rather large influence on contaminant diffusion fields. When the numbers of the supply openings are decreased, the flow units corresponding to the eliminated supply openings disappear and the remaining flow units expand.

(4) The significant effect of the placement of an obstacle on the flow field is usually confined within the space around the obstacle. But the flowfield within the 'flow units' in which the obstacle exists is influenced greatly.

(5) Even if the effect of the placement of an obstacle on the velocity field seems to be small, the contaminant diffusion field is often influenced greatly by the arrangement of a flow obstacle.

(6) The table-type flow obstacle is generally superior to the box-type flow obstacle from the view point of ventilation efficiency.

(7) The new scales of ventilation efficiency, which are the spatial average concentration (SVE1), the mean radius of diffusion (SVE2), and the concentration in case of contaminant generated uniformly throughout a room (SVE3), are very useful measures for comparing the different diffusion fields and for quantitatively comprehending diffusion properties. These scales are strong tools that summarize in clear fashion very complex information on room diffusion fields, which is hard to characterize clearly by any other means.

Table 6. Values of SVE1 and SVE2 (Case 0 without obstacle and Case 1~4 with obstacle)

Case No. \ sources	near the wall (point A)	under supply jet (point B)	between supply jets (point D)	center of the room (point E)
Case 0 (without obstacle)	1.7 *1 3.1 *2	1.3 3.2	1.5 3.6	1.4 4.2
Case 1	2.1 2.5	—	—	1.6 4.3
Case 2	—	1.9 3.7	—	1.6 4.2
Case 3	—	—	1.7 3.2	1.5 4.0
Case 4	1.4 2.5	—	—	1.3 4.2

*1 upper line of the space : SVE1 (non-dimensionalized by Co)
*2 lower line of the space : SVE2 (non-dimensionalized by Lo)

ACKNOWLEDGMENT

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Discussion

Paper 32

J.Van Der Maas (LESO-EPFL, Switzerland)

You have shown that CFD allows the calculation of forced flows. Which developments do you foresee for mixed convection? Which problems do you think will remain unsolvable for a long time?

S.Muracami (Tokyo, Japan)

We have finished the simulation of the flowfield of mixed connection by means of the K-e model with certain accuracy. Furthermore, we have achieved the simulation of the same flowfield by means of Algebraic-stress model with higher accuracy. These results will be reported in the forthcoming AIVC Conference and ASHRAE meeting, etc.

J.Axley (MIT, USA)

In spite of impressive analytical capabilities proved by CFD we must recognise that CFD analysis has the major limitation that it presently can be applied to the analysis of flow regimes of relatively simple geometries (e.g. more or less single rooms) - driven by relatively simple boundary conditions (e.g., steady boundary conditions). CFD analysis will not be able to be applied to whole building simulation for some time.

S.Muracami (Tokyo, Japan)

Yes, CFD has many difficulties in its application now. But its history in this field is only 10 years or so. Since the development of CFD method and the super-computer is rapid, the successful applications of CFD into the ventilation problems are increasing rapidly. Its future seems to be very promising. But, as you say, it will take some time for the application of simulation to whole building. The concept of a micro-macro mixed model is very helpful to overcome such type of difficulties. The treatment of complex boundary can be solved by the technique of composite grid adaptive grid, BFC, FEM, etc. The simulation of unsteady flowfield has become rather easy by the development of Large Eddy Simulation.

VENTILATIOAN SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 33

NATURAL PROVISION OF DWELLINGS WITH SUPPLY AIR BY THE "DORTMUND VENTILATION" SYSTEM

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SUMMARY

The ventilation system described here combines a central air shaft in the hall area with a mechanical waste air extraction system in the bathroom and in the kitchen. If there is a large amount of moisture in the dwelling, the volumetric flow of the waste air fans is increased, the increase being controlled by means of hygrostats.

This ventilation system ensures adequate ventilation of the dwelling. In dwellings without a supply air shaft, this hardly applies any more with the installation of windows with very low joint permeability.

1. INTRODUCTION

The Heat Conservation Ordinance in the Federal Republic of Germany /1/ requires that, when new windows are installed, there be a minimum joint thickness. Thanks to competition between the manufacturers, this has been exceeded by a wide margin. Thus there are today quasi joint-tight windows. These new windows considerably restrict the natural flow of supply air for conventional shaft ventilation systems. Wegner /2/ investigated a number of dwellings with highly noise-insulating ventilation windows which were partly with several seals and measured a mean air exchange of n = 0.35 per hour. Ehrhorn and Gertis, on the other hand, demand in /3/ a minimum air exchange of n = 0.8 per hour for the transitional seasons (autumn and spring) to remove the moisture produced, and for the winter season n = 0.5 per hour. The consequence of the inadequate air exchange is an increase in the steam and pollutant concentration in the room air. In an investigation of the damage caused by mould in modernised dwellings /4/, Ehrhorn found that just under one third of all cases of damage can be attributed to an excessively high room humidity.

To restore a natural supply air route, the "Dortmund ventilation" system was designed as a centralised supply air system in the centre of the dwelling (e.g. in the hall). Furthermore the exhaust air shafts of the "Berlin ventilation" system were fitted with hygrostatically controlled fans to ensure that the moisture produced was also reliably removed by short-term ventilation even with peak loads (cooking, showering etc.).

2. TEST DWELLING

Figure 1 shows the layout of the test dwelling and the adjacent corner dwelling. In the corner dwelling the only interior room is the WC. The ventilation for this room is the widely used "Berlin ventilation" system. The exhaust air is removed via a shaft with natural ventilation. Since the kitchen, which is externally situated in this dwelling, and the bathroom each have a window for ventilation, there is no exhaust air shaft here. The State Building Regulations do not lay down additional ventilation installations for kitchens and bathrooms with external wall. This means that the kitchen and bathroom have to be ventilated, as required, by window ventilation. Only for the internal WC is there an additional supply air flow through the leaks in the building shell.

In the test dwelling, there are in the bathroom and the kitchen the exhaust air shafts of the "Berlin ventilation" system, fitted in addition with an exhaust air fan. Furthermore, in the hall, directly next to the front door, there is the supply air shaft of the "Dortmund ventilation" system for the natural back-up flow of outside air. To conduct tests, there is the possibility of sealing the opening between the supply air shaft and the hall and to remove the fans in the exhaust air shafts. With this arrangement, the "Berlin ventilation" system is achieved. Figure 2 shows a cross section through the whole building. It reveals the supply air shafts of the "Dortmund ventilation" system and the exhaust air shafts of the "Berlin ventilation" system. This makes clear that the supply air can be fed into the building as required on street level or through the roof. For the purpose of comparison there are in the house three other dwellings with the same layout, and an attic dwelling with a similar layout. These dwellings are also equipped with the natural supply air shaft of the "Dortmund ventilation" system and the exhaust air shafts of the "Berlin ventilation" system.

3. CONDUCT OF INVESTIGATIONS

In all five dwellings with the "Dortmund ventilation" supply air shaft, the temperatures and relative humidities of the living and bedrooms were measured constantly and stored as hourly mean values. A survey of the inhabitants of these dwellings revealed that the user behaviour of the inhabitants on the ground floor and the first floor is comparable.

On 12.03.1989, the supply air shaft in the dwelling on the ground floor was sealed, so that in this dwelling only the "Berlin ventilation" exhaust air system is installed. Figures 3 and 4 show the absolute humidities determined for the months of February, March and April 1989 as a difference to the absolute humidity of the outside air. It has proven useful to present the absolute room humidity as a difference to the outside humidity, the purpose being on the one hand to highlight the window ventilation behaviour of the inhabitants and, on the other, to eliminate the influence of the outside climate when considering various periods (as in Figure 5). In Figure 3 it is possible to see from the very low absolute humidity differentials that the bedrooms on the ground floor and the first floor are regularly ventilated through the open window. This is confirmed by the survey conducted among the inhabitants. The inhabitants are elderly people who have been used from childhood to keeping their bedroom doors closed and to opening the window regularly. With the regular ventilation of the bedroom through the window, it is not possible to make any distinction between the two ventilation systems. With regard to the living rooms, however, such a difference is present, as can be seen in Figure 4. This figure makes clear that the room air of the living room on the ground floor is clearly more humid than that on the first floor. This is not the case for the period before 12.03.1989, where "Dortmund ventilation" was installed in both dwellings.

This result is confirmed by a test in the test dwelling on the third floor. Figure 5 shows the absolute room humidities as a difference to the outside humidity in two successive January weeks. In the first week the supply air shaft of the "Dortmund ventilation" system was open, and in the second week it was closed. In this examination, it is necessary to use absolute humidity differentials in order to eliminate the different outside climate in the two weeks. If one considers the whole period, it can be seen that the rooms with an installed supply air shaft are drier than those with a closed supply air shaft. It is not appropriate to consider individual instantaneous values, because even in the same dwelling the user behaviour is different in two successive weeks.

If we consider the volumetric flows with the installed natural supply air shaft of the "Dortmund ventilation" system and the natural exhaust air shaft of the "Berlin ventilation" system, it can be seen that in Winter the dwelling is more than adequately ventilated. In January and February, a volumetric flow of 160 m^3/h is measured at the two exhaust air shafts. In Summer there is no evidence of major air movements because of the lack of, or only very slight difference in density between the room air and outside air. Since the system described is designed as a basic ventilation, i.e. there is no intention to do without window ventilation all together, the volumetric flows in winter are too high, which is not desirable with regard to energy management. This and the only very low volumetric flows in summer necessitated the regulation of the volumetric flows by installing exhaust air fans. These fans each move a volumetric flow of 30 m^3/h at their basic stage. If there is a high humidification, the fans are switched to a higher stage, controlled via a hygrostat. Here they each move a volumetric flow of 80 m^3/h.

In humidity tests, an extremely high humidity was simulated by evaporating water. These tests showed that the humidity level was buffered by the adsorption of the steam in the wall and various items of equipment. Since the hygrostats in the bathroom and kitchen cannot register a high humidity level in the living room and bedroom, and since the humidity level in the room is kept high by desorption of the steam from the walls and items of equipment, it is not possible to dispense with sudden window ventilation.

RESULTS

If there is regular window ventilation, it is guaranteed that dwellings with the "Berlin ventilation" exhaust air system are adequately ventilated. But investigations in modernised rented housing construction, such as /4/ and /5/, show that in many cases no, or at least only inadequate ventilation through the window is practised. An extensive survey conducted in Belgian social housing /6/, covering 1115 one-family houses and 1219 flats, reveals that 30 % of all rooms are never ventilated. A supply air system such as the "Dortmund ventilation" system is absolutely necessary.

The investigations showed that dwellings with the "Dortmund ventilation" natural supply air system and the "Berlin ventilation" exhaust air system are drier than those with only the "Berlin ventilation" exhaust air system. Since the function of a natural exhaust air system is limited in summer, and volumetric flows arise in winter which exceed the necessary level, a hygrostatically controlled mechanical exhaust air system is more beneficial.

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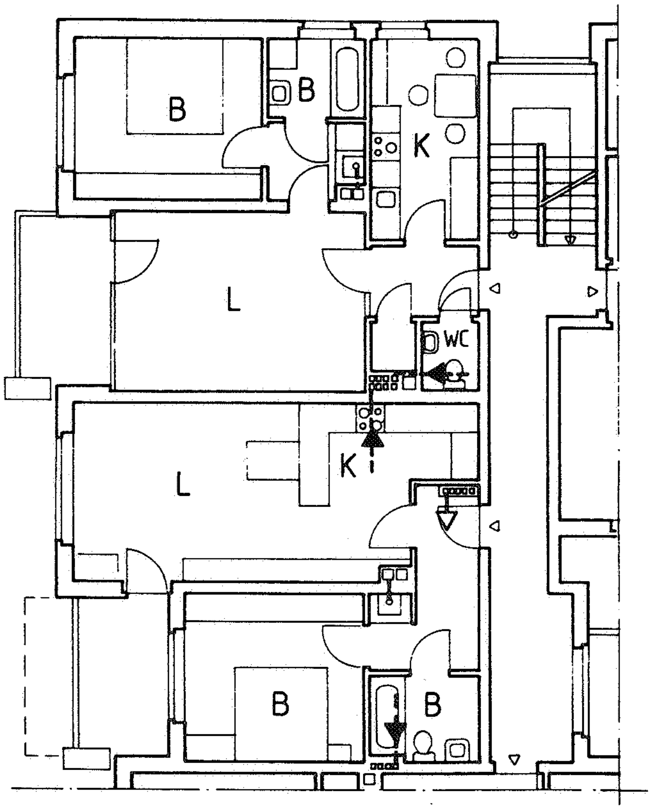

Figure 1: Layout of the test dwelling and neighbouring corner dwelling with arrangement of shafts

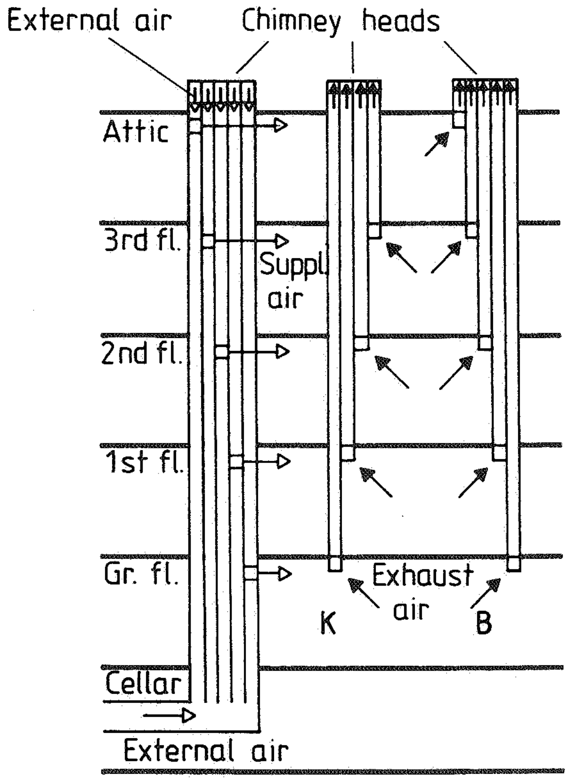

Figure 2: Cross section through the building

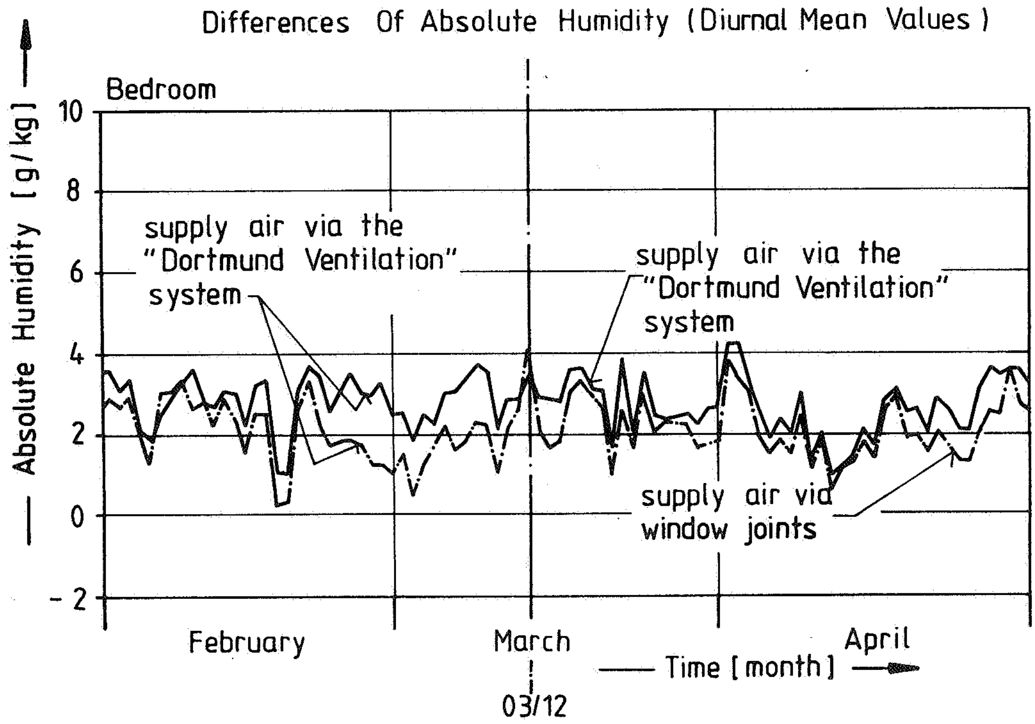

Figure 3: Test results for the bedroom on the ground floor and on the first floor

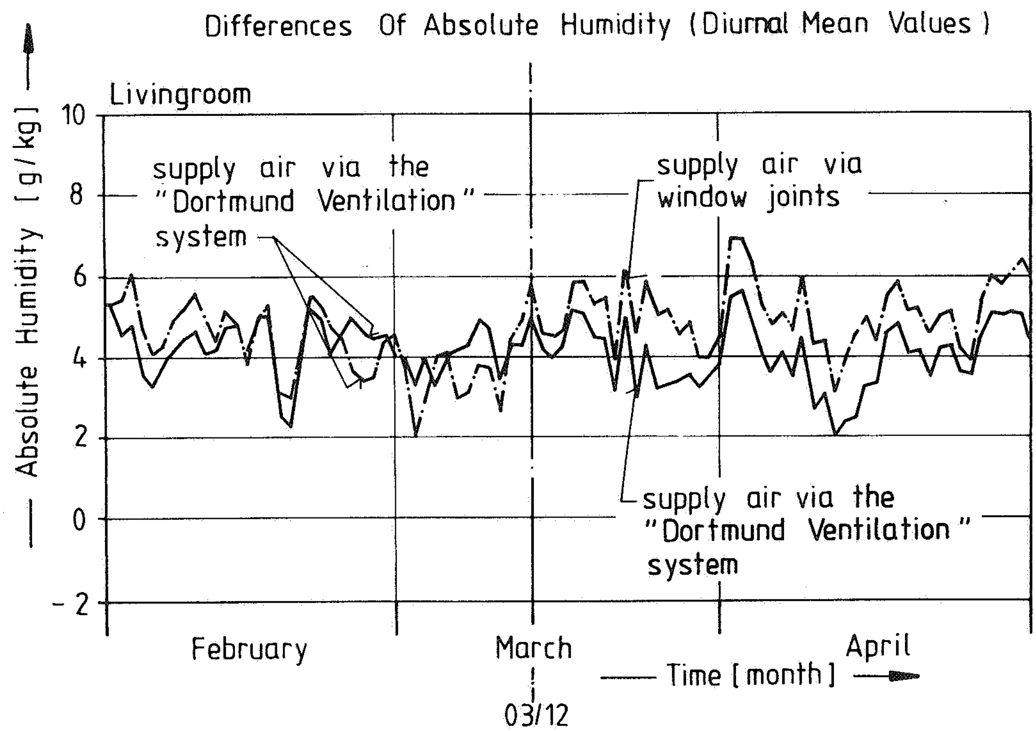

Figure 4: Test results for the living room on the ground floor and on the first floor

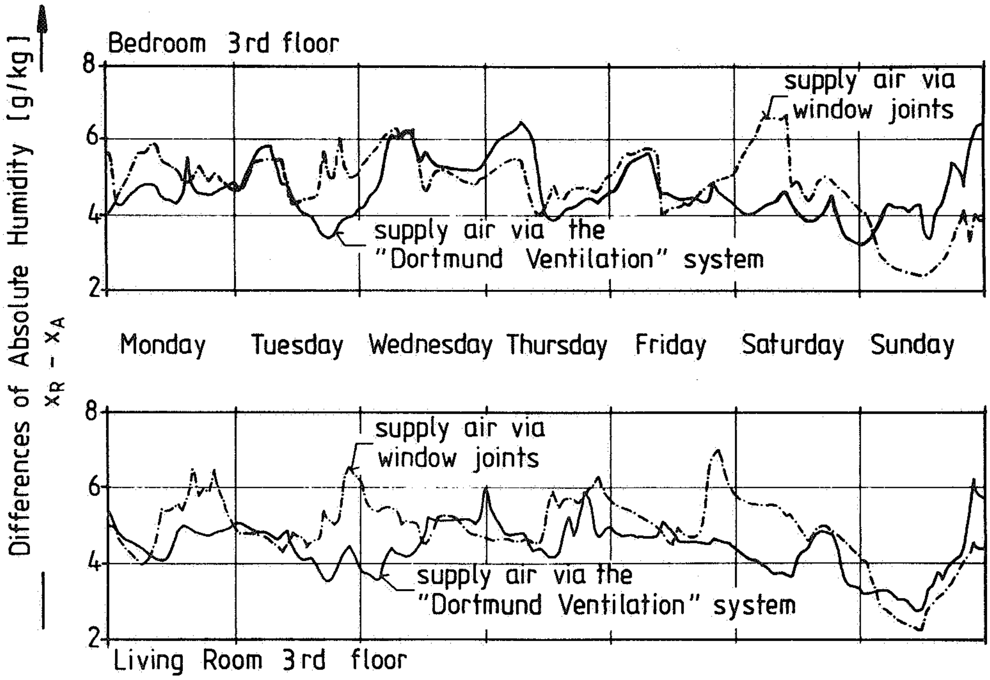

Figure 5: Test results for the bedroom and living room in the test dwelling

VENTILATIOAN SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 34

DETERMINATION OF AIR EXCHANGE RATES FOR DEMAND CONTROLLED VENTILATION

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SYNOPSIS

In this paper the required ventilation air flow rates in residences with different pollutant loads are considered. The calculative study was carried out by using the load data presented in the literature. The results of the study were applied in the development and dimensioning of demand controlled ventilation systems.

The first stage of calculations was to determined the required ventilation air flow rates (range) with different loads for each type of rooms separately. In the analysis of required ventilation air flow rates due to material emissions, the Monte Carlo-method was applied. The second stage of calculations was to analyse indoor air contaminant concentrations in an apartment as a whole using the required air flow rates in room spaces. The time-dependant occupant behaviour in the apartment was based on a certain assumption. Calculations with constant air flow rates ventilation were also done.

When considering different load factors; human based odours, the odours of smoking, humidity loads and contaminant emissions of materials were taken into account. The required ventilation air flow rates in different load situations were determined to guarantee good indoor air quality and humidity conditions, and to prevent health risks due to material emissions, as well.

In most calculation cases it was assumed that indoor air is fully mixed. In addition, the effect of air flows on 2-dimension contaminant field was analysed.

1 BACKGROUND

In Finland the indoor climate and ventilation of residences must fulfil the Building Code which define the satisfactory level of indoor climate in normal conditions. On the other hand the loads of ventilation are variable and therefore the ventilation air flow rates should also be adjustable by occupants.

At the moment there is no clearly recommended air flow rates for demand controlled ventilation systems. The correct required air flow at any given time or room space depends on load factors. With demand controlled ventilation it is possible to minimize energy consumption of ventilation, as well.

In this paper some typical loads and criterias as well as air flow calculations are presented. Loads and criterias are based on literature. This paper deals only with residences.

2 LOADS AND AIR FLOWS

The most important loads in residences are human based odours, odours of smoking, humidity, building and furniture materials and heat loads. The required ventilation rates were determined to guarantee good indoor air quality and humidity conditions, and to prevent health risks due to material emissions.

2.1 Human based odours

Human based odours may be handled by using carbon dioxide (CO_2) emissions and concentrations. If the percentage of dissatisfied is 20 % the CO_2 concentration is about 1000 ppm (cm^3/m^3) and required air flow is 7-8 dm^3/s per person. If there is no special requirements for odour of air, then it could be possible to use smaller air flows, e.g. 4 dm^3/s per person corresponding to the CO_2 concentration 1500 ppm.

In a case more persons than normally (e.g. visitors) occupy the room space, the air flow should be at least 8 dm^3/s per person because dispersing of odours decreases. When physical activity increases the required air flow may be 2-4 times greater compared to a condition where people are sedentary.

2.2 Odours of smoking

Odours of cigarette smoke are usually dealt with carbon monoxide (CO) concentration. Acceptable concentration is 1-2 ppm. Continuous smoking at steady-state conditions requires 100-200 dm^3/s air flow per smoker. With the typical smoking frequencies (1-4 cig/h) the needed air flow is only 20-40 dm^3/s per smoker.

2.3 Humidity loads

Too high humidity level of indoor air may cause mould growth and structural damages. According to calculations the required air exchange rate in residences should be 0.1-0.4 1/h if we use mean humidity loads. The greatest loads appear in the bathroom during shower. Required air flow is then over 50 dm^3/s. Other loads in bathroom requires only 10-30 dm^3/s air flow. In kitchen the greatest air flows, about 15-40 dm^3/s, is needed during cooking.

2.4 Materials

Almost all materials release harmful contaminants. Best known compound is formaldehyde, that is typically emitted from particle board.

To calculate the required ventilation, contaminant emissions of materials and highest accepted concentrations are needed. Both factors are very uncertain and, therefore, very difficult to used as exact initial values of calculations. One method to deal with that is to use distributions of measured emissions and calculate air flows using Monte Carlo-simulation which picks up the emissions (combinations) randomly according to the statistical probability.

Concentration limits can be estimated by the aid of limit values used at workplaces (MAC,TLV etc.). In Finland contaminant consentrations in residences must be lower than 1/10 of limit values at workplaces. Some concentration limits for residences are shown in table 1.

Emission distributions can be found out by using quite few facts about material emissions. Only the mean (or median), minimum and maximum emissions of each type of material and compound are needed. Some typical materials and emissions are shown in table 2. As an example, the normal distribution of emission is calculated in a following way:

Initial parameters

$\bar{E}$ is mean emission
E_{min} minimum emission
E_{max} maximum emission

and

RND_1 and RND_2 which are random numbers between 0...1.

To calculate the normal distribution of emission (E_i) normal deviation (S) is needed. The mean value of S is 0 and standard deviation is 1.

$$S = \sqrt{-2\ \ln(RND_1)}\ \cos(2\pi RND_2)$$

If $S < 0$ then $E_i = \bar{E} + S(\bar{E}-E_{min})/\sqrt{3}$

otherwise $E_i = \bar{E} + S(E_{max}-\bar{E})/\sqrt{3}$

where the terms $(\bar{E}-E_{min})/\sqrt{3}$ and $(E_{max}-\bar{E})/\sqrt{3}$ are approximate values of standard deviations.

Repeating calculation with different random numbers several (even hundreds) times we will get a result which is a distribution of required air flow rates.

In figure 1 the results of two cases concerning room which volume is 55 m^3 are shown. In the first case the floor (22 m^2) is covered with parquet (formaldehyde emission) and the second case with PVC-carpet (phenol emission). Other formaldehyde emitting materials are furniture (5 m^2 painted particle board) and textiles (5 m^2). Wall (55 m^2) material is painted concrete (radon emission). In the first case there is 95 % probability that 0.80 1/h air exhange rate will quarantee the good indoor air quality.

According to the calculations, the minimum air exchange rates for hole apartment should be 0.1-0.5 1/h to prevent too high contaminant concentrations.

Table 1. Concentration limits for contaminants from materials /4/,/5/.

Contaminant	Concentration limit Satisfactory air quality	Good air quality
Formaldehyde, µg/m^3	150	60
Phenol, µg/m^3	1900	400
Radon, Bq/m^3	200	100

Table 2. Emissions of some building materials /1/,/2/,/3/.

Contaminant Material	Emission Mean	Variation
Formaldehyde	(µg/m^2 h)	
Particle board	200	100-300
Painted particle board	6	0-15
Parquet	30	15-100
Textiles	15	0-65
Radon	(Bq/m^2 h)	
Concrete	27	15-30
Painted concrete	7	3-14
Phenol	(µg/m^2 h)	
PVC-carpet	30	30-70

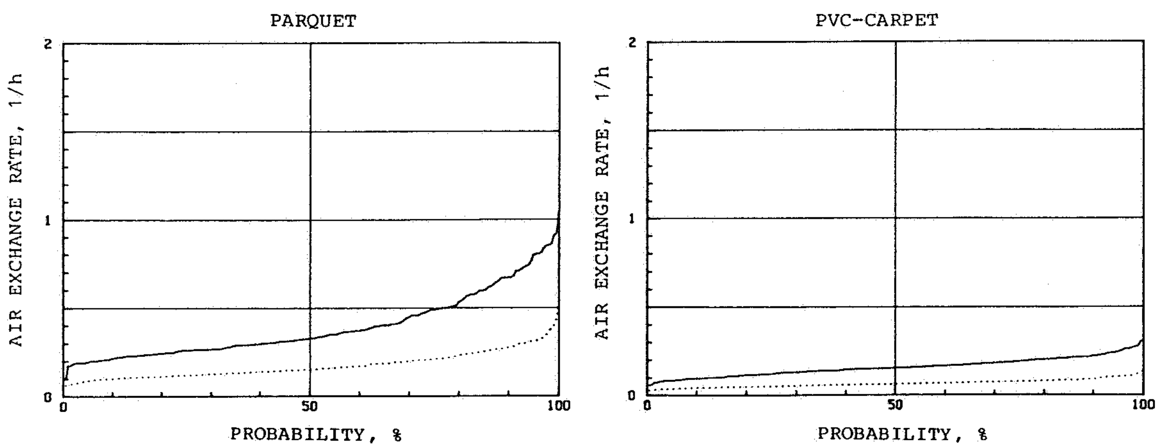

Figure 1. Distribution of required air exchange rates with two concentration limits, ____ = good, ... = satisfactory indoor air quality /5/.

4 SUMMARY

Summary of calculated air flow rates is presented in table 3.

According to the dynamic calculations of whole apartment using time-dependant occupant behaviour, the suitable air exchange rate range is about 0.4-2.0 1/h. The estimated mean air exchange rate of the day (24 h) was about 0.75 1/h. Using the air flows recommended in Finnish Building Code we will get the mean (constant) air exchange rate that is about 1.0 1/h. So the demand controlled ventilation improves the indoor air quality by decreasing the mean air exchange and energy consumption.

Table 3. Calculated air flow rates for residences /5/.

Mean air exchange rates for apartment		
- material emission loads	0.1-0.5 1/h	
- humidity loads	0.1-0.4 1/h	
- heat load (in summer)	1.0-3.0 1/h	
Air flow rates for different loads		
Sleeping rooms		
- human based odours, at night	4 dm^3/s,	person
daytime	8 dm^3/s,	person
Living-rooms		
- human based odours, normally	4-8 dm^3/s,	person
visitors	8 dm^3/s,	person
Kitchen		
- humidity loads, during use	15-40 dm^3/s	
Bathroom		
- humidity loads, during shower	50-100 dm^3/s	
bath	20-30 dm^3/s	
washed clothes drying	10-20 dm^3/s	
Sauna		
- humidity loads, during use	4 dm^3/s,	person
	($\approx$5-10 1/h)	
Toilet		
- removal of odours in 10 minutes	20 dm^3/s	
Wardrobe		
- material emission loads	0.5-1.5 1/h	
Smoking		
- odours of smoking	20-40 dm^3/s,	smoker

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VENTILATION SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 35

ROOM AIR TEMPERATURE CONTROL IN DEMAND CONTROLLED VENTILATION SYSTEMS

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1. SUMMARY

The controllability of room air temperature in different heating systems connected to demand controlled ventilation systems was studied. Studied ventilation systems were exhaust, supply and exhaust and a system with exhaust and an individual supply to each apartment.

Studies were made using PIPNET-simulation program package. It is designed to allow detailed simulation of entire building systems: the building shell, heating and ventilating plant and the dynamic thermal interactions among the subsystems.

First single zone calculations were performed. The control of the supply air temperature in warm air heating was studied in case where heating coil was designed for airflow of 1,0 m^3/s. The airflows varied from 1/4 to 1/1. The supply air temperature was controlled using PI-controller. According to the calculations it is possible to control the room air temperature within +/- 1K even if the heating coil is overdimensioned.

The dynamic behavior of a water radiator and a floor heating panel system was studied in an exhaust ventilation system. The control of the room air temperature was also studied with P-, PI- and PID -controllers. Simulations showed that a thermostatic valve is not fast enough if the airflows in the room change rapidly in the exhaust ventilation system, while the supply and extract ventilation system PI- and PID -controllers kept the room air temperature quite well in the set value.

Secondly multizone calculations with an apartment of 89,5 m^2 were done. The heating system consisted of floor heating panels in 4 rooms and water radiators controlled by thermostatic valves in other rooms. The water rate in the floor heating panels was constant and the supply water temperature was compensated according to the outdoor temperature. The duration curves of room air temperature with different ventilation systems were calculated using daily heat load and ventilation profiles.

Based on the calculations recommendations for the range of air flow rates to provide acceptable room air temperature will be given.

2. INTRODUCTION

Many research projects have shown that there are a lot of shortcomings in the operation of ventilation systems and devices. Most of the existing air change systems in residential buildings do not even meet the present requirements of building codes. The biggest problems in mechanical ventilation systems are draught, stuffy indoor air, odours spreading in the flat and between the flats, noice and condensation. Most of the systems are also difficult to use and maintain.

These lacks in the operation have been initiators/stimulators for the project "Ventilation systems of the future residential buildings". Demand controlled ventilation systems that fit in the future way of living are developed. The aims of this project are to achieve

- good ventilation effectiveness,
- a possibility for inhabitants to control the ventilation rate,
- stability of the system,
- low energy consumption,
- a possibility to change the quality level of the system afterwards.

These aims are set so that the ventilation system has technical preconditions to maintain high indoor air quality and good thermal comfort and energy economy as well.

The study of room air temperature control of demand controlled ventilation systems in residential block of flats is discussed in this paper.

3. WARM AIR HEATING

Studies were made using PIPNET-simulation program pagage. The control of the supply air temperature was studied in case where the heating coil was designed for an airflow of 1,0 m³/s. Two connections of the coil were simulated in figure 1. The valve characteristic of the 3-way valve was logarithmic for the main branch and linear for the by branch. Lograrithmic valve characteristic was chosen to make the characteristic of the whole heating coil connection as linear as possible.

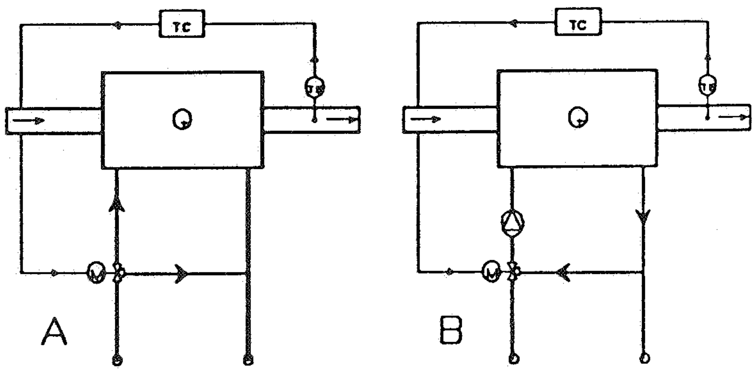

Figure 1. Simulated heating coil connections. Variable water flow control (A) and variable inlet water temperature control (B).

The airflows varied from 1/4 to 1/1 in simulations. The supply air temperature was controlled using PI-controller. The control values K_P and K_I were determined in a situation where the gain of the system was greatest. This was done by making a step response test to the system.

According to the calculations it is possible to control the supply air temperature within ± 1K even if the heating coil is overdimensioned.

4. WATER RADIATOR HEATING

Dynamic behavior of a water radiator was studied using PIPNET-simulation program package. Calculations were made with a single room model. Room air temperature was controlled using thermostatic valves. Some simulations with PI- and PID-controllers were also carried out.

The purpose of these simulations was to study how the room air temperature responds to an increase in the air change rate of the room. The simulated system consists of a single room, radiator, thermostatic valve and an mechanical exhaust ventilation system. Supply air to the room comes through the exterior wall and it's temperature is the same as the outside air's temperature.

If the characteristic of the thermostatic valve is logarithmic, then the characteristic of the whole system (valve, pipe and radiator) is almost linear. This means better controllability of the system than when using a valve with linear characteristic.

While the thermostatic valve is a P-type controller, there is a permanent deviation. The size of the deviation depends on the control curve, the room itself and the loads. In figure 2 there is shown the room air temperature as a function of the proportional position of the value in 4 load conditions. For example in the case 1 the room air temperature settles to 20°C when $H/H_{100} \approx 0,46$.

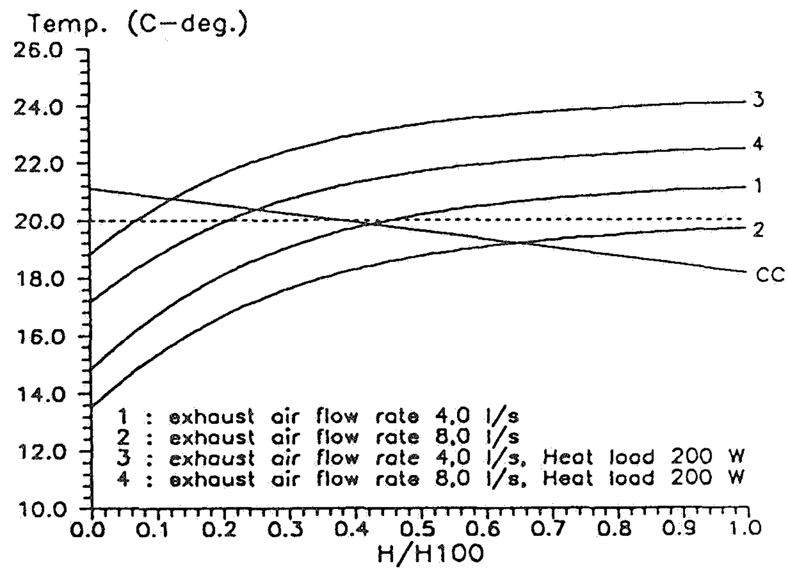

Figure 2. Room air temperature as a function of proportional position of the valve. The outdoor air temperature is 0°C.

The thermostatic valve control curve is set so that the set value 20°C is achieved when the outdoor air temperature is -20°C, exhaust air flow rate is 4 l/s and there are no other heat gains than the radiator. The room air temperature settles in a steady state situation at the intersection of the room air temperature curve and the control curve (CC).

According to the simulations water radiator heating with a thermostatic valve cannot keep the room air temperature in the set value, if the exhaust air flow rate is e.g. doubled and there is no heat load in the room. If there is heat load in the room at the same time then when the exhaust air flow is increased, the room air temperature either decreases or increases depending on wheather the heat loss due to increased air flow rate is greater or smaller than the heat load.

In figure 3 there is shown the room air temperature in different cases. The two outdoor air temperature is the same as the design outdoor temperature, i.e. is -27°C. The supply water temperature to the radiator is 70°C and the radiator 1 is dimensioned for the heating load corresponding to minimum air flow of 6,6 l/s and the radiator 2 for an air flow of 13,2 l/s. The exhaust air flow is increased from 6,6 to 13,2 l/s for 2 hours. The more effective radiator keeps the set value (20°C) better than the smaller radiator. The controllability of a radiator sets requirements for dimensioning the radiator according to the maximum heating load.

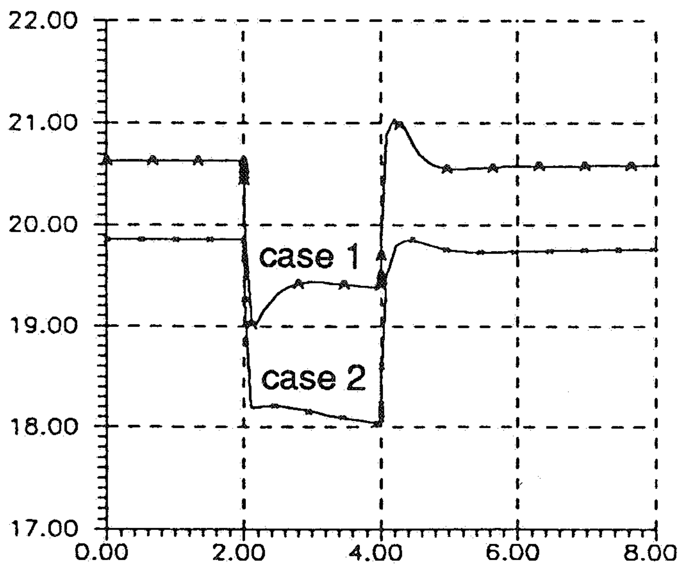

Figure 3. The room air temperature when it is controlled with thermostatic radiator valve (linear characteristic).

In figure 4 there is shown a situation equal to the figure 3 situation with radiator 2 exception with that there are now a PI-controller connnected and 3-way valve instead of thermostatic valve. The control values K_P and K_I were determined in case when the exhaust air flow is 6,6 l/s.

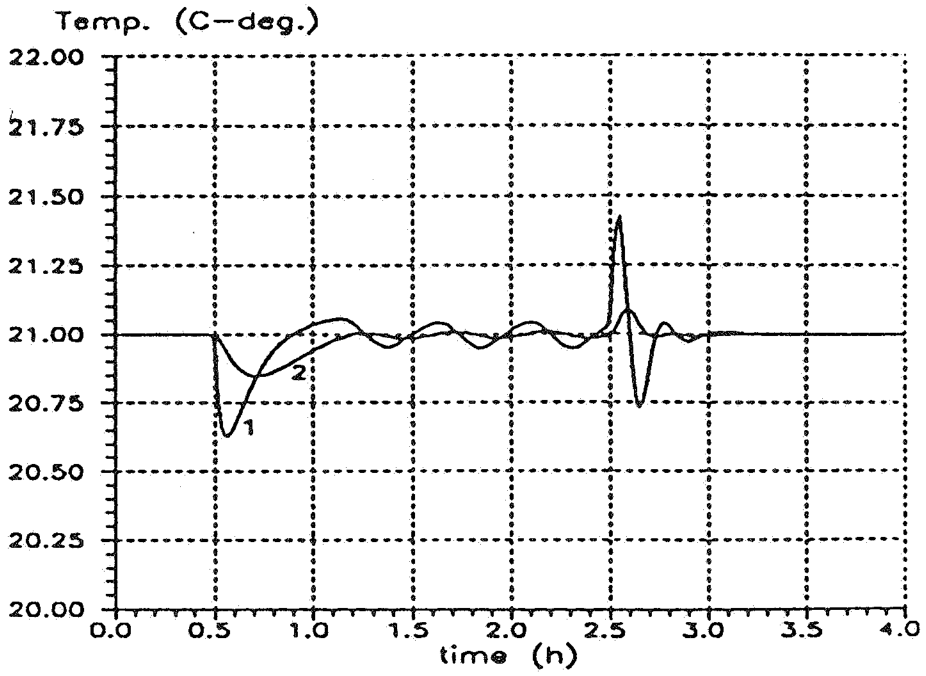

Figure 4. Room air temperature (1) and temperature of the temperature sensor (2) in the room when exhaust air flow is doubled for 2 hours.

5. FLOOR HEATING PANEL

The room air temperature control in a floor heating system and in a combined floor heating and water radiator system were studied using the PIPNET-simulation program package. Calculations were made with a single room model. There was a mechanical exhaust ventilation system.

The calculations showed that the floor heating system doesn't work when the need for heating load varies rapidly. If a radiator is added to the floor panel heating system, the situation gets much better (Fig. 5).

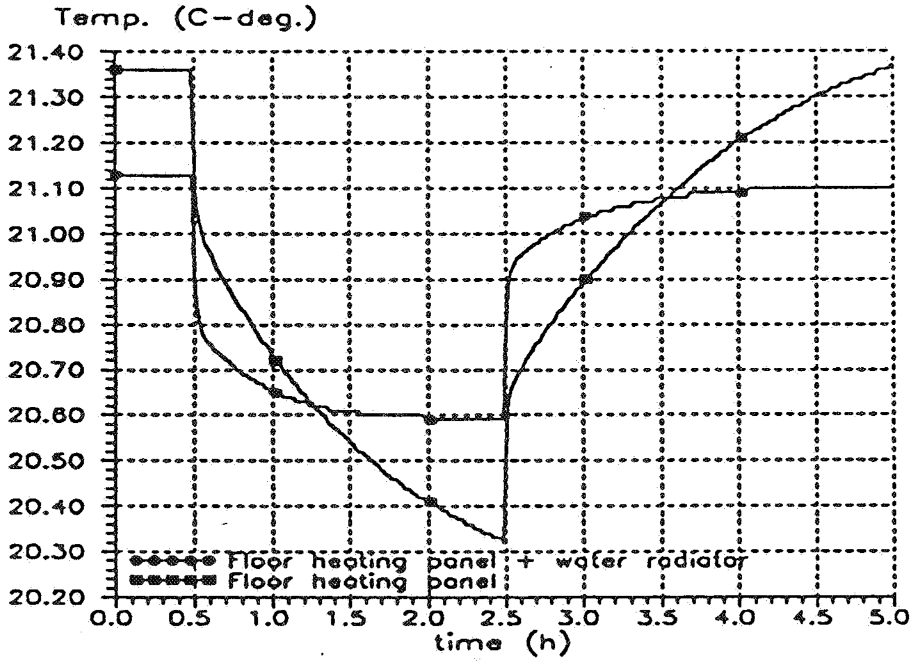

Figure 5. Room air temperature in a floor heating panel system with and without a water radiator when the exhaust air flow is doubled.

6. MULTIZONE CALCULATIONS

The purpose of the multizone calculations was to compare the performance of the studied ventilation systems. The heating system was selected according to the single zone calculations. The simulated apartment had 3 bedrooms, a living room, a kitchen, a corridor, WC, a sauna and a bathroom. Corridor, WC and bathrooms had floor heating panels and in other rooms there were water radiators. The supply water flow to the floor heating panel was constant. The supply water temperature was outdoor air temperature compensated. The radiators had thermostatic valves.

The systems were simulated with 3 different weather periods: January, April and June. The outdoor air temperature varied from 0°C to -25°C in January, from 0°C to +5°C in April and from 10°C to 25°C in June. The maximum solar radiation in June was about 800 W/m^2. The air change rate varied from 0,5 to 1,0 ach.

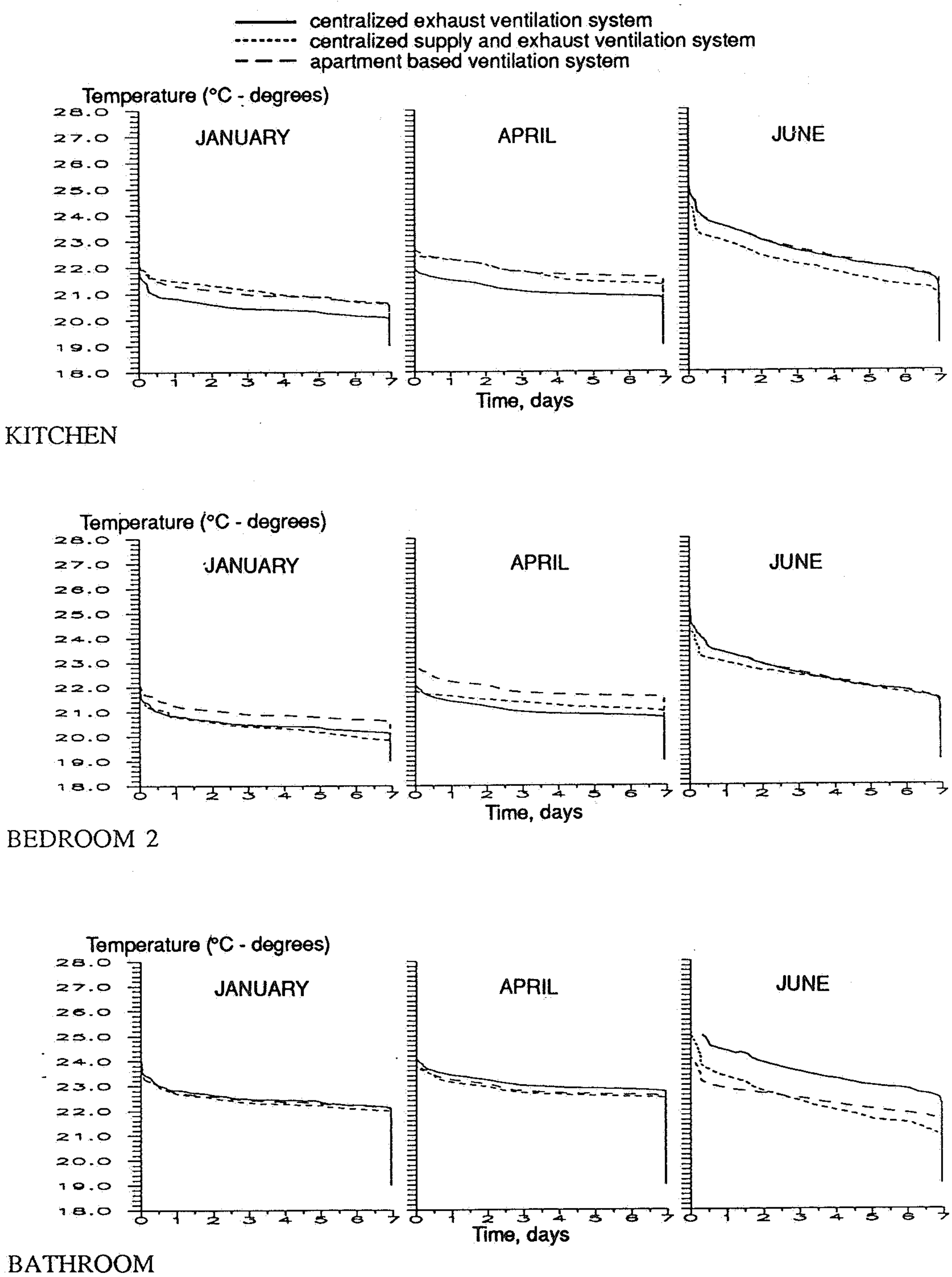

Figure 6. The room air temperature duration curves of kitchen, bedroom 2 and bathroom during the weather periods of January, April and June.

7. RECOMMENDATIONS

According to the calculations the thermal conditions of a single room do not change too much when increasing the air change of the room if the following principles are applied.

Mechanical supply and exhaust and mechanical exhaust and individual supply systems:

- the supply air temperature is over 16°C during the heating season.

Mechanical exhaust systems:

- the maximum air change rate of a room is the basic air change rate (e.g. 0,5 ach.) plus 1,0 ach. This is valid when the outdoor temperature is lower than -20°C.

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VENTILATION SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 36

Demand Controlled Ventilating Systems
- Practical Tests -

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Synopsis

As part of the IEA Research Program Annex 18 "Demand controlled ventilating systems" were tested in an one family testhouse in relating to energy and ventilating specific aspects. The investigation should show whether demand controlled systems are useful in dwellings or not. Following items were checked:

- Infiltration characteristics of the testhouse
- Ventilation characteristics of different systems like temperature distribution, air mouvement, ventilating efficiency, air exchange and air quality

Different systems were linked with various sensors for air humidity, oxidable gazes and carbondioxyd.

1. Infiltration characteristic of the testhouse

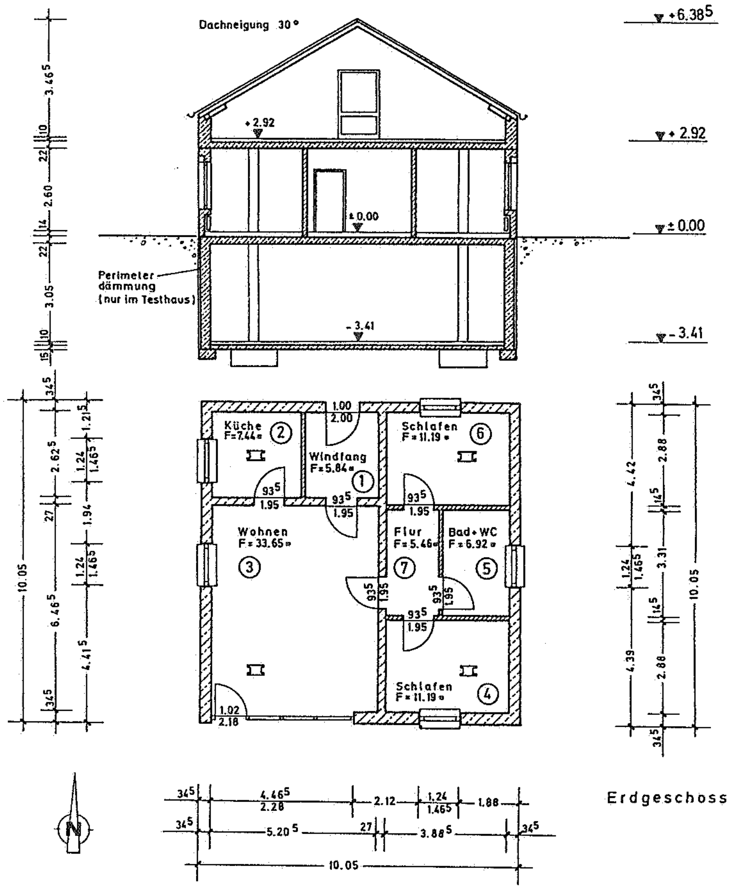


Fig. 1: Plan of testhouse

First the air infiltration characteristic of the test-building without ventilating systems has been measured. The floor area of the testhouse is 82 m^2, the height of the floor is 2,6 m. A "Blower-Door" was used to (de-)pressurize the testbuilding; in both cases a nearly identical leakage of about 300 m^3/h for a pressure difference of 50 Pa was found out. Using the equation

$$\dot{V} = k \cdot \Delta p^n \ [m^3/h]$$

a flow coefficient k = 23,9 and a pressure exponent n = 0,65 has been calculated.

window	wind-speed [m/s]	inside temperature [°C]	outside temperature [°C]	airchange [h^{-1}]
shut	0	21.3	4.2	0.05
	1.5	1.5	-0.2	0.10
	8	21.1	8.0	0.25
open	0	21.0	11.0	0.15
	1,5	21,1	7.1	0.40
	3.0	21.2	16.3	0.50
	7.0	21.2	7.4	1.11

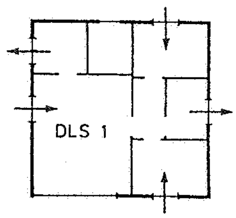
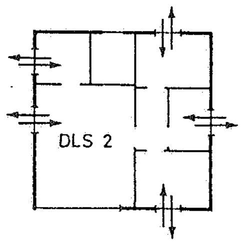
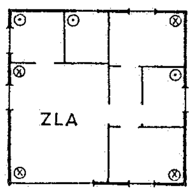
Tab. 1: Airchange rates by shut and opened window in one room by different windspeeds.

Nominal air exchange rates have been determined with the tracer gas decay method. For calm a rate of about 0,05 h^{-1} was measured, for low wind forces the air exchange rate was about 0,1 h^{-1} and for strong wind about 0,25 h^{-1}. Because of these results the testhouse can be called as "air-tight". Window ventilation caused air exchange rates between 0,5 and 7,5 h^{-1} corresponding to the number of tilted windows and different wind forces. After having measured the lenght of all joints a joint-coefficient of 0,13 $m^3/h \cdot m \cdot Pa^{2/3}$ was calculated. Pressurisation test data led to an air exchange rate of $N_{50} = 1,4\ h^{-1}$ at 50 Pa. For calm and low wind forces the equation for the "infiltration air exchange rate"

$$N_{inf} = N_{50}/20$$

could be testified through tracer gas measurement data.

2. Airflow- and air exchange rates

running mode	wind-speed [m/s]	wind-direction rate [h^{-1}]	air-change	Location of Ventilating System
off	1.5	E	0.10	Decentral Vent. Syst. 1 
max. power	1.5	E	0.55	
off	2	W	0.10	Decentral Vent. Syst. 2 
min. power	2	E	0.30	
max. power	2	E	0.60	
off	0 6.5	- W	0.10 0.25	Central Vent. Syst. 
on	2 7	E W	1.0 1.05	

Tab. 2: Measured airchange rates with decentral vent. systems. The ground plan shows the number and position of the devices.

The air flow rates of the decentral ventilating systems have been measured before installation. Maximum airflow rates of most systems were about 35 to 55 m^3/h; systems with a closing device were sufficiently airtight. In some cases the measured maximum air rates have been about 50 % lower than announced by manufacturers. Then the decentral systems were installed in the testhouse and the air exchange rates have been measured under different climatic conditions and running modes. The airtightness of the building was not affected with decentral systems with a closing device (air exchange rates about 0,10 h^{-1} for calm). Using a central ventilating system rates from 0,10 to 0,30 h^{-1} corresponding to different wind forces have been measured. Running with maximum power the decentral ventilating systems caused air exchange rates about 0,50 h^{-1} in the ground floor of the testbuilding. The central system caused rates about 1.0 h^{-1} independent of climatic conditions.

3. Ventilation perfomance

Draught effects have been noticed only with the strongest decentral system with an air flow rate of about 100 m^3/h. The other systems with lower air flow rates located at the top of the window frames or the central system did not cause draught. Using decentral systems noise levels have to be accepted. Comparing measured and calculated air change rates it was noticed, that by use of all decentral systems with combined supply- and exhaust air the measured air exchange reates have been considerably lower than the calculated ones. This effect was noticed for all running modes and can be explained by a short circuit airflow.

4. Humidity controlled systems

A relative humidity controlled ventilating system with a central exhaust fan and decentral air inlets has been installed in the testhouse. For the adjustable summer mode a range of regulation from 45 to 65 % r.h. corresponding to an exhaust airflow from 38 to 110 m^3/h was found out. For winter mode a range of regulation from 35 to 50 % r.h. was measured, corresponding to an exhaust airflow from 75 to 110 m^3/h. Using the booster divice in the kitchen air outlet the airflow could be increased to 120 m^3/h. The additional exhaust air opening was closed automatically after 70 minutes. The measured air exchange rates caused by this system lay between 0,25 and 0,65 h^{-1} corresponding to the chosen running mode and relative indoor humidity; draught effects have not been noticed. The examined system can be used for a humidity controlled basic ventilation of dwellings. The booster device allows a considerable increase of the kitchen exhaust airflow when needed.

In the kitchen and the bathroom of the testhouse decentral ventilating systems were connected with capacitive humidity sensor. Using this configuration relative humidity could be controlled without problems. For a decrease of high humidity caused by the simulation of cooking or showerbath with decentral ventilating systems a runtime of about 1 or 2 hours was necessary. For practical use however inexpensive humidity sensors have to be developed.

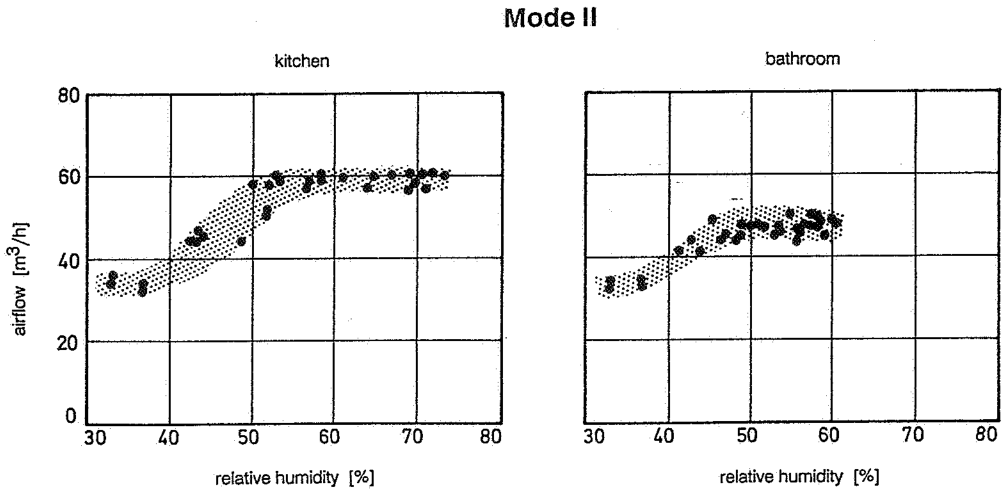


Fig. 2: Measured values of airflow through air outlets in kitchen and bathroom for different rates of relative humidity and "mode II". (Range of regulation announced by manufacturer is from 23 % to 58 % r.h.)

5. Air quality control

The regulation characteristics of the examined air quality controllers seem to be adequate for a demand controlled ventilation. The sensitivity can be chosen and allows a wide range regulation of air quality. The examined sensors reacted on tobacco smoke, solving agents and human odor. In order to decrease the air-pollution caused by the smoke of 4 cigarettes in the living room runtimes of about 1 - 2 hours were necessary with most of the decentral systems. A CO_2-controlled ventilating system could not be realized with most of the decentral systems. The air flow rates of these systems were too low to prevent an increase of the CO_2-concentration over 0,1 Vol % (Pettenkofer-limit) when for instance four persons sat in the living room. Assessing to demand controlled ventilating systems in general can be said that a control of relative humidity can be realized with air change rates about 0,5 h^{-1} (basic ventilation). For an effective control of air quality (tobacco, smoke, CO_2, odor) higher air flow rates are required.

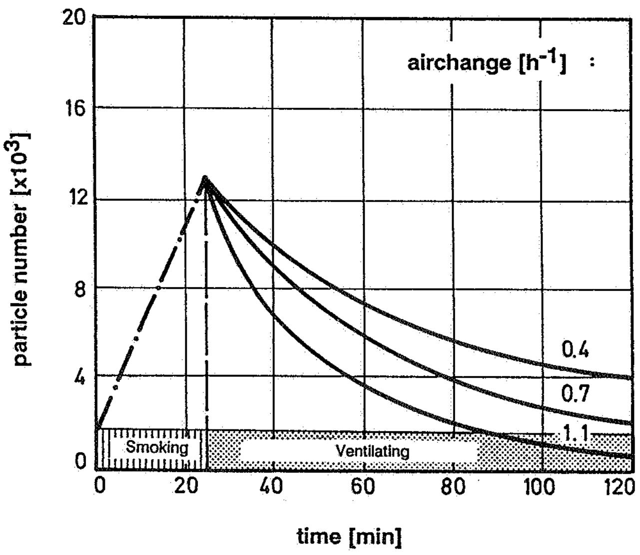


Fig. 3: Number of 1 μm-particles after cigarette smoking with different ventilating systems

6. Summarization

This research has given information about the ventilating performance of different ventilating systems for dwellings and the usability of air quality and humidity sensors for demand control. New questions refering to measurement and calculation techniques are seen:

- measurement methods for leakiness of joints
- determination of air flows between rooms
- distribution of supply air in rooms

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VENTILATION SYSTEM PERFORMANCE

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Paper 37

VENTILATION CHARACTERISTICS OF SELECTED TYPE OF BUILDINGS AND INDOOR CLIMATE

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SYNOPSIS

The paper presents results of ventilation characteristics of a lecture/seminar room obtained by various door-window opening combinations and positions, and the level of comfort and air quality resulted by the given window-ventilating modes. Applying statistical methods, formulae of air change rate for the test room under it´s normal operating condition i.e. when all window and external door are shut and when particular windows are opened is also presented and graphs in relation to dominancy factors such as wind and buoyancy effects, are given. It is found that, under it´s normal operating mode the natural supply of outdoor air is far too deficient for health and comfort purposes. Means of improving the thermal environment and indoor quality to meet the fresh air requirements as recommended by both the ASHRAE and CIBSE Guides are suggested by proper selection of window opening patterns. Assessements of thermal comfort using a thermal comfort meter and occupancy odour or freshness were also conducted.

1. Introduction

As buildings become better insulated the energy demand becomes more dependent upon ventilation losses. However, if the building is too tight this may result in a deficiency because the fresh air supply has to be sufficient to ensure a comfortable and healthy indoor climate. If this can be achieved using naturally ventilated buildings, then this is more energy and cost effective. In the United Kingdom, almost all school buildings are ventilated by natural means, with windows as the usual means for fresh air supply as well as providing daylighting. However, there is not much information with regards to how users of buildings should operate natural ventilation openings, as means of ventilation control. Many users complaints could be reduced if they ´understood´ their building better, but designers have to bear in mind that the occupants are in the building to do a particular job, and operating the building is not necessarily their concern. Occupants´ involvement must only be for simple and common-sense operations; particularly that relating to health, safety and comfort. Clear instructions are essential, even if the mechanisms are obvious, such as which windows to open and by how much. Ways of giving visual representation also need developing.

The essence of this study is to investigate ventilation characteristics and hence the effects of natural ventilation via windows, doors and spaces linked by corridor or passage-ways. This particular lay-out is common in school buildings. Once schools are built the floor plan is rarely changed so that the effects of natural ventilation via windows and doors linked by corridor and entrances takes on particular characteristics for each location. This effect plus the occupancy behaviour and the effects of building orientation, meteorological and surrounding terrains can account for significant differences in ventilation characteristics, and hence the indoor climate and energy needs between otherwise similar school buildings.

2. Experimental Set-up and Instrumentations

The test room is a classroom known as the Synoptic Laboratory in the Department of Meteorology; measuring 10.9 m x 11.0 m and floor to ceiling height 3.05m. The experimentation plan layout and set-up is as shown in Figure 1. Twenty-four thermistor probes are employed to measure the room and outside air temperatures (dry and wet-bulb) and globe temperatures. Three vane cup anemometers are installed on the roof of the test room, at a height 0.5 m and located on the north, south and east walls respectively. Wind direction is measured by a wind vane located on the centre of the roof, at a height 1.95 m. Global solar radiation is measured by a solarimeter also located on the roof of the test room. Resistance switches are fitted to six doors; two on the main entrance doors and the rest on interior doors. A Bruel&Kjaer Type 1212 Comfort Meter is used to evaluate the thermal environmental quality using the predicted mean vote (pmv) scale as an index of comfort. A Dantec Type 54N10 thermal anemometer was used to measure the room air velocity at several points 0.9 m above the floor. An infra-red gas analyser is used for monitoring the carbon dioxide level in the test room and also for measuring the tracer gas decay during ventilation rate measurement. Output from the infra red analyser is fed into a flat-bed chart recorder.

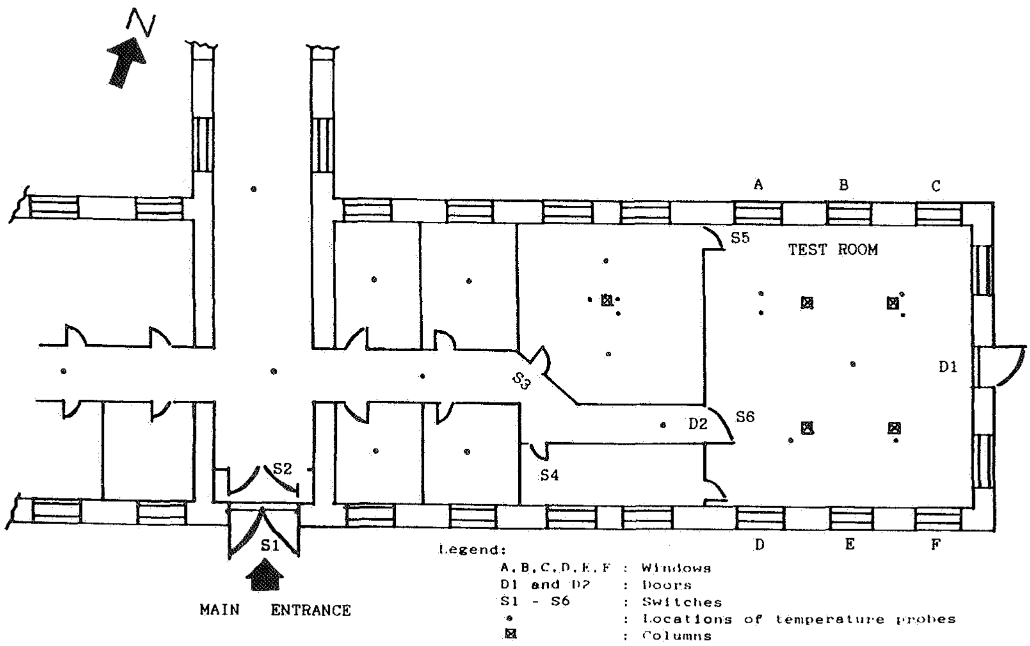

Fig.1. The Test Room and Experimental Set-Up

Data are logged on a BBC B microcomputer via a 32-channel multiplexer feeding into a 13-bit bipolar analogue converter (ADC) connected to the 1 MHz bus port of the computer. The ADC has a basic range of + 2.048 V, and its resolution was 0.5 mV. The program scanned all or a sub-set of the channels, applied calibration factors to the measured voltages, calculated means and standard deviations of all variables over a specified period, and saved the results to disc. A time interval of 1 hour was used for the purpose of this study. Another 32-channel chart recorder and data logger, Molytek Type 2702 was used to record the outputs from the thermal comfort meter as well as the temperatures of the test room.

3.Method

A series of ventilation rate measurements of the test room are conducted using the concentration decay method with carbon dioxide (CO_2) as the tracer gas. Air change rates of particular combinations of windows and their opening positions are measured to investigate the variation in the ventilation characteristics, representing basically the two common types of ventilation i.e. single-sided or cross-ventilation. The air flow pattern for several window opening combinations is also established by measuring the velocity and turbulence at several points in the room and hence, the discomfort i.e. the percentage dissatisfied (PD) due to draught was evaluated. A total of 72 identical experiments were conducted without opening any windows or doors and a statistical model of ventilation rate derived using a multiple regression technique. This was the normal mode of usage of the room during the heating season. Carbon dioxide levels were also monitored during the occupancy periods as a means of assessing the indoor air quality in order to establish a rational basis for ventilation of spaces where body odour is the major pollutant.

A subjective survey of occupants was undertaken; the occupants were asked to vote on their sensations of the thermal environment including impressions of odour based on the ASHRAE seven-point thermal sensation scale. The first group of 46 subjects includes 10 females and 36 males; age range was between 22 - 40 years (comprising of Asian, African, British and South American Nationalities). They were the normal users of the room, participated in the test which was conducted under the normal operating mode of the room, in mid-March 1990. The second subjective test on 18 occupants; age between 22 - 40 years and of various nationalities, was conducted when two diagonally opposite windows (D & E) were opened one-half opening position, at the end of July 1990. The sample of questionnaires for the subjective analysis is as shown in Appendix 2.

4. Results and Discussion

4.1. Statistical Model of Ventilation

A statistical model of ventilation rate of the test room under it´s normal operating mode i.e. when all window are closed and internal door (D2) occasionally opened is given by the equation:

$$N = 0.092 - (0.37x10^{-3} \Delta T) + (0.26x10^{-3} \theta) + 0.019\ V^2 \qquad (1)$$

where:

N: air change rate (h)

ΔT: difference between indoor and outdoor temperature (K)

θ: wind direction, taken from true North (degree)

V: local wind speed (m/s)

The governing equation for the case when windows are opened is:

$$N = 1.44 + 4.65A + 0.59\ V^2 - 0.013\ \theta - 0.023\ \Delta T \qquad (2)$$

where:

A: area of window opening (m^2)

From the test of significance (see the t-test as shown in Tables 1a and 1b) the air change rate is much influenced by the wind factors i.e. speed and direction, and window opening area rather than the temperature difference.

Table 1a. Regression Coefficients

Y-variate : N

Parameter	Estimate	SE	t
Constant	0.09221	0.04904	1.9
θ	0.00026	0.00015	-0.1
ΔT	0.00037	0.00392	1.8
V^2	0.01907	0.00366	5.2

Table 1b. Regression Coefficients

Y-variate: N

Parameter	Estimate	SE	t
Constant	1.4421	1.5068	1.0
θ	-0.01325	0.00549	-2.4
ΔT	-0.02342	0.12011	-0.2
V^2	0.59414	0.28436	2.1
A	4.6471	1.2691	3.7

The plot of air change rate for a range of wind speed and temperature difference ΔT = 0, 7.5, 15.0 and 20.0 K and wind direction θ = 90 and 270 deg. using both equations (1) and (2) is shown in Figures 2a and 2b.

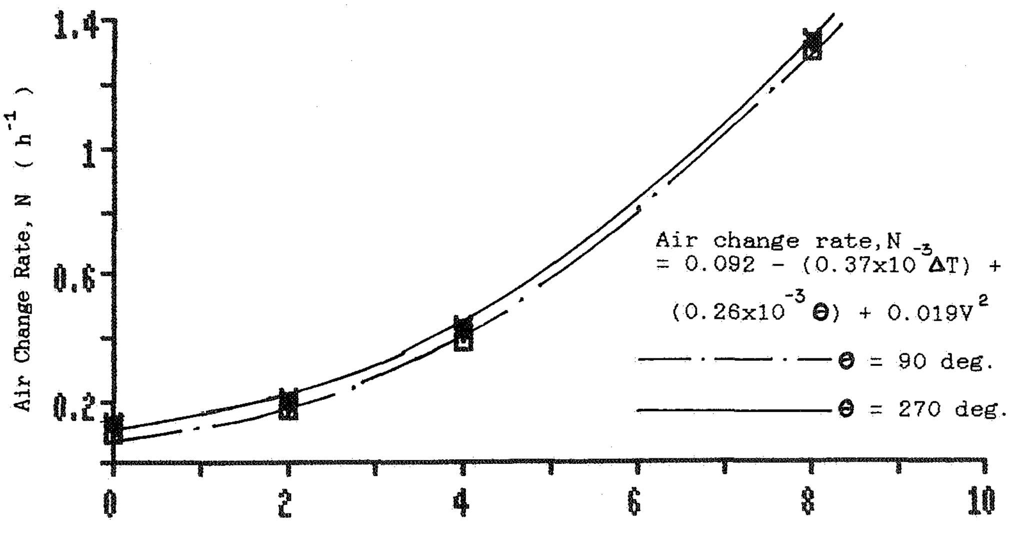

Fig.2a. Predicted Variation of Test Room Air Change Rate with Wind Speed and Temperature Difference when no Window is Open

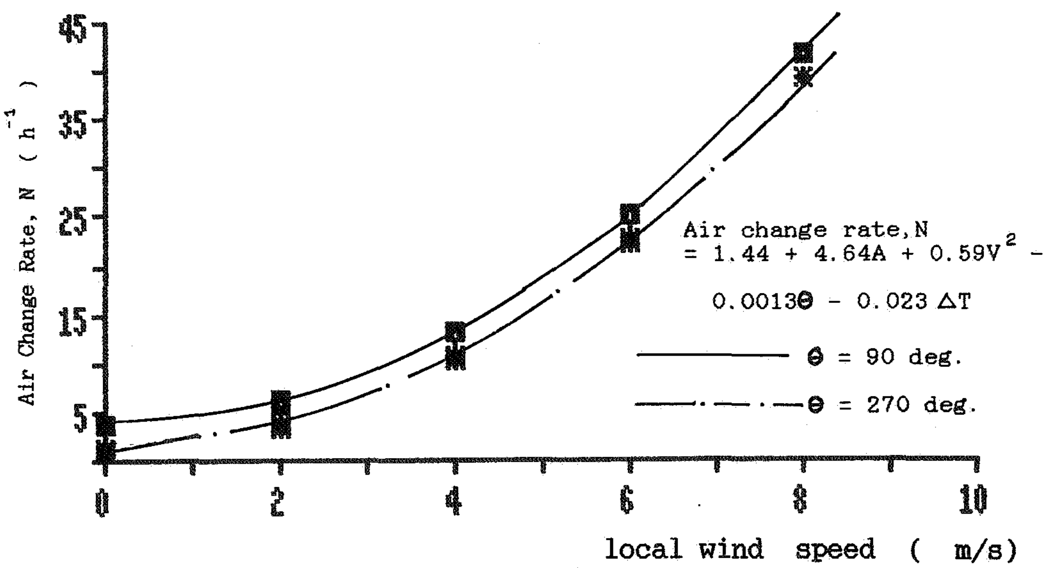

Fig.2b. Predicted Variation of Test Room Air Change Rate with Wind Speed and Temperature Difference when Windows are One-half Open

From Figure 2b, for a local wind speed of 1.8 -2.0 m/s and opening area equivalent to two windows opened by one-half opening position, would produce the required fresh air supply as recommended in the ASHRAE and CIBSE guides.

4.2. Ventilation Characteristics

4.2.1. Under Normal Occupancy Mode

The mean value of air change rate attainable during a normal operating mode of the room i.e. when all windows are shut and internal door (D2) is ocassionally opened, is 0.2 per hour (19.5 l/s). This far too low for comfort and health purposes. As recommended by the ASHREA (7) and CIBSE (8) Guides for the control of body odour, the minimum air change rate should be in the order of 2.75 (265.9 l/s) and 2.6 (252.8 l/s) per hour respectively (Refer to Appendix 1 for calculation).

4.2.2. Ventilation Characteristics Due To Window and Door Opening

The results for the various window combinations and opening modes are summarised in Fig.3a and those due to door opening in Fig.3b.

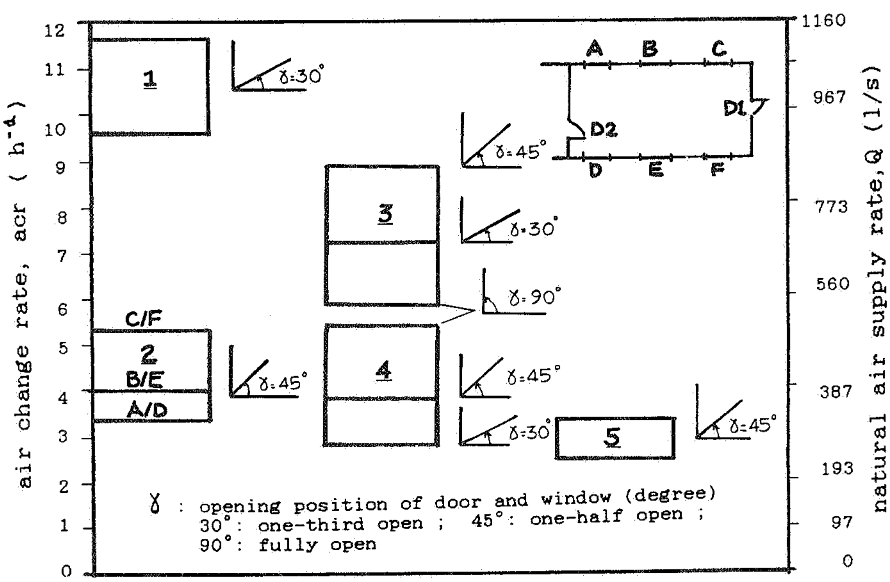

Fig.3a. Ventilation Characteristics of Windows
Local wind speed: 0.8 - 2.2 m/s; direction: 60 - 296 deg.
Legend:
1: All windows on north and south walls are opened
2: Directly opposite windows (A&D,B&E,C&F) are opened
3: All windows (A,B,C) on north wall are opened
4: All windows (c,D,E) On south wall are opened
5: Diagonally opposite windows (C&D) are opened
Note: Q = (3600 x acr (h^{-1})) / Room Volume (m^3)

Air change rate between 2.4 - 11.6 per hour (232 - 1121 l/s) is attainable for the various window combinations and opening positions for local wind speed between 0.84 - 2.22 m/s, direction 60 - 296 deg. from true North. The larger value is obtained when all windows on the north and south walls are opened by one-third of full opening position simultaneously and the main door of the test room is fully open (shown as item 1, Fig.3a). This to some extent includes the effect of the corridor which linked the room to the main enrtance. These results show that cross-ventilation i.e. by opening windows directly opposite each other, produces higher ventilation rates.

Another form of cross-ventilation i.e. by opening two diagonally opposite windows (C & D) produced moderate air change rates of 2.4 - 3.3 per hour (271 - 851 l/s), in the mid-opening position, at wind speed of 0.75 - 1.75 m/s, direction 147 - 285 deg. from North (shown as item 5, Fig.3a). Single-sided ventilation is studied by opening all windows on either the north or south wall and results as shown by items 3 and 4, Fig. 3a.

This mode of ventilation produced air changes between 2.8 - 8.8 per hour (271 - 851 l/s), wind speed range of 1.13 - 2.23 m/s and direction 60 - 296 deg.

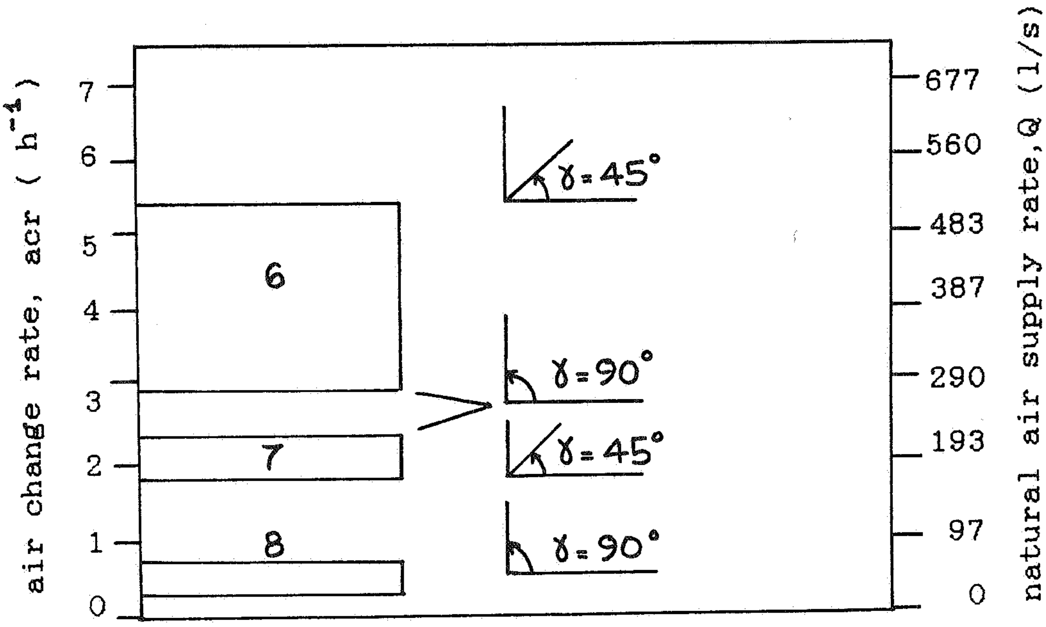

Fig.3b. Ventilation Characteristics of Doors
Local wind speed: 0.8 - 2.2 m/s; direction: 215 - 274 deg.
Legend:
6: Both external and internal (D1&D2) doors are opened
7: Only external door (D1) is opened
8: Only internal door (D2) is opened
Note: Q = 3600 x acr (h^{-1}) / Room Volume (m^3)

The ventilation characteristic due to door opening varied from 0.3 to 5.4 air changes per hour (77 - 522 l/s) for a wind speed range of 1.40 - 2.44 m/s and direction 215 - 274 deg. The larger value was due to opening both the external and internal doors (D1 & D2) in the mid-opening position while the lower value was the result of opening only the internal door in it´s full position. The results are designated as items 6,7 & 8 in Fig.3b. It is important to note from these results the effect of building orientantion vis a vis to the window location and prevailing wind. The generally larger values of air changes when a window or a door was opened in the mid-position rather than in the full-position indicates that chanelling of air through an opening depend on the position of the window relative to the prevailing wind.

4.3.Thermal Environment and Indoor Air Quality

The results of thermal environment and air quality monitoring and subjective studies of the room under it´s normal occupancy mode i.e. without opening any windows or doors are summarised in Table 2.

According to Fanger and Berg Munch(1), and Cain et al(5), the indoor air quality of the test room is not acceptable i.e. more than 20% of the occupants are likely to complain or feel dissatisfied due to odour. However from Muhaxheri´s(3) findings the indoor air of the test room is acceptable.(Note: Values of odour intensity and percentage dissatisfied are extraploted from the results of these researchers using the value of carbon dioxide level monitored during the period of occupancy). For further explaintion refer to Reference (8).

Table 2.Odour Intensity (I) and Percentage Dissatisfied (PD) Votes of the test room

Odour Intensity (I)	Percentage Dissatisfied (PD)	Source
1.9	28 %	Fanger & Berg Munch (1)
1.8, 2.5, 3.1	n.a.	Yaglou (2)
1.5, 2.8	n.a.	Narasaki (4)
1.5	12 %	Muhaxheri (3)
n.a.	22 %	Cain et al (5)
n.a	6 - 12 %	Comfort Meter

The subjective assessment by 46 occupants of the thermal environment and indoor air quality of the room under its´ normal occupancy mode i.e. when all windows and door are closed, produced a total of 73.9 % dissatisfied due to feeling slightly warm to hot, and slightly cool. Taking "moderate odour" i.e. value of 2 on the odour sensation scale and"slightly stuffy" i.e value of 1 on the freshness-stale scale, as the base line values in assessing the acceptance of odour and fresh-stale quality of the indoor air, reveals that 24% and 52.2% are dissatisfied with the indoor environment due to body odour and staleness of the air respectively. This corresponds to 21.7% non-acceptance of the indoor air quality of the room if occupants are to be exposed to the indoor condition during their daily working period. The results are as shown in column 1 Tables 3a,b,c,d. When two diagonally opposite windows (D & E) were opened by one-half opening position, 55.6% of occupants were dissatisfied due to feeling slightly cool to warm, 17.6% are dissatisfied with the indoor environment due to the presence of body odour and staleness of the air respectively. On the overall acceptance, 88.2% voted that the indoor air quality was acceptable. The results are given in column 2 in Tables 3a,b,c,d. This particular combination of window opening was tested because it is the appropriate combination which could meet the recommended air changes i.e. 2.6 - 3.0 per hour, for comfort and odour control. The results of the thermal environment and air quality assessments for both situations, i.e. when the room is operated with and without opening windows, as a comparison, are also shown in Tables 3a,b,c and d.

Table 3.a:Comparison of Results of Subjective Analysis of Thermal Environment

Thermal Sensation Scale	With-out Opening Window		With Opening Window	
	No. of Occupants (%)	Dissatisfied (%)	No. of Occupants (%)	Dissatisfied (%)
3(Hot)	3 (6.5)		0 (0)	
2(Warm)	12 (26.1)		1 (5.6)	5.6
1(Slightly Warm)	14 (30.4)	(63.0)	5 (27.8)	27.8
0(Neutral)	12 (26.1)		8 (44.4)	
-1(Slightly Cool)	5 (10.9)	(10.9)	4 (22.2)	22.2
-2(Cool)	0 (0)		0 (0)	
-3(Cold)	0 (0)		0 (0)	
Total Dissatisfied		**73.9**		**55.6**

Table 3.b:Comparison of Results of Indoor Air Odour Perception

Odour Sensation Scale	With-out Opening Window		With Opening Window	
	No. of Occupants (%)	Dissatisfied (%)	No. of Occupants (%)	Dissatisfied (%)
0(No Odour)	18 (39.0)		8 (47.1)	
1(Slight)	17 (37.0)		6 (35.3)	
2(Moderate)	8 (17.4)	17.4	3 (17.6)	17.6
3(Strong)	1 (2.2)	2.2	0 (0)	
4(V.Strong)	1 (2.2)	2.2	0 (0)	
5(Over Powering)	1 (2.2)	2.2	0 (0)	
Total Dissatisfied		**24.0%**		**17.6**

Table 3.c:Comparison of Results of Indoor Air Freshness-Stale Perception

Freshness Stale Scale	With-out Opening Window		With Opening Window	
	No. of Occupants (%)	Dissatisfied (%)	No. of Occupants (%)	Dissatisfied (%)
-2(Very Fresh)	0 (0)		0 (0)	
-1(Fresh)	4 (8.7)		6 (35.3)	
0(Neutral)	18 (39.1)		8 (47.6)	
1(Slightly Stuffy)	18 (39.1)	39.1	3 (17.6)	17.6
2(Stuffy)	6 (13.1)	13.1	0 (0)	
Total Dissatisfied		**52.2**		**17.6**

Table 3.d:Comparison of Results of Indoor Air Acceptance

Acceptance Scale	With-out Opening Window	With Opening Window
	No. of Occupants (%)	No. of Occupants(%)
0 (Acceptable)	36 (78.3)	15 (88.2)
1 (Not Acceptable)	10 (21.7)	2 (11.8)
Total Acceptance	**78.3**	**88.2**

Notes: The first subjective assessment was conducted under room temperature between 20.3 - 23.4 $^\circ$C, relative humidity between 80 - 85%, air change rate < 0.5 h^{-1}, indoor carbon dioxide level 116 - 1460 ppm above outdoor, local wind speed 3.6 m/s and direction 320 from North and predicted mean vote(pmv) = - 0.6.

The second subjective assessment was conducted under room temperature about 24.4 $^\circ$ C, relative humidity 58%, indoor carbon dioxide level 425 ppm above outdoor, local wind speed 1.90 m/s and direction 340 $^\circ$ from North and predicted mean vote(pmv) = 0.5

The possibility of discomfort due to draught caused by air turbulence is also investigated by calculating the percentage dissatisfied (PD) level at various points in the test room. The PD level as proposed by Madson et al (9) is given by the equation,

$$PD = (3.14 + 0.37 \times V \times Tu)(34 - Ta)(V - 0.05)^{0.62} \quad (3)$$

which is a function of air temperature Ta, mean air velocity V, and turbulence intensity Tu, and valid for:

20 C < Ta < 26   C
0.05 < V < 0.4 m/s
0   < Tu < 70   %

The PD levels were found to be less than 6% at all points in the room when windows C and D were opened in the mid-position as shown in Figure 4.

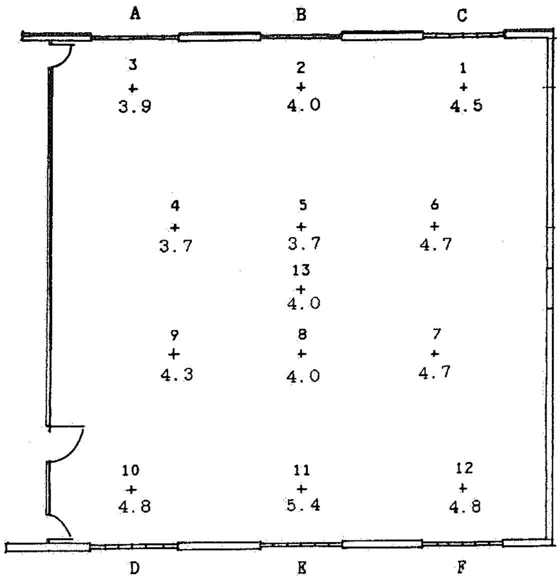

Fig.4. Percentage Dissatisfied (PD) Level due to Draught in Test Room when Two Diagonally Opposite Windows are Opened

Comparison of PD levels in the room as a result of several combination of window opening is shown in Table 4. In all cases the PD levels were less than 10%, indicating that there is no likely discomfort cause by draught i.e. air motion as a result of window ventilating.

Table 4: Percentage Dissatisfied (PD) Level due to Draught

	Window Opening Combinations and Position of Opening			
Point	A & D 1/2-Open	B,D,F 1/2-Open	B,D,F Fully-Open	D,E,F Fully-Open
1	4.6	4.8	3.3	4.8
2	4.0	5.6	3.7	5.6
3	3.9	4.8	3.5	4.8
4	3.7	3.9	3.4	3.9
5	3.7	4.1	3.9	4.1
6	4.7	4.9	3.3	4.9
7	4.7	6.2	3.4	6.2
8	4.0	4.9	3.7	4.9
9	4.3	4.2	4.4	4.2
10	4.8	5.4	3.6	5.4
11	5.4	5.5	3.4	5.1
12	4.8	6.2	4.5	6.2
13	4.0	4.8	3.8	4.8

4. Conclusions

This study shows that the ventilation characteristic of naturally ventilated spaces can be varied by applying different modes of window-door opening patterns. The information gathered would be useful for managing buildings during the warm season, where some means of controlling the indoor climate can be achieved by windows opening. If the result of air change rate measurements for the normal mode of operation of the test room during the heating season is to hold true for school classrooms (i.e. < 0.5 air change rate per hour), this is far too deficient for health and comfort purposes. This study also reveals the importance of building planning i.e. planning layout of building in order to enhance the useage of natural ventilation for both comfort and indoor air quality control.

5. Future Work

- An important aspect of comfort is room air movement due to various window types and operating modes; this will be studied next.

- Investigation will also be carried out on other buildings with distinctive plan lay-out and different window types and arrangements in order to investigate the ventilation characteristics of various window styles in relation to air movement control for comfort.

Acknowledgements

The authors would like to thank Dr.A.Ibbetson and technicians in the Department of Meteorology, at the University of Reading for their efforts in the laboratory and experimental set-up. Also to those who have willingly took part in the subjective tests.

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APPENDIX 1

Estimation of the mininum fresh air requirement for test room according to ASHRAE and CIBSE recommendations (revised)

Air change rate, N = (3600 x Q) / V per hour

where:

Q : Volume flow rate of air (m / s)

V : Volume of test room (m^3) (365.7)

ASHRAE recommendation : 8.0 l/s per person

CIBSE recommendation : 7.5 l/s per person

Normal occupancy of test room is 35 people

N (ASHRAE) = (3600 x 8 x 10)(35 x 10) / 365.7
= 2.75 per hour

N (CIBSE) = (3600 x 7.5 x 10)(35 x 10) / 365.7
= 2.60 per hour

APPENDIX 2

INDOOR CLIMATE INVESTIGATION: SUBJECTIVE SURVEY

Date:________ Day:________ Time:________

Building/Room: ____

Please answer the following questions by putting a tick at the appropriate boxes. Your answers will be treated confidentially.

Question 1. What clothing are you wearing ?

Shirt :	Long-sleeve	☐	Short-sleeve	☐
Trousers:	Thick material	☐	Light material	☐
Sweater/Cardigan:	V-neck	☐	Round-neck	☐
	Wool	☐	Others	☐
Skirt:	Wool	☐	Others	☐

Question 2. Sex: Male ☐ Female ☐

Question 3. Nationality : ______________________

Question 4. When did you last have your bath or shower ?

______ day/s ago.

Question 5. When did you last change your clothings ?

______ day/s ago.

Question 6. How do you feel about the temperature in this room ?

	Beginning of class	End of class
hot	☐	☐
warm	☐	☐
slightly warm	☐	☐
neutral	☐	☐
slightly cool	☐	☐
cool	☐	☐
cold	☐	☐

Question 7. How strong is the odour ?

	Beginning of class	End of class
no odour	☐	☐
slight odour	☐	☐
moderate odour	☐	☐
strong odour	☐	☐
very strong odour	☐	☐
over powering odour	☐	☐

Question 8. Imagine that during your daily work you would be exposed to the present odour. Would you judge the odour af the air as :

acceptable	☐	☐
not acceptable	☐	☐

Question 9. Do you think the air is fresh ?

very fresh	☐	☐
fresh	☐	☐
neutral	☐	☐
slightly stuffy	☐	☐
stuffy	☐	☐

Other Comments:

VENTILATION SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 38

VENTILATION SYSTEM AS AN AIR HEATING SYSTEM MEASURING RESULTS IN A RESIDENTIAL BUILDING.

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1. SYNOPSIS

This paper presents measuring results from experiments with integrated air heating and ventilation system in airtight well-insulated buildings in Stockholm (The Stockholm Project).
The experiments with air heating systems in the Stockholm Project has earlier been presented in a paper at the 9th AIVC conferance 1988 (2).

This paper presents further results from measuring and analyses of the indoor climate and temperature measurements. The results show that it is possible to get a good thermal comfort in winter without down draught protection below the windows and with air inlet at the inner wall of the room, even when the air flow is reduced to a level equal to the need of normal ventilation flow. This means that it is not necessary to mix the ventilation flow with return air from the apartment.

Comfort data has been collected during a period with outdoor temperatures of -10 to -20 dgrC.

The study has been carried out in cooperation with the evaluating group of the Stockholm Project at EHUB at the Royal Institute of Technology, Stockholm, Sweden.

2. INTRODUCTION

The Stockholm Project (1) is a large joint experimental research and demonstration project for evaluation of new energy saving technology in buildings. Primarily established products are used but in each of the six buildings one or more new methods of energy conservation is tested. The energy demand for heating and ventilation is considerably lower in these buildings compared to a larger group of buildings of similar types built during the same period in Stockholm. All the buildings are airtight and well insulated, some of them better than the Swedish Building Code require.

In two of the six buildings in the project the heating distribution is made as forced air heating system together with a mechanical supply air system.
In a paper presented at the 9th AIVC conferance (2) the in door climat in those buildings was compared to buildings where hydronic heating and mechanical ventilation is used.That study was based on interviews and thermal comfort measurements in the apartments during February 1988 from a period not colder then about -5 dgrC.

This paper presents further results from analyses of the indoor climat and thermal comfort based on measurements during December 1989 with outdoor temperatures of -10 to -20 dgrC.

The measurements are carried out in one of the two air heated buildings where two methods of air distribution are tested.

1: Exterior wall supply via floor level ventilators under windows.
2: Interior wall supply from overhead ventilators, thereby providing no direct downdraught protection beneath windows.

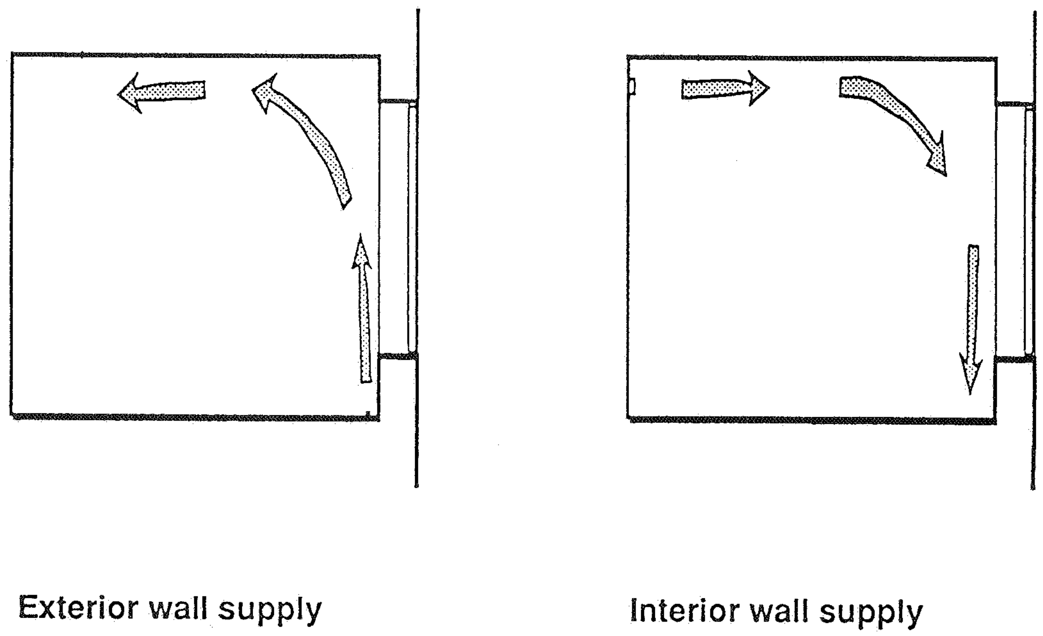

Fig.1. The two methods of air distribution.

The aim of this study was to :

1: Study the thermal comfort in the apartments and compare both types of air heating distribution, using detailed measurements of air mowment and temperature.

2: Give advise on further development and improvements of the air heating technic in combination with ventilation systems.

3. TECHNICAL DESCRIPTION OF THE BUILDING

The ten apartments in the building (3) are heated by forced warm air supplied by an air heating system. Each apartment has its own separate air heating unit, in which the incoming ventilation air, at a rate of 0.5 to 0.6 air changes per hour, is mixed with filtered recirculated air from the apartment to provide a total air flow rate equivalent to 1.3 air changes per hour (design data).

The ventilation supply air is preheated in a central ventilation unit with heatexchange between supply- and exhaust air.
The air temperature is controlled by thermostat to balance each apartments transmission losses.

In five apartments the supply air is distributed from exterior walls at floor level beneath the windows. In the other five apartments the air is supplied from ventilators placed at the interior walls of the room at ceiling level. Which means that there is no direct protection against cold downdraught, such as radiators beneath the windows.

Although during the measurement the total rate of air flow was only 0.7 to 0.8 air changes per hour. This was depending on a bad function of the central fan and wrongly adjusted dampers. This low air flow gave us a good opportunity to study if heating with warm air would function at a flow rate corresponding to a normal ventilation flow without extra recirculation air from the apartment.

If this work it will provide a much simpler instalation, excluding the recirculation unit in the apartments. There for no changes was made to increase the total air flow rate to 1.3 air changes per hour.

4. THERMAL COMFORT MEASUREMENTS

The study of the in door climate was made the 15th of December 1989 which was one day during a period with serval days of cold weather. The night before the 15th the outdoor temperature was below -20 dgrC and the outdoor temperature during the measurements was betwen -20 to -10 dgrC, most of the time around -15 dgrC.

The parameters studied were :
- Air speed close to the ventilators.
- Air speed and direction in the room.

- Air speed and direction at windows.
- Supply air temperatures.
- Air temperatures in occupant zone.
- Air temperatures at windows.

5. RESULTS

The results presents the measurements of air mowment and air temperature in some selected typical apartments in the building. The data from the measurements are also compared to the study made in 1988 (2).

5.1. Rooms with over head air supply on inteior wall

The study of the air movment in both livingrooms and bedrooms shows that with air supply at rooflevel there will be nearly no air velocity in the occupant zone (Fig 2 and 3). The warm air is flowing in a thin layer, 5-10cm, under the ceiling with a velocity of 0.35m/s to 0.15m/s.

The supply temperature is about 30 dgrC. After less than one meter the supply air has been mixed with the roomair so much that the temperature at ceiling level is not more than 2-3 dgrC higher than in the occupant zone.

There is a small downdrought velocity of 0.2-0.3m/s at the windowsurface (Fig 2) but it does not effect the occupant zone. The air velocity near the window in the occupant zone is less than 0.05m/s and the air temperature is more than 20 dgrC.

In one apartment, with a bad window construction with coldbridges and airleaking balconydoors (Fig 3), it was obvious that theese defects had a greater influence on the thermal comfort then the air heating system could have had.

The measurements results are also confirmed by the tenants who did not have any comfort problem during the past winters.

The results are very similar to the results from the previous study in February 1988 (2) when the air flow was hihger and the outdoor temperature was warmer.

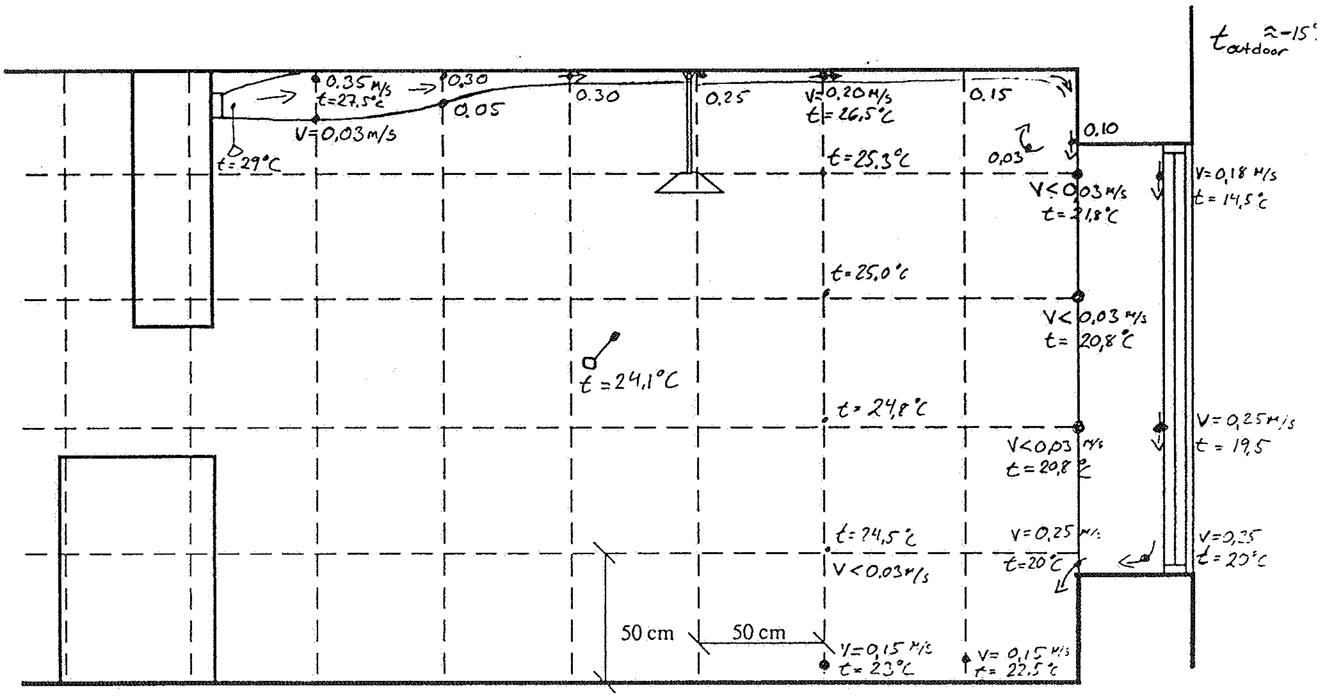

Fig 2. Section through a livingroom with interior wall supply. Air velocity (m/s) and air temperature (dgrC).

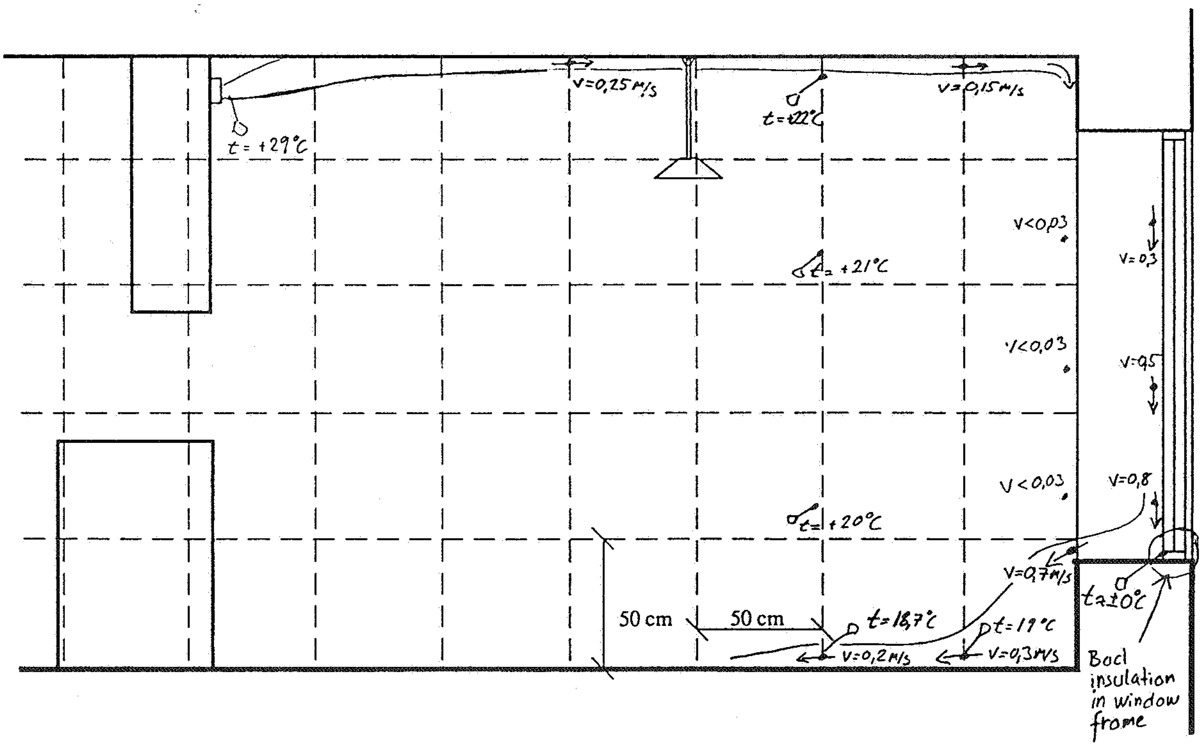

Fig 3. Section through a livingroom with interior wall supply and window with coldbridges. Air velocity (m/s) and air temperature (dgrC).

5.2 Rooms with air supply ventilators under windows

Air pattern in both livingrooms and bedrooms with ventilators at floor level under windows shows the same comfort problem as in the previous study in February 1988,(2). A small obstacle forces the air into an unwanted direction. Here a small protruding edge of the window-sill causes big draught problems in the occupant zone (Fig 4).

In one of the bedrooms we made an adjustment of the air stream direction with a piece of hard paper over the ventilator. Here by we could change the direction of the air stream and reduce the air velocity in the occupant zone (Fig 5). Instead we got a very high velocity, about 0.5-1.0m/s in front of the window.
In both cases no cold downdraught could be detected at the windows.

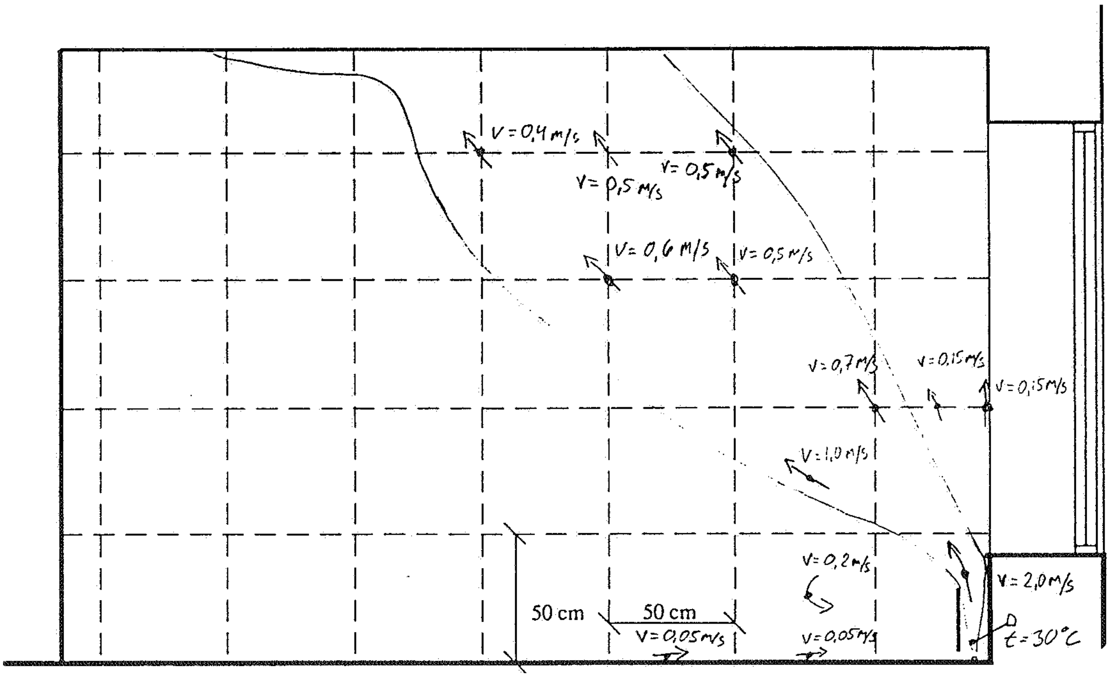

Fig 4. Section through a bedroom with exterior wall supply. Air velocity (m/s) and air temperature (dgrC).

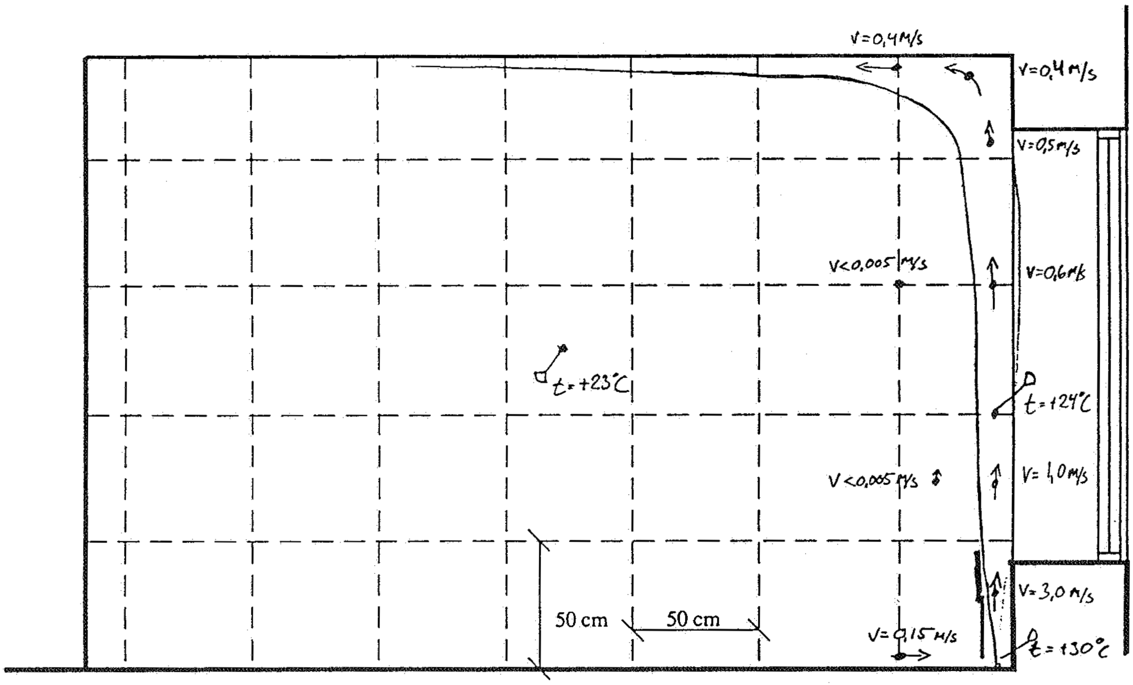

Fig 5. Section through a bedroom with exterior wall supply and with adjusted ventilator. Air velocity (m/s) and air temperature (dgrC).

6. DISCUSSION

One main result of the study is that even in very cold climat we can use air heating, with overhead air supply at interior wall without cold downdraught protection under the windows, with good thermal comfort.

An other important result is that the measurements also shows that we do not need more air flow then we use for normal good ventilation in an apartment.

This means that we can simplify the air heating system and make them without the air recirculation unit in each apartment.

Air leakages and coldbridges has a greater influence on the thermal comfort than the air heating system has, with variations that normaly occurs in the air distribution in systems with overhead air supply at interior walls.

Air distribution under windows can function if there is a careful design for both ventilators and for the details in their surroundings. But overhead air supply at interior wall seems to be a better solution for good thermal comfort.

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VENTILATION SYSTEM PERFORMANCE

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Paper 39

Air change rate and indoor air quality in bedrooms of well tightened residential buildings

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SYNOPSIS

Measurements of air change rates, carbon dioxide concentrations, room air temperatures and relative room air humidities in bedrooms of five well tightened dwellings were carried out in October 1989. With the results of the measurements and also based on simulation calculations, recommendations for an optimal window opening behaviour in bedrooms with the intention of saving energy and of providing sufficient indoor air quality were made. Air change rates between the outdoor air and the bedroom air, and between the bedroom air and the rest of the dwelling measured with the constant concentration method varied between 0.01 and 12.3 h^{-1}, corresponding to an air supply of 0.16 to 320 $m^3h^{-1}pers^{-1}$ depending on inhabitant behaviour. The measured carbon dioxide concentrations at the end of the night varied between 515 and 4286 ppm. The average room air temperatures ranged from 20.1 °C to 22.3 °C and the relative humidities from 42.4 % to 64.7 %.

1. INTRODUCTION

Several studies have been carried out in recent years regarding air change rates, indoor air quality and window opening habits also in bedrooms of well tightened residential buildings (1,2,3,4). These studies revealed that in well tightened residential buildings with natural ventilation the air change rate depends above all on user ventilation behaviour. Again the window opening habits depend on several external (outdoor climate, noise level etc.) and internal (type of room, habits etc.) factors. Depending on these factors, 60 - 80 % of the inhabitants sleep with their bedroom window closed. In these cases, the air change rate can be very low. The aim of this study was to get informations about the influence of user ventilation habits in bedrooms of well tightened dwellings on indoor climate generally and on indoor air quality specifically. Moreover, based on the results of the measurements and on simulation calculations, recommendations for optimal bedroom ventilation behaviour with the intention of saving energy <u>and</u> of providing sufficient indoor air quality should be made.

2. MATERIAL AND METHODS

The measurements were carried out in five dwellings of two specially energy saving designed apartment buildings in Préverenges near Lausanne in Switzerland. These buildings were constructed in the years 1986 - 1988. Since then, they have been the object of intensive studies by the Laboratory of Solar Energy and Building Physics (LESO) of the Swiss Federal Institute of Technology in Lausanne (5). The following investigations were carried out in October 1989:

- Continuous measurements of air change rate and relative humidity over 12 hours at night. Both were measured with CESAR (Compact Equipment for Survey of Air Renewal) of LESO (6). For these measurements the constant concentration method with N_2O as tracer gas was used. In this case the measured air change rate corresponds to the air exchange between the outdoor air and the bedroom air plus the air exchange between the bedroom air and the air of the rest of the dwelling.

- Continuous measurements of the concentration of carbon dioxide over 12 - 20 hours each night or day respectively. The measurements were carried out with a Leybold Binos 100 two chanel infrared gas analyzer.
- Continuous measurements of room air temperature and outdoor air temperature over 24 hours a day with radiation protected PT-100 resistance thermal detectors.
- Registration of the degree of bedroom window and bedroom door opening during the measurements with a questionnaire. Further the inhabitants were asked to leave the bedroom window and door closed for one night during the measurements, independent of their usual behaviour.
- In each of the five dwellings the measurements were carried out over two or three nights in two bedrooms (main and childrens bedroom). Totally 17 measurements could be carried out.
- Simulation calculations were made with a dilution equation (7), taking account of several different user behaviours.

3. RESULTS AND DISCUSSION

3.1. Air change rate and air supply in the bedrooms

Table 1 shows the ranges of the measured and time weighted mean values. Also the corresponding room volume and occupancy dependent air supplies for the measured bedrooms are shown. Only 14 measurement could be used because of an equipment defect.

window	door	Numb. of measurements	Air change rate h^{-1}	Mean ± Standard dev. h^{-1}	Air-supply $m^3h^{-1}pers^{-1}$	Mean ± Standard dev. $m^3h^{-1}pers^{-1}$
closed	closed	9	0.01 - 0.31	0.12 ± 0.1	0.16 - 9.3	3.1 ± 2.6
closed	open	1	0.5	-	10.5	-
open	closed	4	3.2 - 12.3*	9.4 ± 3.6	100 - 320*	193 ± 79

Table 1: Measured air change rates and air supplies (from external and internal air) dependent on different inhabitant behaviour (Measurement error: ± 15 %, * measurement error ± 30 %). Measuring time: 12 hours.

With closed bedroom windows and doors the measured air change rates were the lowest. In five cases the air change rate was even below 0.1 h^{-1}. With opened bedroom windows and closed doors the measured air change rates were very high. The corresponding air supplies were also very low with closed bedroom doors and windows and on the other hand very high with opened windows. These results coincide well with the results of measurements made in the same dwellings during the heating period 1988/89 (5).

3.2. Carbon dioxide concentrations in the bedrooms

Table 2 shows the measured carbon dioxide concentrations dependent on user behaviour and time of room occupation.

window	door	Numb.of measurements	Numb. of persons	CO_2 Median ppm	CO_2 Maximum ppm	CO_2 Increase ppm	Time of room occ. h:min	CO_2 Increase ppm/min
closed	closed	4	2	999-2934	1182-4286	789-3364	7:20-11:40	1.47-7.64
closed	closed	5	1	740-1352	828-2084	196-1399	9:45-13:00	0.25-2.39
closed	open	1	2	986	1318	413	9:00	0.76
ajar	closed	1	2	760	1210	835	9:30	1.46
ajar	closed	1	1	820	909	543	11:10	0.80
open	closed	2	2	519	615- 718	219- 331	8:00	0.45-0.69
open	closed	3	1	419- 542	515- 644	128- 272	7:30- 9:30	0.24-0.47

Table 2: Measured carbon dioxide concentrations (Median, maximum, increase during the time of room occupation and increase per time unit) dependent on user behaviour and time of room occupation (Measurement error: $\pm$ 5 %).

The highest concentrations of carbon dioxide at the end of the night were measured when the bedroom doors and windows where closed all night long. Opening the bedroom door with closed window resulted in a lower concentration at the end of the night. The lowest values were measured with opened windows. A good overview on the effect of different inhabitant behaviour on indoor air quality can be obtained by regarding the increase of carbon dioxide concentration during the night (table 2). Taking in account that the occupancy time of the bedroom during the night was different, the increase of carbon dioxide concentration is divided by the time of occupation (in minutes). The result represents the average increase of the carbon dioxide concentration per time unit (minutes).

Figure 1 shows the distribution of the measured average increase of the carbon dioxide concentrations per minute during the night dependent on different inhabitant behaviour.

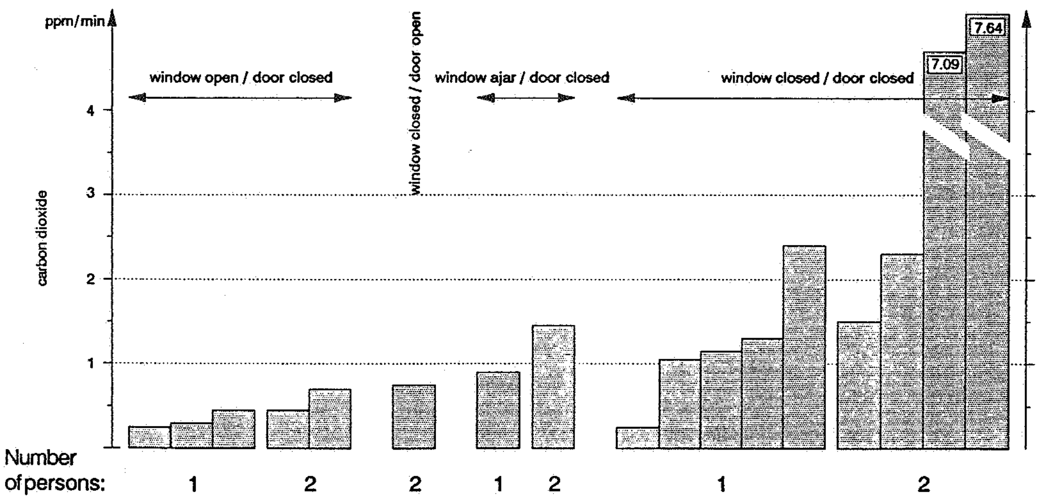


Figure 1: Distribution of the average increase of the carbon dioxide concentrations per minute during the night dependent on different inhabitant behaviour and occupancy.

Direct comparisons of the influence of the inhabitant behaviour could be made in four dwellings during two nights. During the first night the people behave as usual concerning window and door opening habits. During the second night people left windows and doors closed. Table 3 shows the results.

dwelling _room	window	door	Numb. of persons	CO2 Median ppm	CO2 Maximum ppm	CO2 Increase ppm	Time od room occ. Std:Min	CO2 Increase ppm/Min
X_1	closed	closed	2	2016	2910	1627	11:40	2.32
	closed	open	2	986	1318	413	9:00	0.76
W_1	closed	closed	1	1352	1814	1399	9:45	2.39
	open	closed	1	542	644	272	9:35	0.47
W_2	closed	closed	1	1024	1282	907	11:45	1.28
	ajar	closed	1	820	909	543	11:15	0.80
Z_1	closed	closed	2	999	1182	789	8:55	1.47
	ajar	closed	2	760	1210	835	9:30	1.46

Table 3: Measured carbon dioxide concentrations (median, maximum, increase during occupation time and increase per time unit) dependent on different user behaviour and different window and door positions (Error: ± 5).

In all four cases the median values of carbon dioxide concentration were higher with closed windows and doors than with the alternative behaviour i.e. opend or left ajar windows or opend doors. Opening the bedroom window resulted in a five times lower average increase of the concentration level per time unit than to let the window closed. For opening the bedroom door the average increase was three times lower and for leaving ajar the windows the increases were not or about one and a half times lower.

Figure 2 shows the difference between the carbon dioxide concentrations during the night, dependent on whether the window was closed or opened.

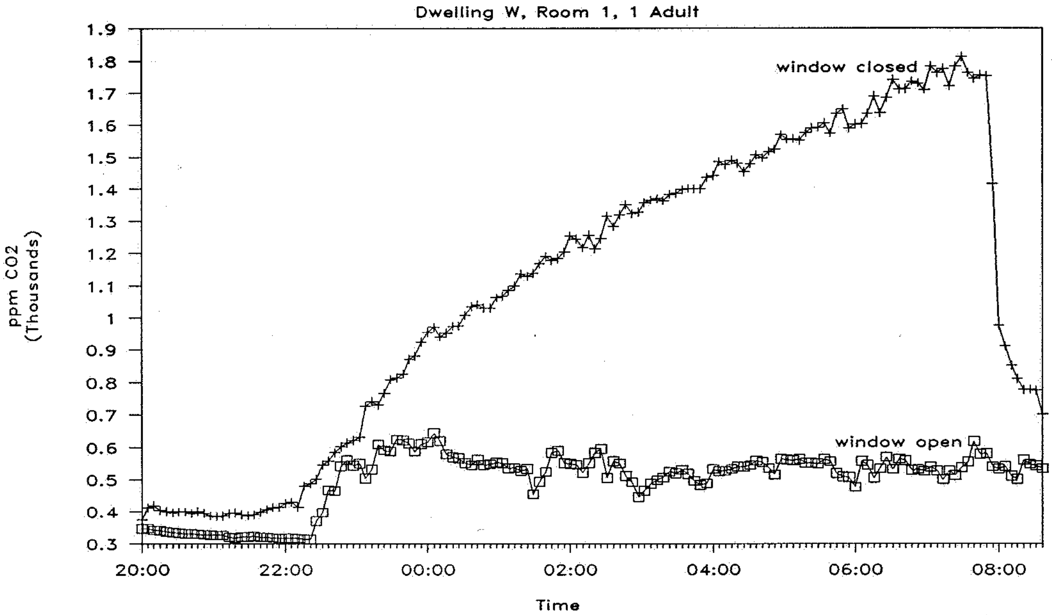


Figure 2: Concentration of carbon dioxide during the night in a bedroom occupied by one adult.

With closed window the concentration rose continuosly until at eight o'clock in the morning the bedroom door or window was opened. With opened window during the night the concentration rested at 500 - 600 ppm.

The following statements can be made regarding the average increase of carbon dioxide concentration per time unit (Tables 2 and 3, Figures 1 and 2):

The increase per time unit is the highest with closed windows and doors. Leaving the windows ajar is better than leaving the windows closed. Opening the bedroom door with closed windows is better than leaving the windows ajar. The best for a low carbon dioxide level in the bedroom is to open the windows widely.

3.3 Simulation calculations

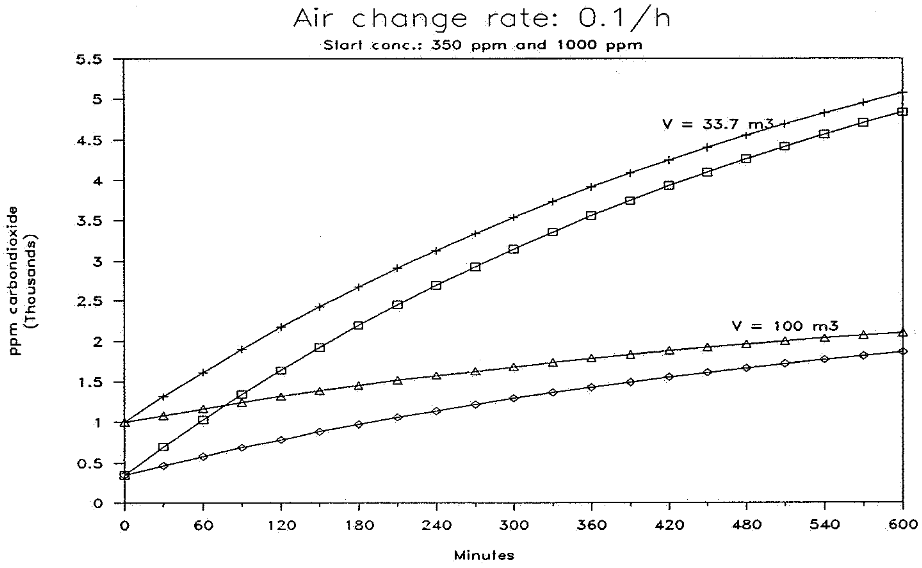
With the dilution equation for gases (7) several simulation calculations have been carried out. In a first phase the results of the measurements made in Préverenges were calculated over again to check the equation and the other input data. In a second phase simulation calculations were carried out. Together with the measurements this form the basis for making recommendations about the optimal ventilation behaviour. The following input data and assumptions were used to do the simulation calculations of the second phase: Occupation of the bedroom with two adult persons producing together 24 liters of carbon dioxide per hour. For the bedroom volume the average volume of the bedrooms in Préverenges (33.7 m^3) was taken. To simulate the opened bedroom door a dwelling volume of 100 m^3 was assumed (bedroom plus living room volume). The calculations have been made for an assumed time of room occupation of 10 hours. The carbon dioxide concentration of the incoming air was assumed to be 350 ppm. Perfect mixing of the room air was implied. The simulation calculations were made for air change rates of 0.01 h^{-1}, 0.1 h^{-1}, 0.5 h^{-1} and 1 h^{-1}. For each air change rate a calculation was made for four times changing the room volumes, and changing the concentration levels at the beginning of the calculation to simulate whether the room had been aired before or not. Figures 3 and 4 (next page) show the results of the simulation calculations for an air change rate of 0.1 h^{-1} and 0.5 h^{-1}.

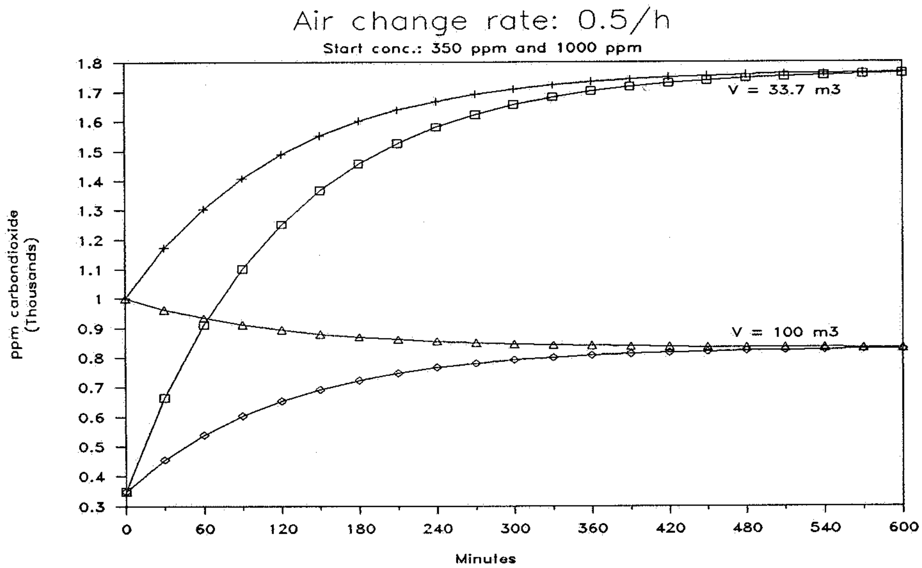
The following statements can be made from the results of the simulation calculations:

1) Bedroom door closed (V = 33.7 m^3):
a) Start concentration 350 ppm: With an air change rate of less than 0.5 h^{-1}, the carbon dioxide concentration passes rapidly over 1500 ppm. With an air change rate at about 0.5 h^{-1}, the carbon dioxide concentration reaches a concentration of about 1700 ppm after 10 hours sleeping.
b) Start concentration 1000 ppm: With an air change rate of less than 0.5 h^{-1}, the carbon dioxide concentration passes rapidly over 1500 ppm and is at all times about 600 ppm higher than if the start concentration would have been 350 ppm. With an air change rate at about 0.5 h^{-1}, the difference between the lapses of the two curves starting from 350 ppm and 1000 ppm gets relatively rapid less with time. After four hours the difference is less than 100 ppm.

2) Bedroom door opened (V = 100 m^3):
a) Start concentration 350 ppm: With an air change rate of about 0.1 h^{-1}, the carbon dioxide concentration reaches a level of slightly more than 1500 ppm. This is the effect of the larger room volume for dilution which is available. With an air change rate of about 0.5 h^{-1}, the concentration remains clearly below 1500 ppm.
b) Start concentration 1000 ppm: With an air change rate of about 0.1 h^{-1}, the carbon dioxide concentration will reach a level of about 1500 - 2000 ppm. With an air change rate of equal or more than 0.5 h^{-1}, the concentration of carbon dioxide is rarefied with time. The reason for this rarefaction is that through the higher room

volume the air supply (in $m^3h^{-1}pers^{-1}$) becomes higher although the air change rate (in h^{-1}) remains at the same level.





Figures 3/4: Influence of room volume and starting concentration to the time sequence of the carbon dioxide concentration with a room occupation of two persons and an air change rate of 0.1 h^{-1} and 0.5 h^{-1}.

3.4 Room air temperature and relative humidity

With closed windows the range of the measured bedroom air mean temperatures was between 20.1 °C and 22.3 °C. With opened windows the measured range was between 20.1 °C and 20.7 °C. At the same time the mean values of outdoor air temperature were from 6.0 °C to 9.8 °C. With closed windows the average values of room air temperatures were about 0.6 °C higher than with opened windows, al-

though the outdoor temperatures during the measurements with closed windows were about 0.6 °C lower on average.
The mean values of the measurements of the relative humidity were between 46.6 % and 64.7 % with closed and between 42.4 % and 45.3 % with opened windows. In both cases the mean values of outdoor air humidity were 76.5 %. With closed windows the average of the mean values of relative humidity were about 10 % higher than with opened windows. In two cases the measured relative humidity was higher than 60 %. The measured increases of the relative humidity during the night were between 1.6 % and 9.3 %.

4. CONCLUSIONS AND RECOMMENDATIONS

Regarded from the viewpoint of hygiene the air change rates in the bedrooms of the appartement buildings in Préverenges were too low when the windows and doors were closed. Above all, in the parents' bedroom, but also in the bedrooms occupied by only one person, the carbon dioxide concentration mostly exceeded 1000 - 1500 ppm. A carbon dioxide concentration of 1000 - 1500 ppm is regarded in several standards (8,9,10) as the upper limit for comfort. A minimum ventilation rate between 12 - 15 $m^3h^{-1}pers^{-1}$ (for 1500 ppm) and 20 - 30 $m^3h^{-1}pers^{-1}$ (for 1000 ppm) is required to stay below the comfort limit. Recent investigations showed that sleeping in well sealed rooms with closed windows, i.e. with low ventilation rates, can cause the typical sick building symptoms such as headache, sore throat, sleepiness, stuffiness etc. (carbon dioxide concentrations up to 3700 ppm (4)).

Regarded from the viewpoint of saving energy, the measured air change rates with opened window during the night would be much too high.

The measured relative room air humidities are at the upper limit if people are sleeping with closed windows and doors. But this is also dependent on the overall window opening habits of the inhabitants. With relative humidities at about 65 %, condensation on cold surfaces followed by mould growth and allergy reactions to the spores can occur. Moreover building damage can also occur.

Based on todays knowledge, the results of the measurements in Préverenges and on the simulation calculations, the following recommendations for optimal bedroom ventilation behaviour with the intention of saving energy and of providing sufficient indoor air quality can be made:
a) To guarantee an indoor air quality in bedrooms with a carbon dioxide concentration of less than 1500 ppm a sufficient outdoor air supply is necessary.
b) Outside of the heating period sufficient outdoor air supply can be obtained by opening the bedroom window during the night.
c) With an air change rate of 0.5 h^{-1}, an occupancy of two persons, a room volume at about 30 m^3 and closed windows and doors, the carbon dioxide concentration will not pass much over 1500 ppm after 10 hours.

d) From the energy point of view it is not very wise to leave the windows open during the night in the heating period. On the other hand, the outdoor air supply with closed windows is too small in well tightened buildings. Moreover the inhabitants may be forced to keep the bedroom windows closed during the night even outside of the heating period (outdoor noise, pollen etc.). In these limited cases it can be recommended that the bedroom door be left open during the night in well tightened buildings. If the whole dwelling is well aired beforehand (opening all windows for 5 minutes) and the air change rate of the whole dwelling is at least 0.1 h^{-1}, the carbon dioxide concentration will not, or only slightly, pass over 1500 ppm.

5. ACKNOWLEDGEMENTS

These investigations have been carried out within the scope of the research works called "Energy Relevant Air Flows in Buildings" financed by the Swiss Federal Fund for Energy Research.
Many thanks to the Laboratory of Solar Energy and Building Physics of the Swiss Federal Institute of Technology in Lausanne for placing the necessary infrastructure at our disposal and for the realisation of the measurements of air change rates and room air temperatures. Furthermore many thanks to the inhabitants of the dwellings in the appartement houses in Préverenges for their trouble in supporting such measurements.

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VENTILATION SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 40

REPORTING GUIDELINES FOR THE MEASUREMENT OF AIRFLOWS AND RELATED FACTORS IN BUILDINGS.

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SYNOPSIS

A set of reporting guidelines has been established. The guidelines take into account the need for data concerning airflow within buildings and air exchange between a building and its surroundings. They also deal with issues such as pollutant production and transport, thermal properties and measurements of buildings and comfort related issues. The comprehensive nature of these guidelines should enable a large amount of data to be accrued in a form suitable for computer modelling and validation work. The extensive use of computers in research has thus been considered and an application is under development to allow a research report to be entered directly into a relational database according to the framework of the guidelines. This will allow a research report to be directly accessible in the AIVC numerical database along side the data it refers to.

1 INTRODUCTION

One of the objectives of the Air Infiltration and Ventilation Centre is to encourage the collection and dissemination of air infiltration and airflow data, as well as energy use data arising from programmes of research and experimental investigation. As part of the Centre's present work program, reporting guidelines (Ref.14) have been produced to provide a common reference, for research workers wishing to plan experimental work or catalogue their experimental data.

The guidelines attempt to provide the necessary parameters for the calculation of airflows within buildings and the necessary input parameters for the operation and validation of airflow models, including those that include pollutant factors, and for the running and validation of thermal models of building energy use. These parameters vary with the application and this has been taken into account in the guidelines. It is hoped that this will make more complete, structured information available for entry into the AIVC numerical database (Ref.1) for subsequent analysis or mathematical model development. The basic reference for the guidelines is AIVC-TN 6, (Ref.2), aided by work in IEA Annex XX.

1.1 Guidelines Structure

The structure of the guidelines is purposely rather loose, to cater to the differing interests of the various investigators who will be using it. It has been made as comprehensive as possible but should not be regarded as exclusive. Correspondingly, the user should not feel impelled to fill in all the sections; however, if the results are entered in the order given, it immediately becomes apparent which items of information are present and which absent. The Guidelines are split into eleven parts. The first ten cover different catagories of data concerning the test site, measurements taken, and results gained. The eleventh part gives two examples of the use of the guidelines. Each part is split into a number of sections detailing different aspects of the data catagory concerned. Each of these sections is in turn split into a number of subject headings under which the user enters the data parameters requested.
Some parameters are more important or more relevant to a particular area of work than others, and this has been highlighted using the Applicability Coding which is described in the following section.

1.2 Applicability Code

The Applicability Code defines the importance and relevace of a data parameter to a particular area of work and is stated in parenthesis following each subject heading. If the code is listed by a part or section heading it is assumed to apply to all subjects within that part or section. The applicability code consists of two parts; a priority code defining the importance of a parameter, and an application code defining its main area of use. These codes should be used as an indicator only, all data collected is useful and should not be excluded simply on the basis of a low prority code or seemingly inappropriate application code.

The priority codes are as follows:

R = Required, I = Important, U = Useful.

Individual applications codes are:

- 1 = Parameters for airflow measurements and models; e.g. model validation, stock characterisation or design studies.
- 2 = Additional parameters for airflow models that include pollutant factors; e.g. indoor air quality work.
- 3 = Additional parameters for thermal measurement and models, e.g. model validation, energy use calculations. If the requirement is for cooling models only it will be noted as -3C;
- 4 = Additional parameters for comfort-related questions that involve: temperature stratification, room airflow, ventilation effectiveness, radiation, etc.

Thus an important parameter that would be applied to a flow model involving pollutants would have an applicability code of (I-2).

1.3 The Guidelines and Computers

These guidelines may be used directly for entering results and should also serve as a useful checklist to aid those who are initiating projects. Recognising that experimental data today relies heavily upon the computer for both collection and storage, the guidelines have taken this into account. Using a compiled dBase IV application being developed by the

Centre, textual data may be entered in the framework of the guidelines, stored on disk in dBASE IV format and a report produced. (See appendix I for an example) This application will be available from the centre on 3½" or 5¼" MS-DOS compatible disk so that data entry can be further streamlined and made more uniform. Any numerical data supplied to the centre should be backed up by a report produced according to these guidelines, preferably on disk.

1.4 Guidelines Format

The Parts, Sections and subject headings are printed on the right-hand pages of the guidelines technical note and are accompanied by explanatory notes on the left-hand pages. Points relevant to the use of the data, the required detail and usefulness of various measurement methods are raised in the notes. Also included are details of minimum standards of measurement where these have been indicated by past experience or are predicted to be future requirements.

The AIVC will be pleased to receive copies of the completed Guidelines along with associated numerical data on disk for inclusion in the Centre's numerical database. An up-to-date record of the contents of this database is maintained and copies of the selected portions of the data will be available through the nominated organisations in the participating countries.

2 USING THE GUIDELINES

The following notes give the ground rules for information entry into the Guidelines.

2.1 Units to aid comparison of results.

All data should be supplied in S.I. units. The use of other sets of units for local convenience, in addition, is optional. Where a quantity is in common use, e.g. air changes/hour, it may be noted for cross comparison.

2.2 Instrumentation and measurement accuracy

For all sections involving measurements, the instrument used and the accuracy of the measurements should be stated:

1) - with the data listing

2) - referenced to Part V of the guidelines "Measurements" (Ref.14)

3) - referenced to the AIVC Measurements Techniques Guide (Ref.4).

Date and time of measurements should be included in Julian format.

Attach relevant photographs, diagrams, tables, graphs to the final report, and give details of any disk based data files as requested in part VIII of the guidelines "Disk Data Files" (Ref.14).

2.3 Data entry in the Guidelines

Any entry for which the data supplied takes up much more space than has been allowed may be supplied as an appendix (not dBASE IV application), which should be clearly labeled with the same letter and number code as the corresponding entry on the form, and should be supplied in the same order.

Users are asked to supply as much of the general information as they can in addition to their own special interest - the more complete the data set, the more valuable it is.

Where blocks of information are common to several sections, these need only be cited once in the appropriate position and referred to as necessary.

2.4 Numerical Data on Disk

If information is on disk it should be in MS-DOS compatible format on 3½" or 5¼" disk. Please indicate file names contents and format (with short example if possible) and include a disk with the report.

2.5 Dissemination Policy

Any information supplied to the AIVC will be regarded as freely available to any bona-fide inquirer with the under standing that the origin of the data set will be acknowledged.

3 THE GUIDELINES CONTENTS

The Guidelines are split into eleven separate parts I to XI, each of which is designed to describe one specific area of a project. These parts are in turn split into various sections, which are further divided into subjects. The eleven major parts are as follows:

I. General Information

II. Test Site Description

III. Building Description
IV. Operation/Function of Building

V. Measurements

VI. Economic Factors

VII. Numerical/Computer Models

VIII. Disk Data Files

IX. General Remarks

XI. Examples of Reporting Guidelines Application.

Below is a brief description of each part its sections and their subjects.

3.1 General Information

General information requires details of the project leaders and workers, and how to contact them. It also requests the basic project information such as start/end dates and the number of manhours required. It also emphasises the purpose and approach of a project and the reasons for selecting a particular test site. These pieces of information are considered important to aid the planning of future projects.

3.2 Test Site Description

Test Site Description is split into two sections:

A. Geographic Information

B. Climatic Information (General)

Section A (Geographic Information) requests the information necessary to describe the location of the test site and its surroundings. The basic climatic information for the site i.e. yearly or ten yearly averages for temperature, wind speeds etc., should be entered in section B (Climatic Information).

3.3 Building Description

Building Description consists of seven sections designed to provide a comprehensive description of the building under test. This third part of the guidelines primarily concerns known or easily observable properties of the building, details of which do not require complex measurements.

The seven sections are:

A. General Description

B. Dimensions

C. Air Leakage

D. HVAC Systems

E. Pollutant Sources

F. Pressure Coefficients

G. Furniture, Interior Fittings

Section A (General Description) concerns the general properties of the building, its type, history and construction. Section B (Dimensions) requires a more detailed description of its physical dimensions, including a plan. Volumes of spaces and areas of components should be detailed here. Section C (Air Leakage) requests details of any known air leakage paths within the building fabric. These are any obvious and measurable cracks or openings. An estimate of the non-measurable leakage (the background leakage) is also requested. Section D (HVAC Systems) allows the building's HVAC systems to be described including the balance report and supplied outdoor air rate for ventilation systems. Section E (Pollutant sources) requires notes on any known interior (due to the building structure itself) or exterior sources of pollutants. Any known pressure coefficients for the building should be given in section F (Pressure Coefficients). If these have not been measured in a previous investigation, estimated values from general datasets (Ref. 5) should be quoted, along with any assumptions made. Section G (Furniture, Interior Fittings) requests details of any furniture or interior fittings within the building so their effect on pollutant concentrations, thermal properties and ventilation within the building can be assessed.

3.4 Operation/Function of the Building

Operation/Function of the Building is split into five sections and aims to gather information on the everyday life of the building. It is essentially concerned with qualitative data from questionnaires or other sources. The five sections are as follows:

A. Occupant Related Data

B. Special Ventilation Requirements

C. Control Values

D. Pollutant Sources/Sinks

E. Additional Heat Sources

Occupation times, window opening and pollution generation by users are among those topics covered in section A (Occupant Related Data). Section B (Special Ventilation Requirements) is included to highlight any unusual purpose provided ventilation, such as that required for clean rooms, hospitals, laboratories, museums, or art galleries. Section C (Control Values) requires the user specified settings for the HVAC equipment, such as airflow rates, temperatures and humidities. Pollutants produced by processes occurring within the building should be detailed in section D (Pollutant Sources/Sinks). Section E (Additional Heat Sources) should be used to quantify any additional heat sources in the building other than those which are purpose provided.

3.5 Measurements

Any diagrams or photographs of the measurement procedures should be attached to the report, as should any relevant tables or graphs. Any disk data files should be named, and their structure and format described in detail in the relevant portion of the guidelines.

Measurements is split into ten sections, which are as follows.

A. Pressurisation Measurements - Internal

B. Pressure Measurements - External

C. Interior Conditions

D. Weather On-Site

E. Weather Off-Site

F. Infiltration - Tracer Gas Methods

G. Inter-Room Airflow Rates

H. Pollutant Concentrations

I. Duct Flow Rates & Temperatures

J. Other

Section A (Pressurisation Measurements - Internal) should detail any pressurisation measurements carried out on the building or its components. Measurements at several pressures and for several wind directions are recommended (Ref. 4). External or wind tunnel measurements of wind pressure coefficients for the building should be described in section B (Pressure Measurements - External) using references 3, 6 and 7 for guidance. Section C (Interior Conditions) covers measured internal temperatures, humidities, airflows (non-tracer gas methods) and pollutant concentrations. Section D (Weather On-site) allows the entry of full on-site climate conditions, including wind, temperature, humidity, radiation, and precipitation parameters. If full on-site weather conditions are not available any gaps should, if possible, be filled by entering data on off-site conditions into section E (Weather Off-site). On-site data must however, always be considered more valuable than off-site data. Section F (Infiltration: Tracer Gas Methods) requires details of any Tracer gas methods used to measure infiltration. This would be coupled in a full investigation with information in section G (Inter-room Airflow Rates) on flow rates between rooms. Measured pollutant concentrations should be detailed in section H (Pollutant Concentrations) these may vary significantly with time depending on the sources, the exterior/interior conditions of the test site, the HVAC systems within the building and building usage patterns. Known duct flow rates and temperatures should be detailed in section I (Duct Flow Rates & Temperatures). These can have a significant impact on heat and pollutant transfer as well as the overall ventilation of individual spaces within a building. Any other measurements carried out on the building or test space, such as Infra Red Thermography or Energy Consumption, should be detailed in section J (Other), which should be repeated for as many different types of measurement as were taken.

Sections A, B, C, F, G, H, I, and J are split into five main subject headings. These are as follows:

1) Technique employed.
2) Equipment used.
3) Calibration procedures/results.
4) Results
5) Comments

These are designed to provide a comprehensive reporting structure for any measurements, with specific emphasis on data quality being provided by subjects 2 and 3 on the instrumentation used and any calibration techniques used. Any diagrams, tables, graphs or photographs of the calibration/measurement procedures should be attached to the report. Any disk data files concerning the calibration of the equipment should be supplied with the report and should be detailed according to part VIII (Disk Data Files) of the guidelines.

3.6 Economic Factors

Economic Factors is concerned with the economics of various ventilation systems, including the resultant health effects on occupants. It is split into three sections:

A. Retrofitting Measures

B. Ventilation Energy & Health

C. Other Factors

Section A (Retrofitting Measures) allows the effects of retrofitting measures to be assessed details of the measures taken and the energy consumption before and after retrofit are required. Such measures as energy signatures (Ref.8), or normalised annual consumption (NAC) should be used for such comparisons (Ref.9). The relationship between ventilation, energy and health is covered in section B (Ventilation Energy & Health). These should include specific issues such as mould growth and such general issues as sick building syndrome. Any other economic factors affecting the building's energy usage or ventilation rate should be detailed in section C (Other Factors).

3.7 Numerical/Computer Models

Numerical/Computer Models concerns Computer models (Ref 10) and theoretical work not directly involving measurements, and is split into two sections:

A. Type of Model/Correlations

B. Other Theoretical Work of Interest

Section A (Type of Model/Correlations) requires a detailed description of the model in question, the assumptions and algorithms used, its speed and size. It also requires a description of the input data required and the eventual output of the program. Any comparisons carried out using actual data to validate the model are considered very important and should be described in detail. Section B (Other Theoretical Work of Interest) concerns any new algorithms which have been developed independently or from the measurement work described in part V.

3.8 Disk Data Files

Disk Data Files has two sections:

A. Measurements

B. Numerical/Computer Models

Section A (Measurements) concerns any disk based data files associated with the work detailed in part V. of the guidelines. Section B (Numerical/Computer Models) concerns the input/output files required by the computer models detailed in part VII of the guidelines.

Each section is split into four principle subject headings:

1) Nomenclature
2) File Names/Contents
3) File Formats
4) Comments

Nomenclature requires complete details of the structure of the disk file names and any variables used in the files. File names/contents should contain a list of all the files required or output along with a brief description of their contents. File formats should detail the exact structure of the files listed sufficient for any programmer to read the files and make use of the data contained within.

3.9 General Remarks

General Remarks allows any points not included in the preceding parts to be described. It also gives the report writer a chance to detail any recommendations or conclusions arising from the studies detailed in the rest of the guidelines.

3.10 Examples of the Guidelines Application

This final part of the guidelines gives two examples of how they can be implemented in practice. The first is a report on a Canadian HUDAC low energy house (Ref. 13). This was generated using the Guidelines Report Generator (a dBASE IV application) being developed at the Centre. The second is a report on the Swiss LESO laboratory, a three storey fully instrumented working office building (Refs. 11 & 12). The first of these examples is reproduced in Appendix I of this paper.

4.0 THE DBASE IV APPLICATION PACKAGE

To complement these guidelines a dBASE IV application is being developed. This consists of a relational database with a menu driven front end. This allows the user to enter data into the database via the framework of the reporting guidelines. Having entered the data the user can produce a printed report in the format of the guidelines (See appendix I). The program begins with a title screen and copyright message. This is replaced with the main bar menu, whose options are described below.

4.1 Name/Select Report

The first option of the main menu 'Name/Select Report' allows the user to select whether to start a new report or edit an old one, it also gives the opportunity to delete reports.

4.2 Enter/Edit Text

The second option of the menu 'Enter/Edit Text" produces an extensive series of popup menus, which the user can use to select the part or parts of the guidelines into which they wish to enter data. The selected parts of the guidelines are then displayed as one or more on screen forms. Each form shows the part, the section and the subject under which data is being entered. Notes on the required data, as shown on the left hand page of the guidelines, are displayed at the bottom of the screen.

4.3 Print Report

The third option 'Print Report' gives the user a number of options. These allow the user to print a report of all the guideline subjects (including part and section headings), or to print out the same report including the subject notes. Once data has been entered into a report the user can also print out a report of the parts, sections and subject containing data, or a complete report of all headings and any entered data. Appendix I is an example of the former; only printing those headings which contain data.

4.4 Exit

The fourth and final option of the main menu is exit this will return the user to DOS or whatever program he started the application from. It will also save any changes the use has made to the database, unless instructed otherwise.

5.0 CONCLUSIONS

A new set of reporting guidelines have been established. These take into account the need for data concerning airflow within buildings and air exchange between a building and its surroundings. They also deal with issues such as pollutant production and transport, thermal properties and measurements of buildings and comfort related issues. The comprehensive nature of these guidelines should enable a large amount of data to be accrued in a form suitable for computer modelling and validation work. The extensive use of computers in research has thus been considered and an application is under development to allow a research report to be entered directly into a relational database according to the framework of the guidelines. This will allow a research report to be directly accessible in the AIVC numerical database along side the data it refers to.

APPENDIX I

This example of the use of the guidelines was originally outlined in reference 13. It is presented here as a printout from a prototype of the dBASE IV application being developed at the centre. Relevant diagrams, photographs, and graphs are attached at the end of the report and referenced in the text.

GENERAL INFORMATION

Report date

1980

Principal Researcher

C.Y. Shaw

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Other Researchers

G.T. Tamura

Project Title

The Mark XI Energy Reserch Project.

Air tightness and air infiltration measurements.

Project Purpose

To measure energy cosumption and factors which affect it, including infiltration.

Project start/end dates

1978/9

Building Selection

The Division of Building Research of the National Research Council of Canada and the Housing and Urban Develepement Association were participating in a joint programme to study energy conservation in four detached two-sorey houses. One of the

GENERAL INFORMATION

Building Selection

houses H1 was built to a construction standard similar to houses in the same area. The other three were built with added insulation and a specially applied polyethylene vapour barrier to improve the air tightness of the house envelope.

References

1. Quiroutte, R.L.
   The Mark XI Energy Research Project:
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   Building Res. Note No. 131

2. Shaw, C.Y. and Tamura, G.T.
   Mark XI Energy Research Project:
   Airtightness and Air Infiltration Measurements.
   Building Res. Note No. 162, June 1980.

Comments

Tests to measure the air tightness were carried out on four houses. Air infiltration was measured for the standard house (H1) and the upgraded house with heat pump (H4). Because the building envelope of the standard and upgraded houses differs primarily in degree of airtightness, a comparison of simultaneously obtained infiltration data should show whether or not there is a correlation between infiltration and air tightness.

TEST SITE DESCRIPTION

Geographic Information

Location

Fortune Drive
Orleans
Ontario
Canada
(5km east of Ottawa)
See Fig. 1

Terrain/Site Plan

Flat with low buildings (houses)
See Fig. 1 for site plan.
Shielding is moderate.
Buildings within 2 house heights + 2.5m earth berm.

Building Orientation

Front facade points 24°W of North

Climatic Information

Meteorological Station

On site

Comments

Detailed measurements taken on site.

BUILDING DESCRIPTION

General Description

Building type

Single detached houses.
2 storey, 3 bedroom, 1½ bathroom and basement, attached garage.
See Fig. 2.

History

Built to Ontario Building Code 1975 by Talback Construction of Ottawa. Construction began on 6th July 1977, essentially completed by end of December 1977.

Construction

Standard House (see Fig. 2)

Wood frame construction. 2 x 4 stud walls, 2 x 8 wood joists, wood trusses 24" oc.
Cast-in-place concrete foundations, 8" walls.
Wall insulation: Glass fibre, paper backed, R12.
Ceiling insulation: Glass fibre, paper backed, R20
Basement insulation,glass fibre, paper bached,R7, inside, extending 2ft below grade.
Windows: Double glazed, wood frame (sliding and double hung).
Exterior doors: Metal insulated, ~R6, no storm door.
Sliding Horizontal: Alum. 8" ivory white.
Roof: Asphalt shingles 210 lbs.
Brick on front of house only, one storey garage.
Soffits continually vented, A6.
Facia: Aluminium.

Upgraded house.(See Figs. 3 & 4)

As Standard, except:

6" walls, 2x4 studs + 2x2 horizontal strapping inside.
Wall insulation: Glass fibre, friction fit, R12+R7, 4mil polyethylene vapour barrier throughout.
Ceiling insulation: Glass fibre friction fit, R20 + R12
Exterior sheathing: 1" fibreboard, thermal value, R3
Basement insulation: Closed cell polystyrene 1½", R7.5, outside extending to footing.
Windows: triple glazed, wood frame, casement, awning,

Exterior doors, metal insulated, with storm door, ~R7.5.

Comments

Estimated annual heating consumption:

BUILDING DESCRIPTION

General Description

Comments

Standard - 20212 kW.h
Upgraded - 15125 kW.h (conventional heating)
Upgraded   7560 kW.h (with solar heating)
Upgraded - 9980 kW.h (with heat pump)

Dimensions

Plan

See Figs. 2 & 4.

Elevation

See Figs. 2 & 4.

Total volume

Including basement 386 cubic metres.

Internal floor area

Gross: 1249 ft^2/118m^2

Ceiling height

Ceiling area: 673 ft^2/63.7m^2

Facade (wall) area

Above grade: 1525 ft^2/144.4m^2

Foundation wall: 891 ft^2/84.4m^2

Windows

Area: 164 ft^2/15.5m^2

External Doors/Hatches

Area: 44 ft^2/4.2m^2

Rooms

See Figs. 2 & 4.

Attic/Cellar/Crawlspace

Gross basement enclosure area: 1437ft^2/136m^2

Internal Walls/Partitions

See Figs. 2 & 4.

Comments

See reference 1 for more details.

Air Leakage

Windows

Length of sash crack Standard - 42.85m
Upgraded - 67.59m

Frame/wall leakage - Negligible
Window leakage see figure 8.

Chimneys, flues

No chimney.

Comments

Wood frame enclosure area: 227.7m^2.
Gross basement enclosure area: 136m^2

The area of the building envelope is defined as the area of the exterior walls above grade lus that of the ceiling of the upper floor.

HVAC Systems

Type of system

Standard House.

Heating system:	Forced air electric furnace 15 kW.
Design heating load:	46400 Btu/h - 13.6 kW

Upgraded House.

Heating system:	Forced air electric furnace 10 kW.
Design heating load:	13 755 Btu/h - 10.186 kW

Solar House.

Heating system:	Solar, air to air, with forced air distribution.
Suplementary heating:	Forced air electric furnace 10 kW.
Design heating load:	13 755 Btu/h - 10.186 kW

Heat Pump House.

Heating system:	Heat pump, air to air, with forced air distribution.
Suplementary heating:	Forced air electric furnace 10 kW.
Design heating load:	13 755 Btu/h - 10.186 kW

Balance report

Air flows measured in forced airducts by orifice plate - results not given.

Comments

See reference 1 for more details.

OPERATION/FUNCTION OF BUILDING

Occupant Related Data

Occupation times/numbers

Unoccupied during tests, but furnished. Will be let to families in future and monitoring will continue.

Window Opening

Windows instrumented to detect opening, but not used in this experiment.

Door opening

To be instrumented, but no results given.

Additional Heat Sources

Electrical energy to each room, also to appliances - monitored, results not given.

MEASUREMENTS

Internal pressurisation

Technique employed

See Fig. 5.
A centrifugal fan with a capacity of 380 l/s was placed in the living room of each house. The discharge side of the fan was connected by a 10 cm diameter duct to an outside window, which was replaced by a plywood panel. The flow rate of the fan was adjusted manually with a damper and was measured with a liminar flow element.
The air leakage rates through windows and doors of the heat pump house were also obtained by comparing the overall air leakage rates taken before and after the particular components were sealed with plastic sheets.

Equipment used

MERIAM LFE Element: accuracy of 5% of measured value.

Results

See Figs. 6,7 & 8.

Using $Q=CA(\ P)^n$
Q in l/s, C in $l/(s.m^2.(Pa)^n)$, P(Pa)
(A = 227.8 for both)

Standard - C=0.11 n=0.71)) whole house

Upgraded - C=0.075 n=0.71)

Comments

See reference 2 for full results and discussion.

Wind pressure & Wind Tunnels

Technique employed

The exterior walls and ceilings were fitted with pressure taps, but no measurements were made in this series.
Presure differences have been measured at four different levels in calm weather, wind speed < 1m/s, to find the neutral plane - results not given.

Interior Conditions

Temperature (dry bulb)

Measured by thermocouple (See tables 1 & 2)

Relative humidity

Monitored - results not given.

Other

Moisture in the building fabric was also monitored - no results given.

Weather - on site

Instrumentation

Measured at 10m to rear of house at 18m above ground.

Wind speed

See tables 1 & 2.

Wind direction

By octant - see tables 1 & 2.

Dry bulb temperature

See tables 1 & 2.

Infiltration

Measurement technique

Used tracer gas decay method. Tracer gas was CO_2 produced by placing pieces of dry ice on a hot plate in the living room. After a pre-determined amount of CO_2 gas was generated, the remaining dry ice was taken out of the house. After allowing sufficient time for the tracer gas to mix with the air inside the house, using the forced-air circulation system, the CO_2 concentration was measured periodically by sampling from the return air duct of the forced-air system. during the tests, a sample of air was drawn alternately from the return air duct of each test house using 0.63cm polyethylene tubing, and was

MEASUREMENTS

Infiltration

Measurement technique

analysed using an infrared gas analyser. An automatic system was used to take air samples and measure the CO_2 concentrations to avoid introducing additional CO_2 into the houses by the presence of research personnel.

Equipment used

Infrared analyser - accuracy 1% of full scale.

Results

See Figs. 9,10,11 & 12.
See tables 1 & 2.

Autumn, Winter and Spring measurements were made simultaneously on both houses. In Summer the standard house was occupied so only the results from the upgraded house were available.

Comments

See reference 2 for full results and discussion.

ECONOMIC FACTORS

Ventilation rate effects

See Fig. 13.

See reference 2 for full results and discussion.

Other factors

Heat Loss Analysis indicates ventilation heat loss of 4.415kW (32.1%) for the standard house.

See reference 2 for full results and discussion.

NUMERICAL/COMPUTER MODELS

Other Theoretical Work

See Fig. 14.

See reference 2 for full details and discussion.

GENERAL REMARKS

Air leakage rates were measured in the four energy-conservation research houses using the fan-pressurisation method. It was found that the air leakage of the standard house was about 50 per cent higher than that of the upgraded houses. The high leakage of the solar house could probably be attributed to the additional air leakage through the ductwork and solar collector of the aor to air solar heating system. There was no detectable air leakage through joints around windows and doors. The air leakage through the windows of the heat-pump house, which were tested as installed was about 50% lower than the maximum value permitted by ASHRAE 90-75 for new building design.

Air infiltration rates were measured simultaneously in the standard and heat-pump houses using the tracer gas method with CO2 as a tracer gas. It was found that for wind speeds lower than 3.5m/s, the air-infiltration rate can be expressed in terms of inside-outside temperature difference by an equation similar to the air flow equation with the same exponent. The ratio of the infiltration rates of the two houses is approximately equal to the ratio of the flow coefficients, which indicates that there is a correlation between infiltration and air leakage as measured by fan pressurisation tests. The significance of inside-outside temperature is reduced as wind speed increases.

Air infiltration, on average, accounted for about 20% of the total energy purchased for the standard and heat pump houses in the 1978-1979 heating season.

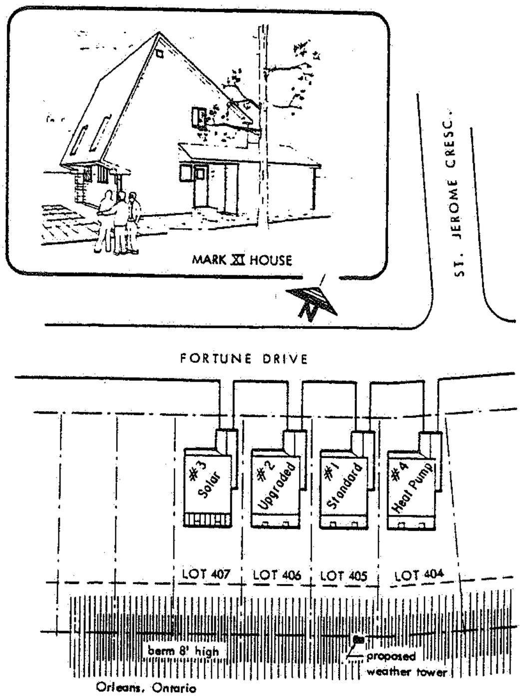

FIGURE 1

SITE PLAN - MARK XI PROJECT

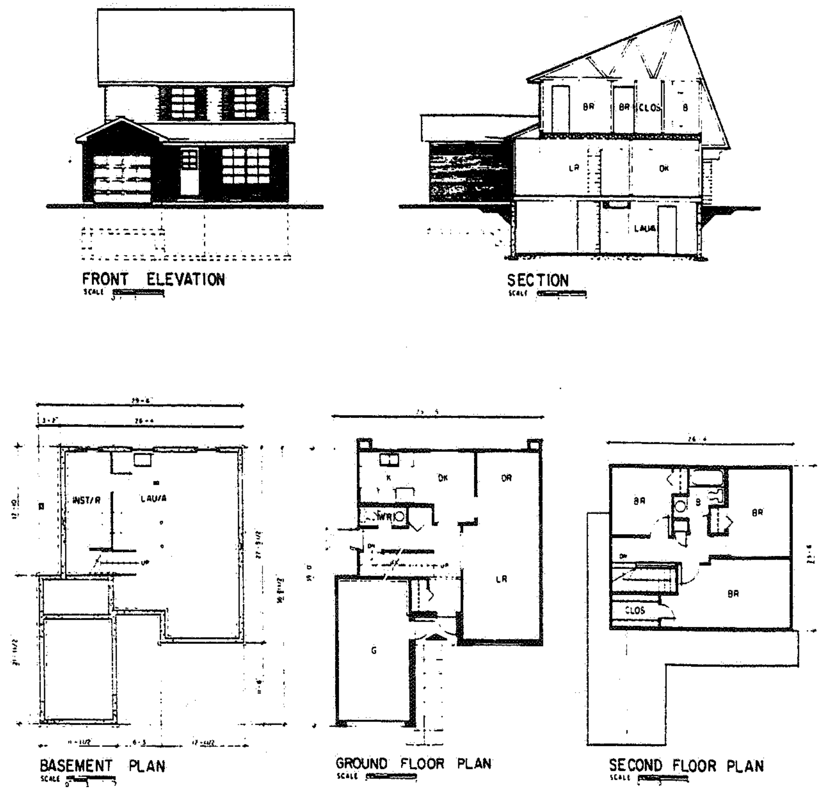

FIGURE 2

HOUSE NO. 1 - STANDARD CONSTRUCTION

(TYPICAL ARCHITECTURAL DESIGN OF ALL 4 HOUSES)

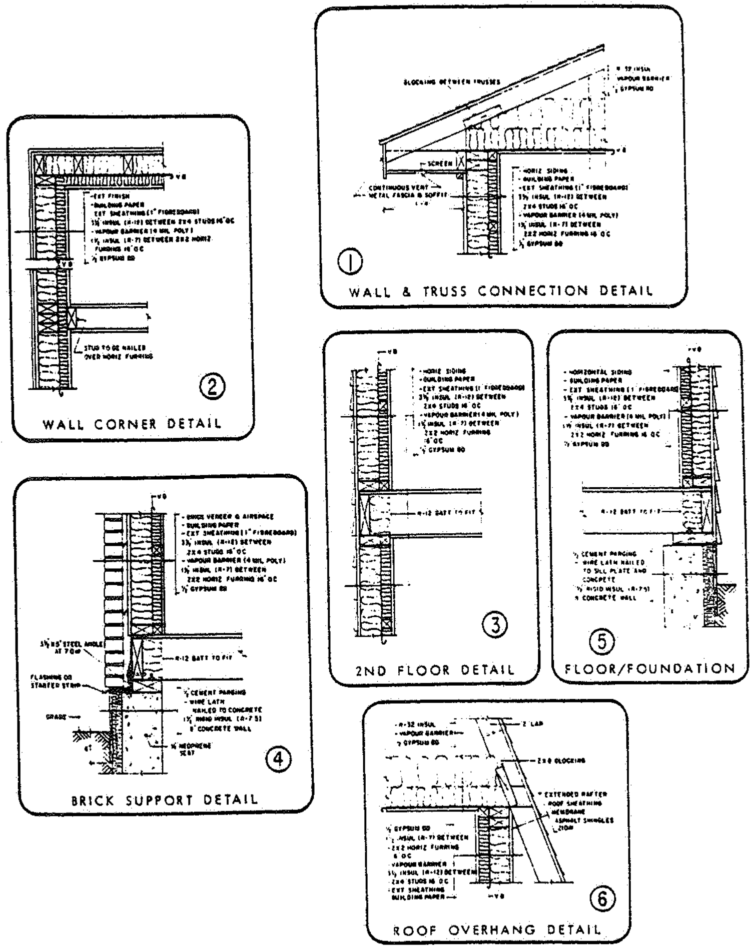

FIGURE 3

HOUSE NO. 2 - UPGRADED CONSTRUCTION

(CONSTRUCTION DETAILS FOR HOUSES 2, 3 & 4)

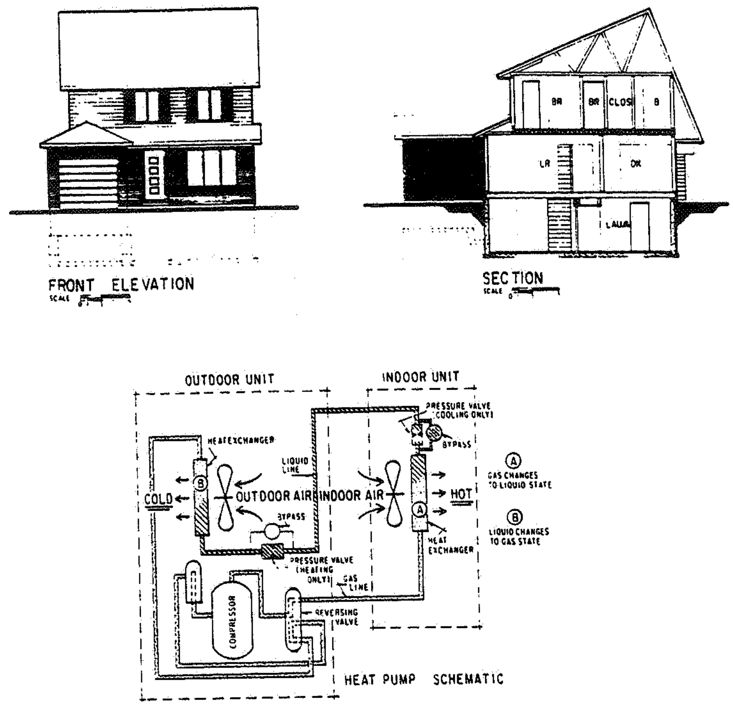

FIGURE 4

HOUSE NO. 4 - UPGRADED + HEAT PUMP

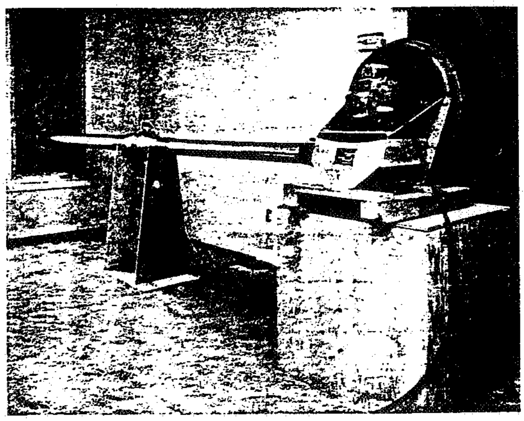

FIGURE 5 Equipment for fan-pressurization test

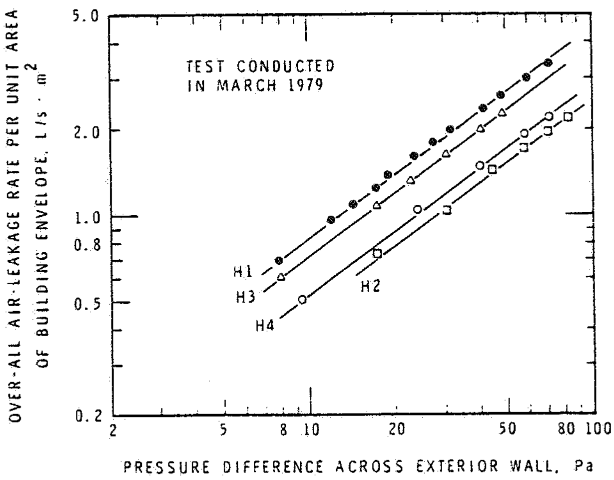

FIGURE 6

OVER-ALL AIR-LEAKAGE RATE FOR THE FOUR ENERGY-CONSERVATION RESEARCH HOUSES

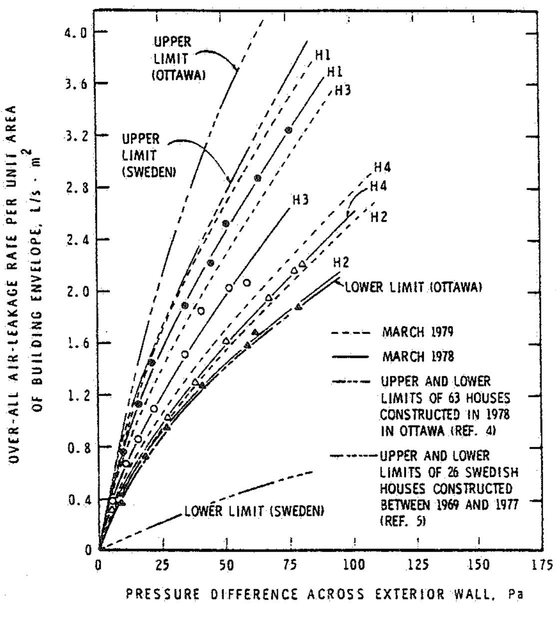

FIGURE 7

COMPARISON OF AIR-LEAKAGE RATE OF THE FOUR HOUSES AND OTHERS

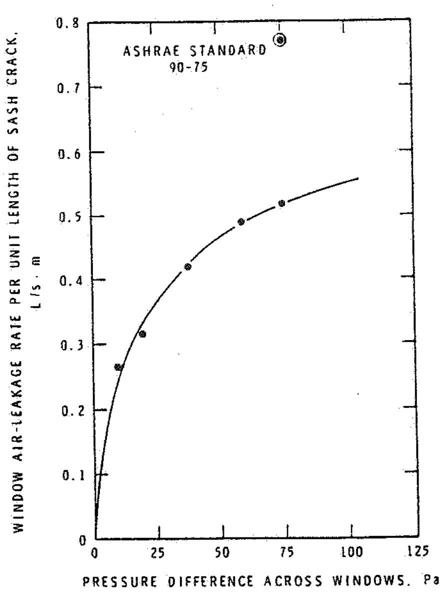

FIGURE 8

WINDOW AIR-LEAKAGE RATE OF THE UPGRADED HEAT-PUMP HOUSE

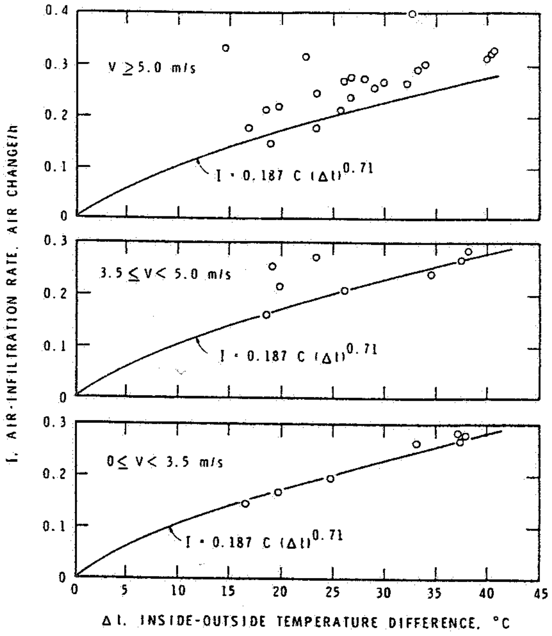

FIGURE 9

AIR-INFILTRATION RATE VS INSIDE-OUTSIDE TEMPERATURE DIFFERENCE FOR HOUSE H1

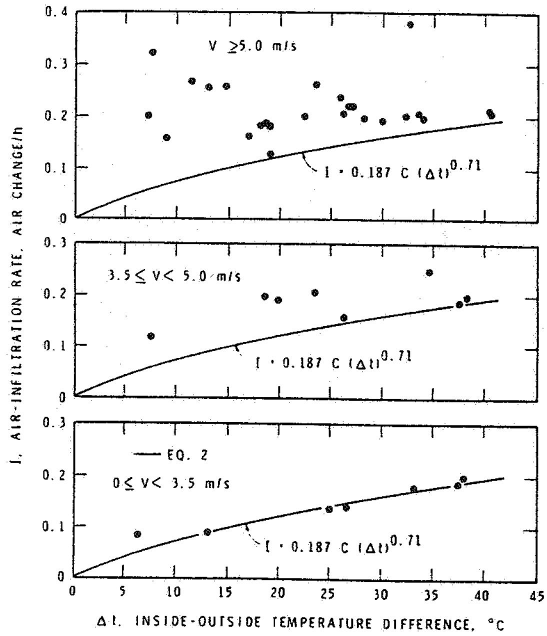

FIGURE 10

AIR-INFILTRATION RATE VS INSIDE-OUTSIDE TEMPERATURE DIFFERENCE FOR HOUSE H4

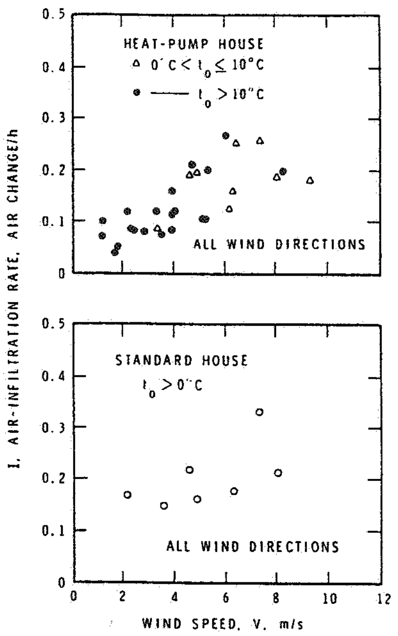

FIGURE 11

AIR-INFILTRATION RATE VS WIND SPEED

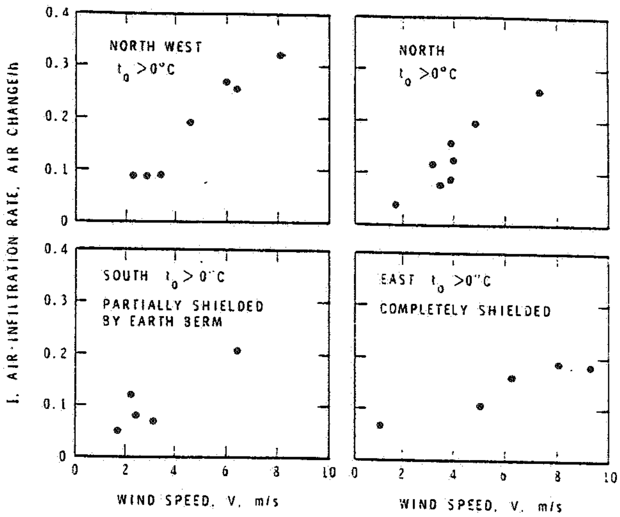

FIGURE 12

AIR-INFILTRATION RATE AT VARIOUS WIND SPEEDS AND WIND DIRECTIONS FOR HEAT-PUMP HOUSE

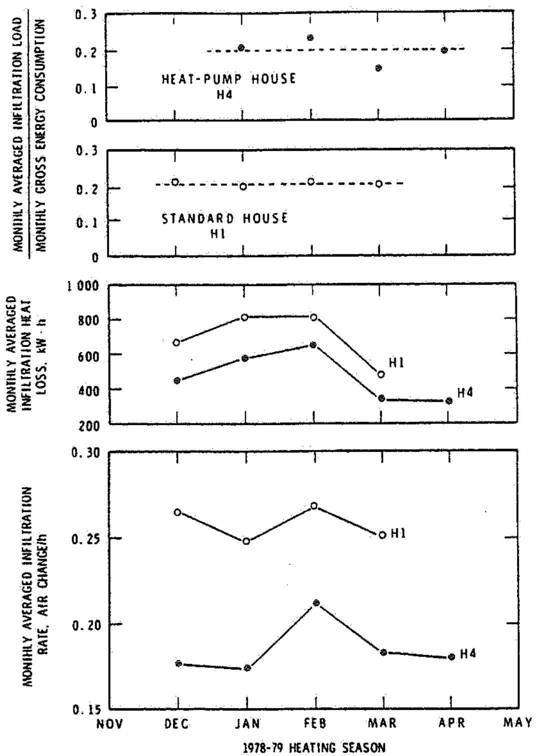

FIGURE 13

MONTHLY AVERAGED INFILTRATION LOAD AND ITS CONTRIBUTION TO TOTAL ENERGY CONSUMPTION

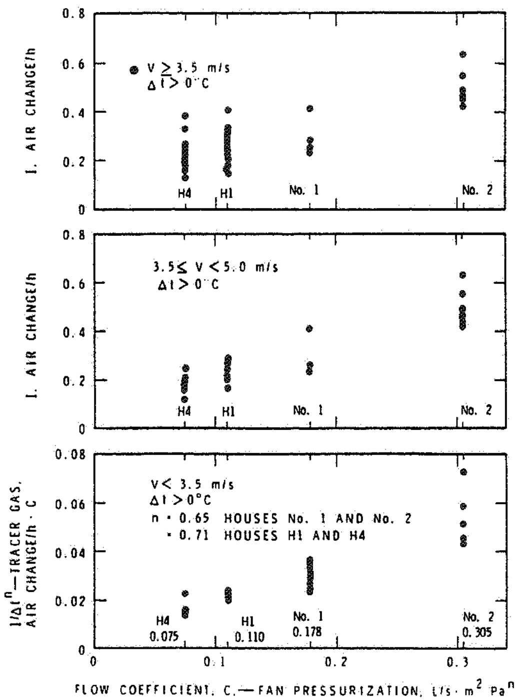

TABLE 1
Air Infiltration rates - Spring, Autumn and Winter Results
(Tracer Gas Decay)

DATE 1978-1979	Wind Speed (m/s)	Wind Direction	Air Temperature Inside (°C)	Outside	Standard House Air changes/hr	Upgraded House Air changes/hr
Jan. 26	4.83	N	22.4	4.0	0.219	0.181
Dec. 20	1.65	N	20.4	-12.7	0.264	0.176
Feb. 12	0.98	N	21.8	-15.6	0.269	0.178
Apr. 10	7.33	N	22.0	7.5	0.334	0.258
Feb. 26	5.14	NE	20.6	-2.6	0.178	
Feb. 21	1.61	NE	22.8	-2.0	0.195	0.135
Apr. 9	9.34	E	22.5	4.6		0.182
Aug. 10	5.01	E	21.1	12.5		0.107
Apr. 2	8.05	E	22.2	3.8	0.214	0.149
Jan. 23	1.43	E	21.0	-5.6	0.236	0.148
Jan. 24	8.00	E	22.0	-4.7	0.260	0.195
Feb. 19	1.16	E	22.0	-15.9	0.279	0.196
Feb. 23	7.64	E	22.4	-1.0	0.281	0.247
Jan. 17	7.78	E	20.5	-19.5	0.314	
Feb. 20	6.48	S	22.4	-3.7	0.270	0.206
Apr. 6	7.38	S	22.0	-0.2	0.316	0.200
Jan. 5	5.95	SW	20.0	-9.0	0.256	
Apr. 20	3.84	W	22.5	15.0		0.114
Apr. 12	3.35	W	22.5	9.5		0.087
Feb. 28	1.07	W	21.4	3.9	0.155	0.103
Feb. 27	2.19	W	20.2	0.5	0.168	
Mar. 16	4.43	W	21.8	-4.4	0.210	0.156
Jan. 22	9.61	W	21.0	-4.7	0.222	0.201
Jan. 4	5.68	W	19.8	-9.3	0.255	0.181
Feb. 1	5.23	W	22.6	-9.6	0.265	0.201
Jan. 31	4.69	W	22.5	-7.4	0.267	0.108
Feb. 15	3.84	W	22.1	15.4	0.270	0.188
Jan. 30	5.63	W	22.6	-5.5	0.271	0.196
Jan. 19	2.32	W	19.0	-18.2	0.284	
Mar. 15	6.30	W	21.3	-12.2	0.292	0.206
Jan. 29	6.12	W	23.4	-3.4	0.301	0.201
Feb. 2	5.99	W	23.1	-10.8	0.303	0.197
Feb. 9	5.23	W	21.8	-18.7	0.326	0.212
Feb. 14	5.86	W	20.7	-19.9	0.331	0.208
Apr. 20	2.82	NW	22.5	16.2		0.082
Aug. 15	5.27	NW	22.5	15.5		0.201
Apr. 18	6.39	NW	22.2	9.3		0.255
Apr. 14	5.99	NW	22.5	0.3		0.268
Aug. 15	8.05	NW	22.5	15.0		0.322
Feb. 13	3.93	NW	20.7	-2.6	0.274	0.205
Feb. 16	4.11	NW	22.1	-16.1	0.288	0.199
Feb. 5	10.55	NW	22.0	-10.6	0.415	0.352

TABLE 2
Air Infiltration rates - Summer Results
(Tracer Gas Decay)

DATE 1978-1979	Wind Speed (m/s)	Wind Direction	Air Temperature Inside (°C)	Outside	Upgraded House Air changes/hr
Jul. 17	3.49	N	22.2	25.1	0.075
Jul. 17	3.13	N	22.2	25.4	0.116
Aug. 21	3.84	N	22.2	24.6	0.122
Jul. 17	3.93	N	22.2	25.1	0.123
Aug. 20	1.12	NE	21.7	21.5	0.073
Aug. 20	1.74	S	22.8	24.4	0.050
Aug. 13	2.41	S	22.5	23.6	0.080
Jul. 31	7.09	S	22.2	26.8	0.118
Jul. 31	6.39	S	21.9	24.5	0.120
Jul. 18	2.28	W	22.2	26.5	0.087
Jul. 19	5.14	W	21.1	29.0	0.105
Aug. 16	4.60	W	21.7	19.0	0.212

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:                                                                      :
:                                                                      :
: Report prepared according to: INTERNATIONAL ENERGY AGENCY            :
:                               energy conservation in buildings and   :
:                                        community systems programme   :
:                                                                      :
:                                                                      :
:            Air Infiltration and Ventilation Centre's                 :
:                                                                      :
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: CV4 7EZ                                                              :
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VENTILATION SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 41

A SIMPLE METHOD USING TRACER GAS TO IDENTIFY THE MAIN AIR AND CONTAMINANT PATHS WITHIN A ROOM.

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Synopsis

The main air- and contaminant flow paths or the spatial distribution of the age of air (or contaminant) in a room are of great interest to estimate the ventilation efficiency performance. A simple measurement method is presented, which consist to inject one or more tracer gases at locations of interest and to analyze the concentration at several other locations, carefully chosen for best accuracy.

Response functions can be fitted on these measurements, which are the age of the tracers or of the air or the concentration of the tracers in function of the location. As well the salient paths as the dead zones are determined from these functions.

The contribution presents the method and its application and validation on a well known and controlled room.

List of symbols

$\underline{A}$	vector containing the coefficients a, b_i, b_{ij} $(i \neq j)$ and b_{ii}	
a, b_i, b_{ij}:	coefficients in a model	
$C(\underline{r},t)$	concentration of a tracer at location $\underline{r}$ and time t.	[-]
$C_e(t)$	concentration measured in the exhaust duct	[-]
$C_o(t)$	concentration measured outdoors	[-]
$C_x(t)$	concentration measured at the location x	[-]
$f(\tau)$	probability density function of the age of air particles	$[s^{-1}]$
$F(\tau)$	probability function of the age of air particles	[-]
$\underline{\underline{M}}$	effect matrix, resulting from an experimental design and a model	
n	air change rate	$[s^{-1}]$ or $[h^{-1}]$
$\underline{r}$	vector locating a point in the space	[m]
t	time	[s]
$\underline{V}$	vector containing the measured quantities $(v_1, v_2, \ldots, v_n)$ at n locations	
V	volume of the room	$[m^3]$
x_i	coordinates of a measured point (i = 1 to 3)	[m]
$\delta\underline{A}$, $\delta\underline{V}$, $\delta\underline{\underline{M}}$:	errors in the measurements of $\underline{A}$, $\underline{V}$ or $\underline{\underline{M}}$ respectively	
$\lVert\underline{\underline{M}}\rVert$	norm of the matrix $\underline{\underline{M}}$	
$\lVert\underline{V}\rVert$	corresponding norm of the vector $\underline{V}$	
cond $\underline{\underline{M}}$:	condition number of the matrix $\underline{\underline{M}}$	
Δt	time interval	[s]
ε	air change efficiency	[-]
ε_c	ventilation efficiency or pollutant removal effectiveness	[-]
μ_1	largest eigenvalue of $\underline{\underline{M}}^H \underline{\underline{M}}$, ($\underline{\underline{M}}^H$ being the hermitic conjugate of $\underline{\underline{M}}$)	
τ	age of air	[s]
$\tau(\underline{r})$	mean age of air at location $\underline{r}$	[s]
τ_n	nominal time constant of the room	[s]
$\langle\tau\rangle$	room mean age	[s]

1. Introduction

An efficient ventilation should bring fresh air to the inhabitants and quickly eliminate the indoor contaminants. The flow paths of the fresh and contaminated air are hence of great importance and a method useable to map either the age of air or the contaminant concentration is needed.

Demand controlled ventilation, in which the air flow rates are controlled to maintain the concentration of a characteristic contaminant under a target limit, needs gas sensors installed at locations which are representative of the inhabited volume of the room. These locations should be chosen according the pollutant sources, the characteristics of the ventilation system and the room itself to optimize the control of the air flows, hence to minimize the contaminant concentration. The knowledge of the air flow pattern of the room may be of great help in the choice of these locations.

This pattern may be calculated using sophisticated codes[1] but can also be estimated in existing building by the measurement of the concentration pattern within a room resulting from a particular pollutant source or, if the pollutant source is not known, by the measurement of the age of the air in the room.

Various measurement methods, most of them using tracer gases, can be used to analyze the ventilation system characteristics and the spread of the contaminants in the room, and therefore to choose the best location of the contaminant probes.

More precisely, in a ventilated room, the scope of the measurements could be the followings:

To characterize **the ventilation system**:

1) determine the flow rate in the air ducts (pulsion, extraction, short-circuits);
2) measure the global air change rate in the room;
3) determine the ventilation efficiency of the system, that is the part of the total flow rate which comes from the ventilation system and the part coming from infiltration

To characterize the **indoor air quality**

4) determine the age of the air at different locations in the room;
5) get a map of the contaminant concentration resulting from several contaminant sources and a given flow pattern resulting of ventilation.

To characterize the complex **room-ventilation system**

6) determine the air exchange efficiency that is how the fresh air is distributed in the room;
7) determine the ventilation efficiency, that is the efficiency in extracting the contaminants generated in the room.

Each particular scope cited above in the introduction has its own measurement method(s) allowing to measure the necessary quantities. In this contribution, only the mapping experiment is described, together with an example measured in a well controlled test room.

2. Planning of mapping experiments

Mapping the contaminant concentration of the age of air in a room can be of great advantage in studying the contaminant or air flow pattern ant their effects on the occupants. Such maps were already calculated using computer codes[1] and some qualitative representations were drawn from measurements[2].

The purpose of this paper is to propose a systematic way to obtain a coarse map of contaminant concentration or of the age of air from measurements. Such map allows to locate the dangerous locations or the dead zones within the measured room, and in the conditions prevailing during the experiments.

2.1. Minimum number of measurements

A map of any scalar variable v in a three-dimensional room is basically obtained by measuring the variable at each node of a network and interpolating between these nodes. Such measurement are however very expensive if not unfeasible: if only 5 values are taken on each axis, there are not less than 125 measurements. Therefore, it makes sense to look for methods needing a minimum number of measurements points.

The minimum number of measurements depends on the scope of the mapping experiment, or more precisely on the empirical model which is chosen to represent the map of the variable v. If a **linear** model is adopted, such as:

$$v = a + \Sigma_i b_i x_i \tag{1}$$

where x_i are the three coordinates of the measured point, only 4 measurements are needed to obtain a set of coefficients $\{a, b_i\}$. If more measurements are made, the coefficients may be obtained by a least square fit procedure, provided there is no (or negligible) uncertainty on the coordinates. It their coordinates differ for the other points, these supplementary measurement points give information on the validity of the used model.

If the linear model does not appear to be valid, more sophisticated models may be used. For example, a quadratic model:

$$v = a + \Sigma_i b_i x_i + \Sigma_{i \neq j} b_{ij} x_i x_j + \Sigma_i b_{ii} x_i^2 \tag{2}$$

which contains 10 coefficients can be chosen. Such model may already fit many practical situations and present minimal and maximal value(s). For that model, measurements at 10 locations is the minimum.

2.2. Location of the measurement points

The next problem is: where should we locate the measurement points? There are numerous possible experimental designs, but they do not give the expected results with the same accuracy. For example, it is obvious that, to fit a linear model of one dimension only ($y = ax + b$), the location of the two measurement points (the minimum number) which gives the best accuracy on a and b is at the ends of the experimental domain. If the model is more sophisticated or for a larger number of dimensions, the choice is not so obvious. However, several tools exist for planning such experiments, which are found in the literature[3], [4] and are applied below.

Let us take a coordinate system in a rectangular room using as unit, for each direction, the half-length of the room in that direction. Any point in the room is then located by three numbers included in the interval [-1, +1].

The experimental design can be represented by a rectangular matrix with 3 lines (one for each coordinate) and as much columns as measured points. For example the design:

Point No	1	2	3	4
x =	1	-1	1	-1
y	1	-1	-1	1
z	1	-1	1	-1

is very well suited to obtain the coefficients of a linear model. This design may be expanded the following way up to 10 locations for a quadratic model:

Point Nr	1	2	3	4	5	6	7	8	9	10
x	1	1	1	1	1	0	-1	-1	-1	-1
y	1	1	-1	-1	-1	0	1	1	-1	-1
z	1	-1	1	0	-1	0	1	-1	1	-1

which represent the 8 corners of the room, its center and the center of a vertical angle. This design looks pretty, but cannot provide the coefficients, because the resulting system of equations is singular. To elaborate the most efficient design, it is necessary to look at the method used to obtain these coefficients.

2.3. Criteria for location of the measurement points

For each point, the model (equation (1) or (2) or any other model) is applied, replacing the x_i by their values given by the experimental design. A system of equations (one equation for each location) is obtained this way, which can be written in a matrix notation:

$$\underline{V} = \underline{\underline{M}}\,\underline{A} \tag{3}$$

where:

$\underline{V}$ $(v_1, v_2, \ldots, v_n)$ is a vector containing the measured quantities at the n locations,

$\underline{\underline{M}}$ is a matrix, each line of which corresponding to one location. Its first column is filled with ones and correspond to a constant in the model. The next 3 columns may contain the coordinates of the locations if the model contains linear terms. The next 3 columns may contain the products of these coordinates two by two (e.g. x_1x_2, x_1x_3, x_2x_3) in case of interaction terms and, for a quadratic model, the next three columns contain the squares of the coordinates. Other models will produce other matrices.

$\underline{A}$ is a vector containing the coefficients (e.g. a, b_i, b_{ij} $(i \neq j)$ and b_{ii}) of the model.

In the general case, $\underline{\underline{M}}$ is rectangular and the least square fit procedures is used:

$$\underline{A} = (\underline{\underline{M}}'\underline{\underline{M}})^{-1}\, \underline{\underline{M}}'\, \underline{V} \tag{4}$$

where $\underline{\underline{M}}'$ is the transposed matrix of $\underline{\underline{M}}$. This equation is also valid if M is a square matrix, but reduces to the simpler equation:

$$\underline{A} = \underline{\underline{M}}^{-1}\ \underline{V} \tag{5}$$

Anyway, a matrix shall be inverted and the determinant of this matrix should not be zero, as it is the case for the 10 points design shown above! Since this determinant can be calculated before making the experiments, it is a first criterion for the choice of the experimental design.

If there are experimental errors $\delta\underline{V}$ in $\underline{V}$ or $\delta\underline{\underline{M}}$ in $\underline{\underline{M}}$, these errors will propagate through the equations and errors $\delta\underline{A}$ will result in the coefficients in $\underline{A}$. In case of complete experiments, that is if equation (5) is used, it is shown[5] that these errors are smaller than:

$$\frac{\|\delta\underline{A}\|}{\|\underline{A}\|} \leq \frac{\|\underline{\underline{M}}\|\cdot\|\underline{\underline{M}}^{-1}\|}{1 - \|\delta\underline{\underline{M}}\|\cdot\|\underline{\underline{M}}^{-1}\|} \cdot \left(\frac{\|\delta\underline{V}\|}{\|\underline{V}\|} + \frac{\|\delta\underline{\underline{M}}\|}{\|\underline{\underline{M}}\|} \right) \tag{6}$$

where $\|\underline{\underline{M}}\|$ is a norm of the matrix $\underline{\underline{M}}$ and $\|\underline{V}\|$ the corresponding norm of the vector $\underline{V}$. Any norm can be used but it may be advantageous to use the smallest ones, which is the euclidian norm of the vector $\underline{V}$:

$$\|\underline{V}\|_2 = \sqrt{(\Sigma_i\, v_i^2)} \tag{7}$$

and the corresponding spectral norm for the matrix $\underline{\underline{M}}$:

$$\|\underline{\underline{M}}\|_2 = \sqrt{\mu_1} \tag{8}$$

where μ_1 is the largest eigenvalue of $\underline{\underline{M}}^H\,\underline{\underline{M}}$, ($\underline{\underline{M}}^H$ being the hermitic conjugate of $\underline{\underline{M}}$).

However, the following norms, which leads to faster calculations are often used:

$$\|\underline{V}\|_1 = \Sigma_i\, |v_i| \tag{9}$$

and the corresponding norm for the matrix $\underline{\underline{M}}$:

$$\|\underline{\underline{M}}\|_1 = \max\,(\|\underline{M}_j\|_1) \tag{10}$$

where $\underline{M}_j$ are the column vectors of $\underline{\underline{M}}$.

The number

$$\text{cond}\ \underline{\underline{M}} = \|\underline{\underline{M}}\|\cdot\|\underline{\underline{M}}^{-1}\| \tag{11}$$

appearing in (6) is of first importance here, since $\|\delta\underline{\underline{M}}\|\cdot\|\underline{\underline{M}}^{-1}\|$ is generally much smaller than 1. It is the **condition number** of the matrix $\underline{\underline{M}}$, which multiplies the experimental errors and transmit these errors into the result $\underline{A}$. **This number depends only on the experimental design and on the chosen model.** It can hence be calculated **before** doing any experiment and constitutes a second (and even better) criterion for the choice of the experimental design.

2.3. Examples of experimental designs

Several experimental designs were examined with the aim to map a 3-D cube with 10 measurements. The tested models were those of equations (1) and (2) plus an interaction model which is (2) without the square terms.

Several of these designs were found to be unusable (singular effect matrix or too large condition number for the quadratic model). The remaining designs are the followings:

C3 has 6 points in the center of the faces and 4 points at opposite corners. It was obtained by selecting 10 points out of the 27 points of a three level factorial† design in such a way that the matrix $\underline{\underline{M}}$ gets the lowest condition number(figure 1). The same result is obtained starting from the 125 point of a 5-level factorial design.

C4 has 6 points near the center of the faces and 4 points at opposite corners. It was obtained by selecting 10 points out of the 65 points of a four level factorial design which present the lowest

† A k-dimensional, n-level factorial design is obtained by dividing the experimental domain (e.g. the interval [-1, 1]) on each axis into n equidistant levels. The complete factorial design contains all the points obtained by the n^k combination of the n possible values of the coordinates.

condition number for the matrix $\underline{\underline{M}}$.

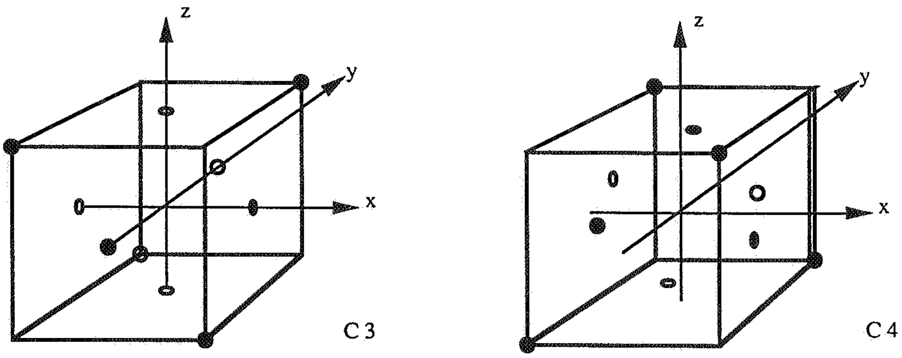

Figure 1: Experimental designs C3 (left) and C4 (right)

The **hybrid** design has two points on the z-axis (z = -0.136 and z = 1.2906), four factorial points on the plane z = 0.6386 and a star-design in the plane z = -0.9273. Since there are points at locations larger than 1, all the coordinates should be divided by 1.2906. (figure 2).

The 3 Δ design is made with points at the summits of 3 triangles placed at levels z = -1, 0 and 1, the middle one being in opposite direction. The 10th point is at the center of the room.

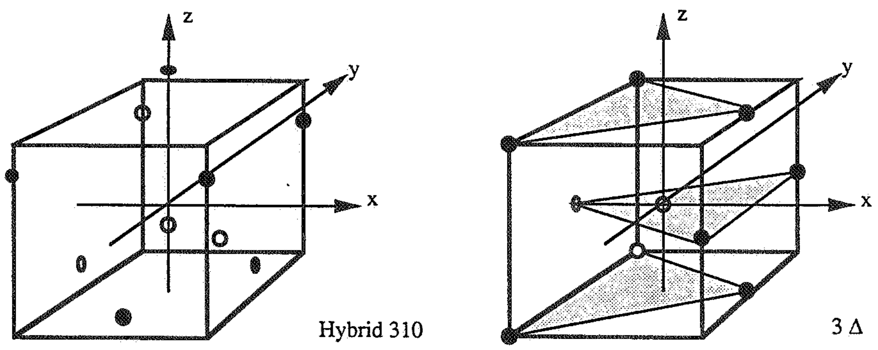

Figure 2: Hybrid (left) and 3 Δ (right) experimental designs

The **Hoke D2** and the **Rechtschaffner** design have both 7 points at the summits of the cube and 3 points at the center of faces, but the position of these last three points with respect to the other differ strongly (figure 3). These designs were found in the literature[3] and chosen among others for having 10 measurement points.

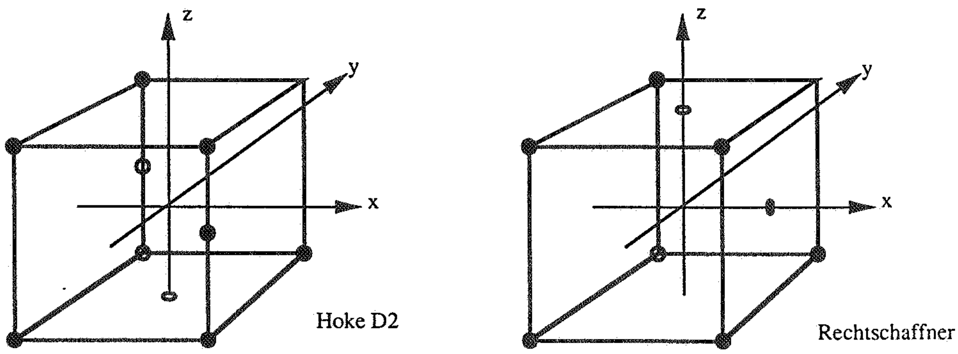

Figure 3: Hoke D2 (left) and Rechtschaffner (right) experimental designs

The condition number of $\underline{\underline{M}}'\underline{\underline{M}}$ calculated according the equation (10) and (11) for these designs and three models are given in table 1.

Experimental design	Quadratic model	Interactions model	Linear model
C 3	4.33	3.15	**1.00**
C 4	6.67	2.62	1.15
Hybrid 310	6.70	**1.00**	**1.00**
3 Δ	8.88	1.414	1.11
Hoke D2	7.29	*3.19*	*1.66*
Rechtschaffner	*9.24*	2.03	1.23

Table 1: Condition number of $\underline{M}'\underline{M}$ for some experimental designs and three models. For each model, he best values are in bold characters and the worse in italics.

The three latter designs lead to a large condition number for a quadratic model, that is to large errors in the estimates of the coefficients. C 3 seems to be the best if the model is quadratic or linear, when the hybrid and 3 triangles are better for interactions and linear models. However, the hybrid model does not map the whole domain and looks very asymmetric.

Finally recommended designs are C 3 if the quadratic or linear model is used and 3Δ for interaction or linear model.

It may also be interesting to draw the map of a contaminant (e.g. the CO_2) in the flat volume where the heads of the inhabitants are located. In this case, a two-dimensional design may be used. Using a limited number of sampling points, the following designs may be recommended:

A 2-D full factorial design with 3 levels:

X =	-1	0	1	-1	0	1	-1	0	1
Y =	-1	-1	-1	0	0	0	1	1	1

which gives a condition number of 4.39 for a quadratic model. If 10 points are wanted, adding (-0.5, 0) and (0.5, 0) lowers the condition number to 3.48. The minimum number of experiments is 6 for a quadratic model. The design:

X =	-1	1	-1	0	1	-1
Y =	-1	-1	0	0	0	1

has a condition number of 6.

For an interaction model, a recommended design is:

X =	-1	1	0	1	-1
Y =	-1	-1	0	1	1

which is also best for a linear model, even when the point (0,0) is removed. These designs have a condition number of 1, which is an absolute minimum.

3. Mapping the concentration of a contaminant

If the characteristics of the "natural" contaminant source(s) is known, they can be simulated either by the contaminant itself if it is easily analyzed for non-toxic concentrations (like CO_2 or H_2O) or by the injection of tracers at appropriate flow rates at the same locations.

The air is sampled at regular intervals at the locations corresponding to the experimental design and analyzed. For each sampling time, or if the air- and contaminant flow pattern are in a steady state, a model is fitted on these points to estimate its parameters.

Once the parameters estimated, the model can be used to find the probable location of a minimum or maximum concentration, to estimate the average concentration or the total amount of contaminant within the room.

The ventilation efficiency, also called the pollutant removal effectiveness is generally defined as the ratio of two net concentrations:

$$\varepsilon_c = \frac{C_e - C_o}{C_x - C_o} \qquad (12)$$

where:

C_e is the contaminant concentration in the exhaust air

C_o is the contaminant concentration in the outdoor air

C_x is the contaminant concentration of interest.

The concentration of interest may be teh concentration at a particular location in the room, the maximum concentration or the average of the whole room. Depending on that definition, different efficiencies may be defined and measured.

It is obvious that from the map of the concentration C_x and the measurement of the exhaust concentration C_e, a map of the ventilation efficiencies can be obtained, as well as any other value as maximum, average or minimum efficiency. That assumes that the air leaves the room through a single exhaust duct.

When a first map is obtained, a "zoom" experiment may follow to get more details in an interesting part of the room. This experiment may use the same design but for a smaller volume.

4. Age of the air

The particles of air coming from outside or from the ventilation system arrive at a given location in a room after a time τ which vary from one particle to the other. τ is called the age of the particle, as if it were born when entering the room.

Since there is a large number of air particles, a probability density $f(\tau)$ that the age of particles arriving at a given location is between τ and $\tau+d\tau$ and a probability $F(\tau)$ that this age is smaller than τ can be defined. It is shown that these probability functions can be measured by registering the concentration $C(t)$ of a tracer versus time, this tracer being either at uniform concentration at the beginning of the experiment, or injected at constant rate or as a short pulse at a single air inlet[6], [7].

The mean age of air at a given location in the room can be deduced from these distributions. The convenient formulas to be used are found in the literature[6], [7]. If, as it was done in our experiments, the age is deduced from the decay in concentration of a tracer initially distributed in the room, the local mean age is calculated from:

$$\tau(\underline{r})= \frac{\int_0^\infty C(\underline{r},t)\,dt}{C(0)} \tag{13}$$

The average of the local mean age of air over the whole room is the room mean age $\langle\tau\rangle$. If the air leaves the room through a unique opening or duct, this room mean age can be also measured with tracer gas, the methods of injection being the same as before but the concentration $C_e(t)$ being measured in the exhaust duct and using equation (15) below.

If, as it is often the case, the air leaves the room through multiple openings, leaks and outlets, the technique mentioned above cannot be used. However, the local age of air $\tau(\underline{r})$ can be measured and mapped as well as a concentration, using the same experimental design. The average of the local mean age of air over the whole room is the room mean age of air $\langle\tau\rangle$.

5. Measurements

5.1. The experimental chamber

In order to evaluate the feasibility of the method presented above, measurements were done in a well controlled room called CHEOPS for "CHambre Expérimentale d'Observation de Phénomènes de Stratification" (figure 4). This room has airtight, lightweight walls, floor and ceiling. Its internal dimensions are 3.5 × 4.0 m with a height of 2.7 m, its volume is hence 37.8 m^3. It is completely contained in a second enclosure measuring 5 × 6 × 4 m, which is made of panes plated with 10 cm polystyrene foam, in order to control the temperature and temperature gradients within the internal room.

For that particular experiment, an inlet and an outlet openings were managed in two walls, as shown in figure 4. The airtightness of the room with closed openings was measured and found to be so high that air flows trough parasitic leaks can be neglected when compared to the main air flow.

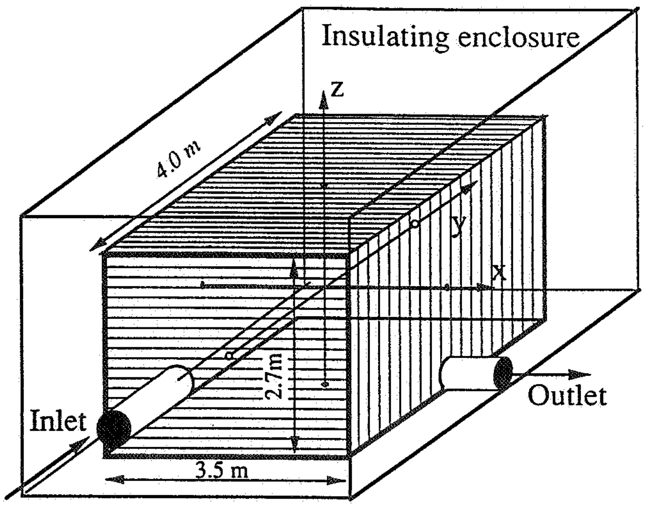

Figure 4: The experimental chamber CHEOPS, with the inlet and outlets used for the present experiment.

5.2. The experiments

Before the experiments, the ends of 10 sampling tubes are fixed at 10 different locations according the two selected experimental designs, namely C3 and 3Δ. The other ends of these tubes are connected to a sampling scanner allowing to analyze in sequence the air samples coming from these 10 locations. The analyzer was a Binos from Leybold Heraeus and the tracer was N_2O.

During the experiments, an air flow rate of 18.9 m^3/h, (which corresponds to an air change of 0.5 /h or a nominal time constant of 2 hours) was blown at the inlet. The air flow rate was controlled by both an air flow meter and tracer gas (constant concentration) techniques.

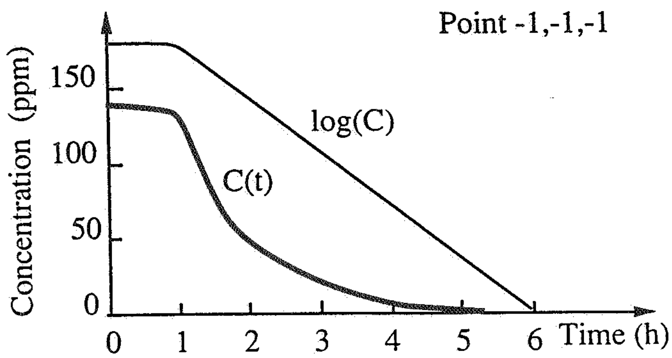

Figure 5: Record of the concentration versus time (full line) at one location in the room, after cutting off the tracer injection. The dotted line shows the log of the concentration on an arbitrary scale. Note that the decay starts nearly one hour after the tracer injection stops.

After stabilization of the air flow and of the concentration, that is after about 4 hours, the tracer injection was stopped and the decay of the concentration was measured at 10 different locations according the two selected experimental plans (figure 5). The scanning interval between two channels was 42 seconds, so the time period between two measurements at the same location was 420 s or 7 minutes.

From these 10 decay curves, the local mean age of air were deduced using equation (13). These results are reported on table 2, for two experiments with two different designs.

x	y	z	3Δ	C3
-1	-1	-1	1.43	1.86
0	0	-1		1.83
1	0	-1	2.16	
-1	1	-1	1.41	
1	1	-1		1.56
0	-1	0		1.05
1	-1	0	1.02	
-1	0	0	1.09	1.06
1	0	0		0.99
0	0	0	1.06	
0	1	0		1.08
1	1	0	1.02	
-1	-1	1	0.94	
1	-1	1		0.98
0	0	1		0.99
1	0	1	1.00	
-1	1	1	0.96	0.96
Average of the ages (eq. 16)			1.209	1.236

Table 2: Ages of air in the CHEOPS chamber for two experiments. The coordinates are given in units corresponding to the half dimensions of the chamber. The air outlet is at 1, 0, -1 and the inlet at -0.5,-1, 0

First a quadratic model was adjusted on these data for both experiments. Since the coefficients of the y, xy and yz terms were very small, a limited model was also tested, using:

$$\tau(\underline{r}) = a + b_x x + b_z z + b_{xz} x z + b_{xx} x^2 + b_{yy} y^2 + b_{zz} z^2 \qquad (14)$$

The table 3 gives the coefficients of these two models and the room mean ages obtained by different methods.

One of the measurement point in design 3Δ is located at the air outlet. The room mean age of air can hence also be measured using:

$$\langle\tau\rangle = \frac{\int_0^\infty t\, C_e(t)\, \mathrm{d}t}{\int_0^\infty C_e(t)\, \mathrm{d}t} \qquad (15)$$

where $C_e(t)$ is the tracer concentration in the only air outlet of the room. This value can be compared with the other values obtained either by the direct average of the measurements of the local mean ages:

$$\langle\tau\rangle = \frac{1}{N} \sum_i \tau_i(\underline{r}_i) \qquad (16)$$

or by the average of the obtained map: if a linear or interaction model is used, the room mean age of air is simply the constant term of the model. If a quadratic model is used, then:

$$\langle\tau\rangle = a + \frac{2}{3} \sum_i b_{ii} \qquad (17)$$

where a is the constant term of the model and b_{ii} are the coefficients of the square terms.

Model:	Quadratic		Limited	
Coefficient for:	3Δ	C3	3Δ	C3
Constant	1.06	1.08	1.06	1.08
X	0.08125	-0.03	0.08125	-0.0583
Y	0	0.01		
Z	-0.4075	-0.42	-0.4075	-0.3867
X*Y	0	-0.05		
X*Z	-0.1725	0.09	-0.1725	0.08
Y*Z	0.01	0.03		
X^2	0.1112	-0.055	0.1112	-0.055
Y^2	-0.2325	-0.015	-0.2325	-0.015
Z^2	0.3275	0.33	0.3275	0.33
Mean age (eq. 17)	1.197	1.253	1.197	1.253
Room mean age measured at the outlet (eq. 15, 3Δ design):				1.21 h

Table 3: Coefficients of two models calculated from measurements done with similar conditions but with two different experimental designs. The coefficients which are very small in the quadratic model are put to zero in the limited model.

A map of the local mean age of air can be drawn using the model, as it is shown in figure 10.

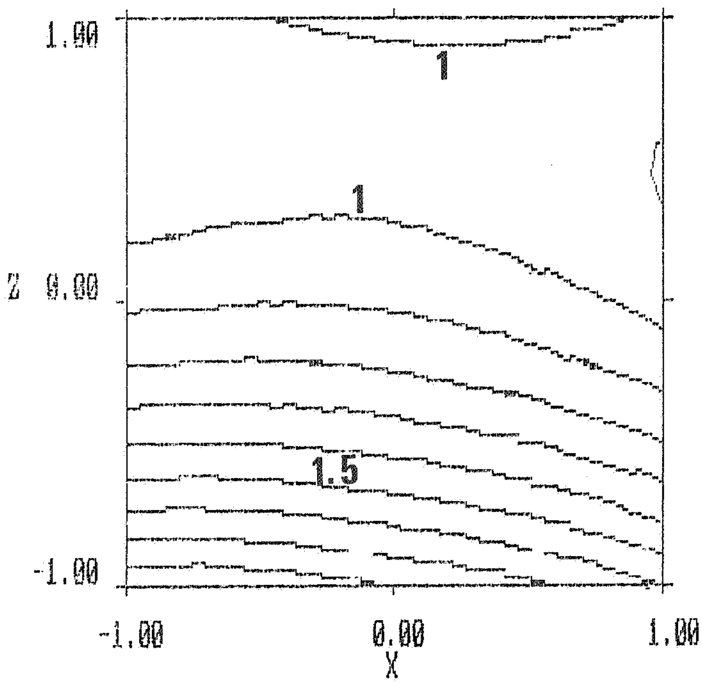

Figure 10: Lines of constant age of air in the plane y = 0 drawn with the limited model adjusted on the measurements done with the C3 design. The younger air is on the top of the room even with the inlet at the mid-plane.

Finally, the air change efficiency, which expresses how quickly the air in the room is replaced, is calculated by:

$$\varepsilon = \frac{\tau_n}{2\langle\tau\rangle} = \frac{2}{2.4} = 0{,}83 \tag{18}$$

where $\langle\tau\rangle$ is the room mean age of air and τ_n is the nominal time constant of the room. This efficiency is 1 for piston ventilation and 0.5 for complete mixing.

6. Discussion and conclusions

The comparison of the figures in table 2 show that:

1) as expected, there are important differences (up to a factor 2) in the age of air at different locations. It confirms that either a strong mixing or sampling at several places is necessary to measure the air change rate or the nominal time constant of the room with tracer gas techniques.
2) There are slight differences between the two experiments, as it can be seen by the values marked in bold characters, which are measured at the same locations. This effect is particularly high at the point -1, -1, -1. Even when the experiment is done in very similar conditions, tiny changes in air temperatures may affect the flow pattern.

As shown in table 3, the coefficients of the model obtained with two different designs does not differ much. Moreover, the room mean ages obtained by various ways from the measurements are very close together. This value is affected neither by small changes in air temperature nor by the experimental planning, as far as it is good enough.

Hence, if there is not only one air outlet, or a fortiori when the air outlets cannot be clearly identified, the room mean age of air may be obtained by averaging several measurements taken at convenient locations. It can be safer however to deduct it from the coefficients of a fitted model.

Finally, it is shown that, if the experiments are well designed, it is possible to draw a map either of the concentration of a contaminant and the related quantities as the ventilation efficiency or of the local mean age of air, with a reasonable amount of measurements. From these maps, as well the main fresh or contaminant flows as the dead or well ventilated zones can be deduced.

However, sharp details cannot be reproduced with a limited amount of measurement points. It is the case of the distribution of the age of air (or of the contaminant concentration) near the fresh air inlet (or the contaminant source). If the number of measurement points is limited, these few measurements can be used to calibrate a simulation with a sophisticated code (e.g. Phoenics) at these locations. The code is then used to interpolate correctly between the measurements and more confidence is gained in the description of the details.

7. Acknowledgements

This research was sponsored by the NEFF ("Nationaler Energie-Forschungs-Fonds" or National Energy Research Funds). The codes NEMROD from LPRAI, Marseille and XSTAT (John Wiley and sons) were used to design and analyze the experiments.

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Discussion

Paper 41

S.Irving (Oscar Faber, UK)

How much of this new information on measurement techniques will be included in the revised Measurement Techniques being done within Annex 20?

C-A Roulet (LESO, Switzerland)

A text close to the present paper is already included in part IV of the Measurement Guide.

M.Sherman (LBL, USA)

One cannot compare the accuracy of two models by comparing their condition numbers or regression coefficients. The linear model may have a better condition number than the others, but that does not imply it is better - a model with no parameters fits the data perfectly. When one is selecting different models the choice of model should be based on the underlying physics not the statistics.

C-A Roulet (LESO, Switzerland)

I completely agree with you. As proposed in the paper, the condition number is useful to compare two experimental designs, once a model is chosen.

J.Axley (MIT, USA)

If the form of variation of a scalar variable throughout the domain is known a priori (e.g. to be well approximated by linear interpolation, say) then the procedure described is appropriate. But for age of air we do not know the form of variation a priori, specifically we should expect significant gradients across boundary layers. Thus with this in mind it would be best not to measure as the actual physical limits of the domain (i.e. room) but rather somewhat within this physical boundary.

C-A Roulet (LESO, Switzerland)

1. Your comment is completely right. In fact measurements points were located at the extremities of a volume 1m smaller than the room in each direction, that is the points were located at 50 cm of the walls. 2. If the model is not known a priori, a linear model (the simplest) may be chosen and an experimental design planned accordingly. Between the various possible designs, just choose the one having the more common points with a design which is good for 2nd degree model. This way, you can add a minimum number of experiments to the first measurements done if the linear model does not reproduce the realities well enough. And so on for more sophisticated models.

R.Grot (Lagus Applied Tech., USA)

Did you consider using incomplete function basis's. This technique is used in finite element methods. You could also reverse your question; given N measurements, what is the best interpolation scheme?

C-A Roulet (LESO, Switzerland)

There are techniques to answer the reverse question, but a criteria shall be first defined: what is best? (*see references in the paper). However, I remain convinced that it is best to know what knowledge is needed, that is to define a model and adjust the experimental plan accordingly.*

R.J.F.Van Gerwen (Cauberg-Huygen, Netherlands)

The best condition number for the linear model is smaller than for the quadratic model. As the condition number gives more or less the ratio of errors in the results and errors in the measurements one would expect a linear model to be better than a quadratic model. On the other hand a quadratic model uses more measuring points and therefore is more accurate. This seems a contradiction to me.

C-A Roulet (LESO, Switzerland)

It is not. A low condition number signifies that the parameters will be determined with a better accuracy, but does not mean that the model will better fit on the reality. Usually, a model with a large number of parameters fits the reality better than another with a low number of parameters. However, it is not recommended to multiply the number of parameters without limit.

VENTILATION SYSTEM PERFORMANCE

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Paper 42

ERRORS IN THE MEASUREMENT OF LOCAL AND ROOM MEAN AGE USING TRACER GAS METHODS.

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ERRORS IN THE MEASUREMENT OF LOCAL AND ROOM MEAN AGE USING TRACER GAS METHODS

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SYNOPSIS

Local and room mean ages of the air in a room may be measured by three versions of the tracer gas technique; which are the pulse method, the tracer step-up up method and the tracer decay method. The values of mean age obtained are of course subject to errors in the measurement of the tracer gas concentrations. The sensitivity of the three methods to errors in the tracer gas concentration is not the same, and in some cases can be very large. In order to examine this problem, test measurements have been carried out in a model room using the three different methods. The results obtained from these tests were then compared with each other, and with theoretical data, which was generated with differing levels of error. It has been found that the step-up method was the least reliable and local and room mean ages generated using this method varied substantially from mean ages measured using the other two tracer gas techniques.

1. INTRODUCTION AND OBJECTIVES

Local mean age, room mean age, and air change efficiency are now well established concepts in the assessment of ventilation systems. They are particularly useful in measuring the distribution of fresh air in ventilated rooms. The concepts themselves are normally defined for a mechanically ventilated room with one inlet and one exhaust duct (see, for example, Sutcliffe [1]). Nevertheless the definitions are such that all three concepts are independent of the means by which air enters and leaves the room. Thus, mechanical ventilation, natural ventilation, partial mechanical ventilation, and uncontrolled infiltration and exfiltration can all be included. However, the measurement of room mean age is most easily accomplished when there is only one inlet and one exhaust duct, and becomes progressively more difficult as the number of inlet and exhaust openings increases. Measurement is most difficult where the exhaust routes are not clearly defined. The most usual methods of measurement are by use of tracer gases in pulse, step-up or decay modes [1]. All three methods may be used where there is only one inlet and one exhaust, and where there is one inlet and several clearly defined exhausts. Where there are multiple inlets and multiple exhausts, it is still possible in principle to use all three methods, thought in practice only the decay method may be reliable. When there is a significant amount of infiltration and exfiltration, then only the decay method is feasible, and when exfiltration is present, only the local mean age can be measured.

The present study arose from an interest in measuring local mean age in industrial buildings. Ventilation in such buildings is often an arbitrary mixture of natural and mechanical ventilation, with a high proportion of uncontrolled air movement through doors, roof vents and leakage paths. Hence, in most cases, only the decay method would be valid. In order to determine the reliability of the decay method, a series of measurements were carried out in a mechanically ventilated model room. The local mean age and the room mean age were measured by all three tracer gas methods, and the measurements repeated several times at each of three ventilation rates. The objective was to compare the results obtained by the three methods and also to determine the repeatability of the measurements. Some calculations were also carried out using simulated data with varying degrees of random error.

2. THE MODEL AND THE MEASUREMENT PROCEDURE

The model consisted of a sealed wooden box of internal volume $0.96m^3$ with one inlet and one exhaust duct, and a variable speed fan in the inlet duct. Six points were sampled, one close to the inlet, one at the exhaust, and the other four distributed within the model as shown in figure 1.

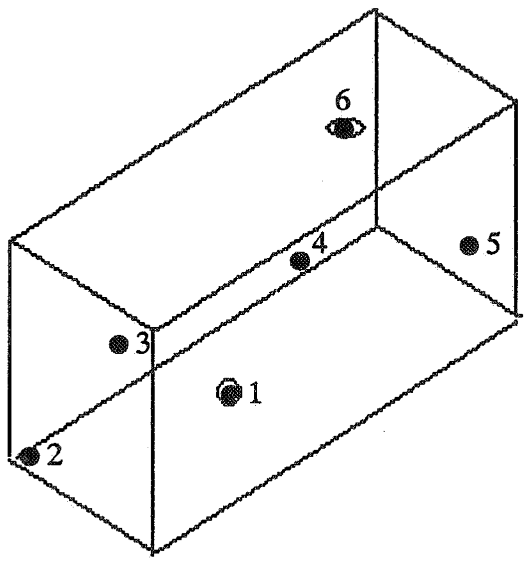

Figure 1. Positions of measurement points within the model

The tracer gas was sulphur hexafluoride, and tracer gas concentrations were measured by means of an on-line ECD chromatograph sampling at the rate of 1 sample per minute on all channels. For pulse tests, tracer gas was premixed with air to a convenient concentration and then injected by syringe into the inlet duct. For decay tests tracer gas was injected into the model, and brought to a uniform concentration by means of an internal fan, which was then switched off. For step-up tests, a mixture of sulphur hexafluoride in air was introduced into the inlet duct from a commercially supplied cylinder via control valves and regulators. The variable speed fan in the inlet duct had three speed settings, giving rise to three ventilation rates. These ventilation rates were measured by tracer decay tests with the internal mixing fan switched on, giving the results shown in table 1.

Table 1. Model Ventilation Rates

Fan Speed Setting	Air Flow Rate (l/min)	Air Change Rate (ach)	Nominal Time Constant (mins)
1	16.41	1.03	58.50
2	35.07	2.19	27.37
3	41.05	2.57	23.39

Pulse, decay and step up tests were typically continued for a period of 90 minutes, which included five or six minutes of measurement of the starting conditions. Thus each test typically produced 85 useful measurements of tracer gas concentration on each channel. Generally this ensured that the measurement period was comfortably in excess of one time constant, and that the measured tracer gas concentrations covered a range in excess of one decade.

3. ANALYSIS OF RESULTS

The local mean age, $\bar{\tau}_p$, was calculated using the usual equations, that is :-

1 Pulse method $$\bar{\tau}_p = \frac{\int_0^\infty t.C_p(t)\,dt}{\int_0^\infty C_p(t)\,dt}$$

2 Step-up method $$\bar{\tau}_p = \int_0^\infty \left(1 - \frac{C_p(t)}{C_s}\right) dt$$

3 Decay method $$\bar{\tau}_p = \int_0^\infty \frac{C_p(t)}{C(o)}\,dt$$

Similarly, for room mean age, $<\bar{\tau}>$, the equations used were :-

1. Pulse method $$<\bar{\tau}> = \frac{Q}{2V}\frac{\int_0^\infty t^2 C_e(t)\,dt}{\int_0^\infty C_e(t)\,dt}$$

2. Step-up method $$<\bar{\tau}> = \frac{Q}{V}\int_0^\infty t\left(1 - \frac{C_e(t)}{C_s}\right) dt$$

3. Decay method $$<\bar{\tau}> = \frac{Q}{V}\int_0^\infty t\,\frac{C_e(t)}{C(o)}\,dt$$

The integrals were evaluated in two parts. The first part was found directly from the data set by summation up to and including the final reading. The remaining, unmeasured, part of the integral was treated as an end correction and evaluated on the assumption that the remainder of the time evolution of concentration followed a single exponential law with decay constant, The value of λ was determined separately for each curve by first examining the plot of the natural logarithm of concentration (or (C_s - C(t)) in the case of step-up) versus time, determining visually the time at which the plot became essentially a straight line, and then performing a least squares fit on the relevant data points to find the gradient. If t_f is the time of the final measured data point, and $C(t_f)$ the concentration value, the end corrections are then as follows :

$$\int_{t_f}^{\infty} C(t)\,dt = \frac{C(t_f)}{\lambda}$$

$$\int_{t_f}^{\infty} t.C(t)\,dt = \frac{C(t_f)}{\lambda}\left(t_f + \frac{1}{\lambda}\right)$$

$$\int_{t_f}^{\infty} t^2.C(t)\,dt = \frac{C(t_f)}{\lambda}\left(t_f^2 + \frac{2t_f}{\lambda} + \frac{2}{\lambda^2}\right)$$

Clearly the magnitudes of the end corrections are large whenever the decay constant is small, and when this is the case, errors in determining λ from the gradient of the log plot are likely to be large. In the case of the step-up test, $C(t_f)$ is replaced by $(C_s - C_p(t_f))$ and since both C_s and $C_p(t_f)$ are subject to error, the error in their difference could become highly significant, contributing even more to the error in the end correction. Thus the end correction may not only be large, but it may also be subject to a large error.

4. RESULTS OF MEASUREMENTS

Table 2 is a complete summary of the results of the measurements of local mean age.

Table 2. EXPERIMENTAL DATA, Local Mean Ages (Mins)

Method	No of Tests	Ch1	Ch2	Ch3	Ch4	Ch5	Ch6
				Fan Speed 1			
Pulse	4	91.0± 9.6	74.1± 6.8	82.5±10.6	92.9± 3.3	77.6± 4.1	76.7±12.9
StepUp89	3	18.3±10.5	59.3±26.2	51.2±11.5	91.5±24.8	46.9± 4.7	78.3±12.3
StepUp90	8	0.0± 0.0	71.8±16.9	75.3±14.3	85.6+10.4	70.8± 8.6	74.7± 5.9
Decay	5	84.8±10.2	80.7± 3.6	93.5± 8.3	93.4± 4.5	92.9± 9.3	95.0± 6.3
				Fan Speed 2			
Pulse	3	29.9± 6.9	36.9± 1.7	34.3± 3.2	52.0± 2.3	31.1± 2.6	26.4± 1.8
StepUp89	3	63.4±25.8	29.6±13.7	32.7± 9.2	59.4±10.3	33.3± .1	63.9±11.4
StepUp90	5	0.0± 0.0	28.7± 3.4	37.2± 2.4	42.5± 3.5	32.3± 4.5	33.0± 2.1
Decay	6	29.6± 3.3	28.9± 2.1	30.9± 3.1	34.1± 3.9	35.0± 6.1	33.7± 3.3
				Fan Speed 3			
Pulse	3	31.8± 1.5	52.2±13.2	39.0± 4.7	54.8± 4.4	30.1± 2.2	29.9± 2.2
StepUp89	4	28.6±25.2	19.2± 3.1	29.6± 5.3	58.5±11.8	30.9± 1.9	60.1± 8.5
StepUp90	5	0.0± 0.0	42.0± 9.8	45.1± 6.3	53.7± 6.6	34.4± 7.5	30.3± 9.1
Decay	4	32.8± 4.1	33.3± 6.0	41.0±10.4	43.3±10.0	40.5± 7.5	37.6± 6.1

Each result is the average of the number of tests indicated, and the standard deviation is also given. The results for step-up tests are given in two separate sets because a number of improvements to the experimental technique were introduced for the second (Stepup 90) set. These were introduced because it was observed that the results for the first set (Stepup 89) were often inconsistent, as shown by large values of the standard deviation, and by some large discrepancies with respect to the pulse and decay methods. Three improvements were introduced, all three aimed at improving the stability and mixing of the tracer gas in the inlet duct. These improvements were :

1. The introduction of a high quality flow controller in the tracer gas inlet stream, with a claimed control capability of 1 part in 5000;

2. The introduction of a flow straightener immediately downstream of the fan in order to stabilise the flow in the inlet duct;

3. A six port injection device to distribute the injected tracer uniformly across the inlet duct according to the probable velocity profile in the duct.

It can be seen that these improvements reduced the standard deviations of the step-up test, and brought the local mean ages more into line with those obtained by the pulse and decay methods. Figures 2, 3 and 4 show the same information as table 2, and the inconsistencies of the first set of step-up tests are clear. Channel 1 was found to be difficult to analyse because its proximity to the inlet duct made it sensitive to turbulence in this region, so that the recorded tracer gas concentration did not follow a smooth curve. Hence results for channel 1 are particularly unreliable, and indeed for the second set of step-up tests channel 1 was not analysed.

Table 3 is a complete summary of the results for room mean age. These results are plotted in figure 5, and the inconsistency of the first set of step-up tests is even more marked.

Table 3. EXPERIMENTAL DATA, Room Mean Ages (Mins)

Method	No of Tests	Fan1	No of Tests	Fan2	No of Tests	Fan3
Pulse	4	72.3±18.9	3	19.0± 1.9	3	23.5± 3.7
StepUp89	3	119.3±23.8	3	119.4±23.8	4	191.4±43.0
StepUp90	8	60.4± 9.1	5	25.0± 3.6	5	34.8±14.6
Decay	5	95.2± 6.6	6	32.3± 3.3	4	38.9± 9.0

5. RESULTS OF SIMULATIONS

In order to explore the effect of random error in the measured tracer gas concentrations, sets of simulated data with different levels of error were prepared. Unfortunately, it is not easy to produce theoretically exact tracer gas curves which match those produced experimentally in the model. Hence the simulations were carried out for the case when the air in the model is fully mixed, in which case the equation for the curve at each point is the same, and follows a simple exponential law. Also the pulse and decay methods produce identical results. The air flow rate was chosen to give a nominal time constant of 33.3 minutes, approximately equivalent to fan speed 2 of the experimental data set. Error was introduced into the data by means of a random error generator with a normal distribution and with a standard deviation set to a fixed percentage of the maximum value in the data set. Results were obtained for 1%, 2%, 5% and 10% levels of error, and four sets of data were generated for each level of error, corresponding to four independent tests. The results were analysed in exactly the same way as the experimental results. Table 4 is a summary of the results for local mean age, and Table 5 a summary for room mean age.

Table 4. SIMULATED DATA, Local Mean Ages (Mins)

% Error	No of Tests	Ch1	Ch2	Ch3	Ch4	Ch5	Ch6
				Step-Up Tests			
1	4	34.6± .3	34.5± .3	34.1± .3	34.4± .2	34.2± .3	34.3± .5
2	4	33.9± .7	34.1± .9	34.2± .3	34.6± .5	35.0± .4	35.0± .5
5	4	34.2± .5	33.9±1.5	33.2±1.2	36.2+1.6	32.7±1.3	33.4±1.3
10	4	32.9±2.2	35.7±3.9	33.4±2.9	35.6±5.0	35.4±4.0	35.9±3.9
				Decay Tests			
1	4	33.4± .2	33.7± .3	33.2± .5	33.2± .4	33.8± .4	33.2± .3
2	4	33.5± .8	33.4± .9	34.0± .8	33.1± .8	33.7±1.1	33.1± .5
5	4	33.6± .9	32.6±1.1	32.0±1.3	32.8±1.8	33.9± .9	33.8±1.0
10	4	31.7±2.7	37.0± .2	31.6±1.4	30.6±1.1	36.3±5.0	32.7±1.8

Table 5. SIMULATED DATA, Room Mean Ages (Mins)

Percentage Error	No of Tests	Step-Up Tests	Decay Tests
1	4	34.2± 1.6	32.8± .5
2	4	36.7± 1.6	33.2± .6
5	4	32.7± 4.3	35.1± 1.7
10	4	43.4±13.6	29.8± 1.9

As an illustration, the results for local mean age for Channel 5 are plotted in figure 6, and the results for room mean age in figure 7. Overall the standard deviations appear to be lower for the results obtained from decay tests. Also, because in a fully mixed room all channels should give the same result, all the local mean age results can be combined, giving a global mean and standard deviation over 24 independent tests at each error level for both step-up and decay tests. These results are shown in table 6.

Table 6. SIMULATED DATA, Global Local Mean Ages (Mins)

Percentage Error	No of Tests	Step-Up Tests	Decay Tests
1	24	34.3± .4	33.4± .4
2	24	34.5± .7	33.5± .9
5	24	33.9±1.7	33.1±1.4
10	24	34.8±3.9	33.3±3.5

When evaluating the simulated data for the step-up tests, it was assumed at first that the value of the tracer gas concentration in the inlet duct, Cs, which is also the asymptotic value C (infinity) within the room, is known exactly (a value of 200 vpb was used). The effect of an error in this assumption was also explored, by re-calculating with different values of C (infinity). These results are shown in table 7 for local mean age, in which the global for 24 tests is shown. Table 8 gives room mean age for four sets of results

Table 7. SIMULATED DATA, Local Mean Ages (Mins)
Step-Up Tests, Effect of Error in C(infinity)

	Error on data				
C(inf)	0%	1%	2%	5%	10%
180	25.3±0.0	26.3± .2	26.5± .5	27.2±1.4	29.4±2.3
190	29.2±0.0	30.2± .3	30.3± .6	30.4±1.4	31.5±2.6
200	33.4±0.0	34.3± .4	34.5± .7	33.9±1.7	34.8±3.8
210	37.7±0.0	38.6± .4	38.8± .8	38.0±2.2	38.8±4.2
220	42.1±0.0	43.0± .5	43.2± .9	42.4±2.4	42.0±5.0

Table 8. SIMULATED DATA, Room Mean Ages (Mins)
Step-Up Tests, Effect of Error in C(infinity)

	Error on data				
C(inf)	0%	1%	2%	5%	10%
180	27.2± 0.0	29.6± .6	31.2± 1.4	28.5± 2.6	34.7± 4.7
190	29.5± 0.0	30.5± 1.5	32.9± 1.4	32.3± 2.2	40.7± 5.9
200	33.4± 0.0	34.2± 1.6	36.6± 1.5	32.6± 4.2	42.5±13.1
210	37.7± 0.0	38.5± 1.7	40.8± 1.6	36.5± 3.9	46.3±13.9
220	42.2± 0.0	42.9± 1.7	45.2± 1.6	40.8± 3.7	52.3±14.4

6. DISCUSSION

It must first be pointed out that the visual appearance of the tracer gas concentration versus time graphs, whether they originated from measured or simulated data, gave little indication of the likely quality of the results obtained from them. The graphs themselves, and the results, would always appear plausible. The exploration of the effect of error is therefore of some importance. Although limited to the case of a fully mixed room, the results from the simulated data show several interesting features. These are :

1. As errors in the data increase, step-up tests become less reliable than decay test.
2. The un-reliability of the step-up tests becomes more pronounced when the error in C (infinity) is also included.

3. Step-up tests are only marginally worse than decay tests when evaluating local mean ages, but are much worse when evaluating room mean age.

It is difficult to draw similarly precise conclusions from the results of the experimental data because at least some of the variability is due to turbulence in the model as well as small differences in experimental conditions between tests. Nevertheless, the results tend to support the conclusions drawn from the simulated data. The poor results of the first set of step-up tests confirm the importance of accurate control and measurement of the tracer concentration in the inlet duct. Comparison of the second set of step-up tests with the pulse and decay tests shows again that when measuring local mean age, step-up is marginally worse than the other methods, but when measuring room mean age, step-up is much worse.

The poor performance of step-up tests that have been found here may be partly explained by the difficulty in estimating the end correction. Even though tracer gas measurements were continued well beyond one time constant, the end correction was often of similar order of magnitude to the principal measurement of area. This is illustrated in table 9.

Table 9. Average size of end correction expressed as a Percentage of the Measured Area

Method	No of Tests	Ch1	Ch2	Ch3	Ch4	Ch5	Ch6
				Fan Speed 1			
Pulse	4	62.3	45.0	54.5	64.7	48.6	48.0
StepUp89	3	49.5	74.4	49.8	115.5	25.1	92.0
StepUp90	8	0.0	128.8	52.8	58.0	43.0	59.5
Decay	5	79.4	67.2	73.0	72.6	74.4	78.3
				Fan Speed 2			
Pulse	3	8.9	11.6	9.2	19.5	8.0	6.0
StepUp89	3	273.1	26.1	10.5	43.5	5.3	103.3
StepUp90	5	0.0	37.3	27.6	21.6	13.2	24.0
Decay	6	9.1	8.2	8.8	9.3	7.8	8.7
				Fan Speed 3			
Pulse	3	10.8	25.2	13.9	23.8	8.7	9.1
StepUp89	4	162.8	6.5	7.3	62.5	3.4	93.0
StepUp90	5	0.0	23.4	21.1	19.8	16.2	45.1
Decay	4	23.0	18.0	25.4	25.2	20.3	21.0

7. CONCLUSIONS

The original objective was to determine whether or not the decay method is more or less reliable than other methods in determining local and room mean ages. The results suggest that the decay method may be slightly better than the pulse and step-up techniques and it was found to be the easiest method to use from a practical point of view. It was also discovered that the step-up method can give rise to large errors, due to uncertainties in the value of C (infinity) and the effect that the end corrections can have upon the final answers. These errors were especially predominant in the measurement of the room mean age. measurement of room mean age.

8. REFERENCES

1. Sutcliffe, H.C. : "A Guide to Air Change Efficiency", AIVC, Technical Note 28, February 1990.

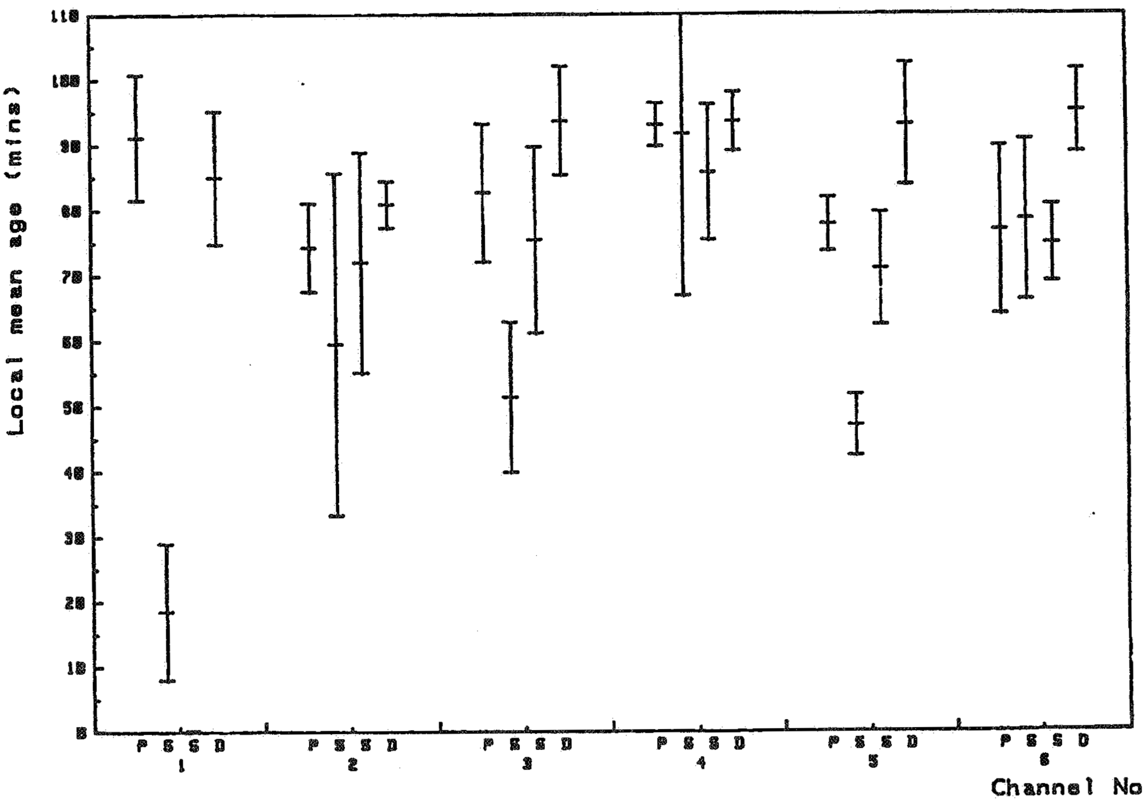

Figure 2. Local mean ages(mins), Fan Speed 1

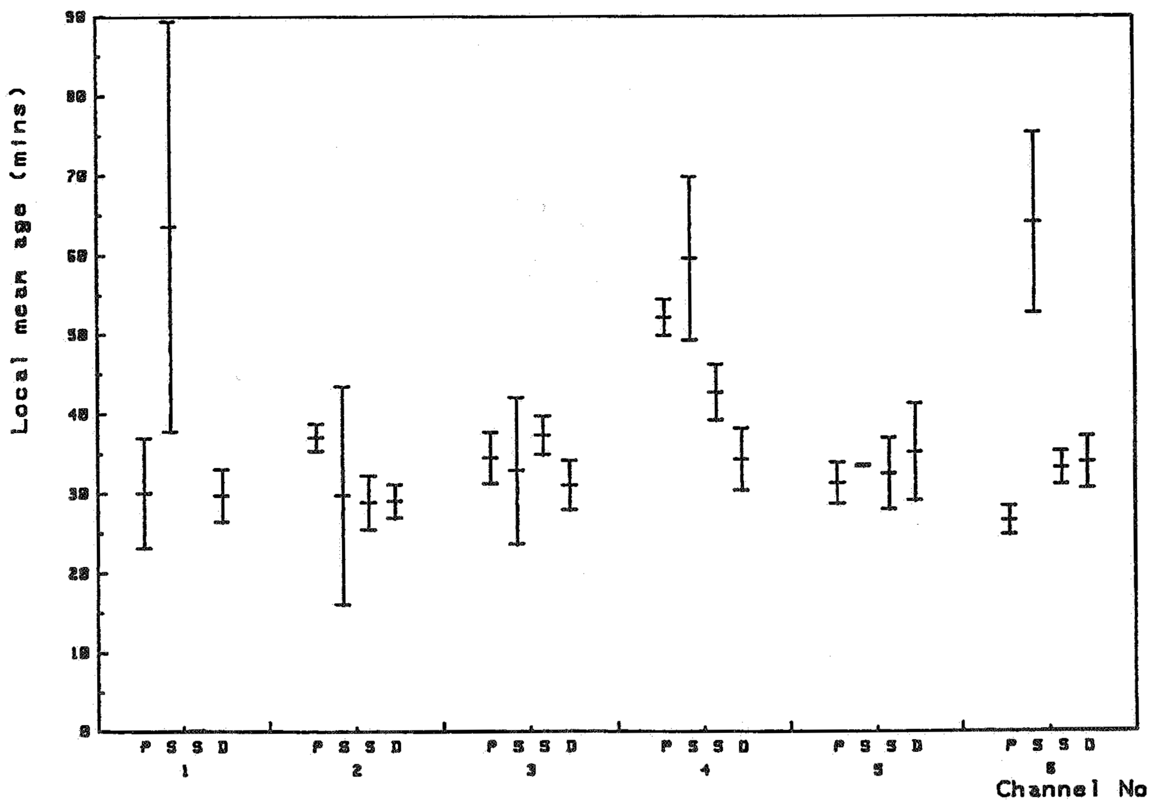

Figure 3. Local mean ages(mins), Fan Speed 2

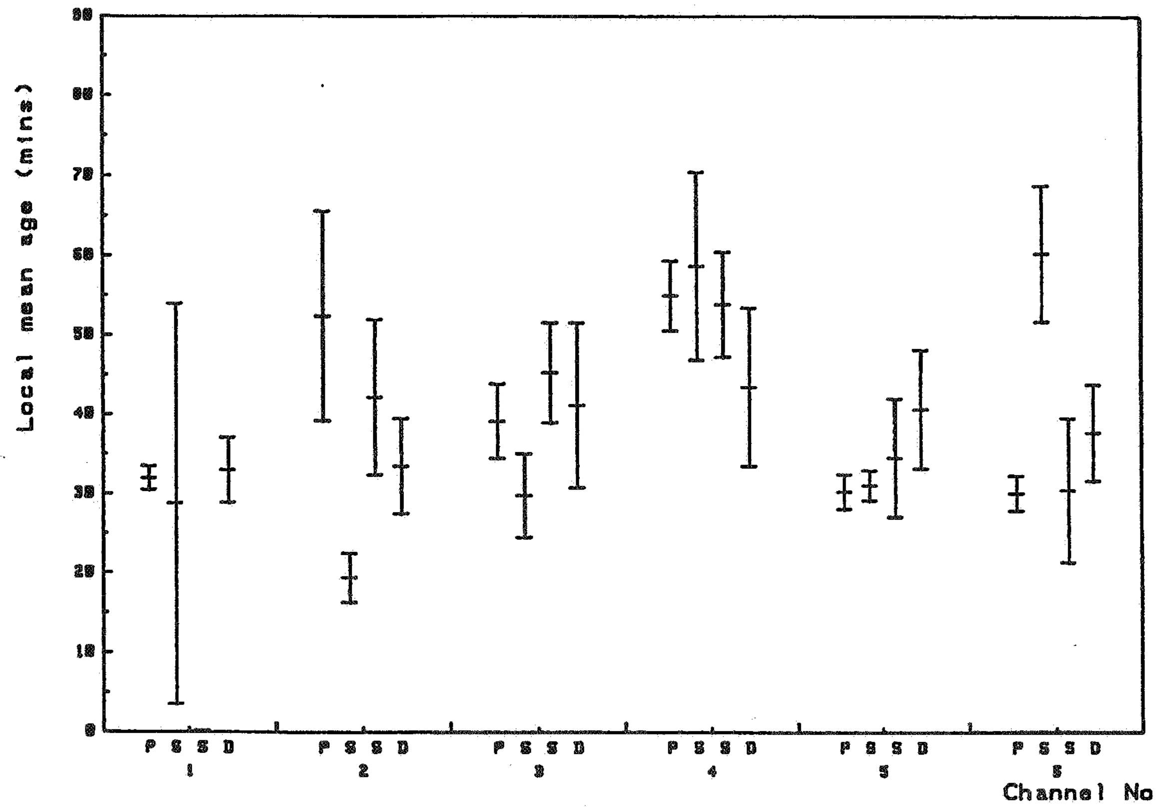

Figure 4. Local mean ages(mins), Fan Speed 3

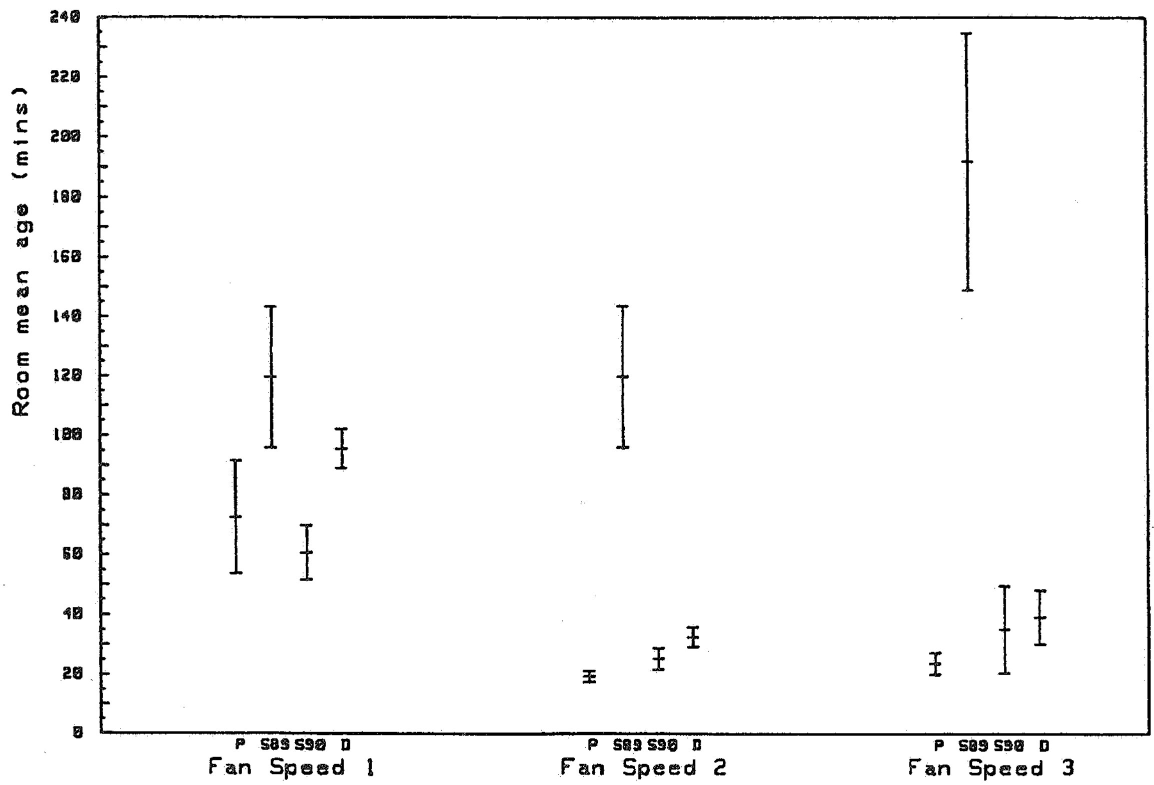

Figure 5. Room mean age (mins)

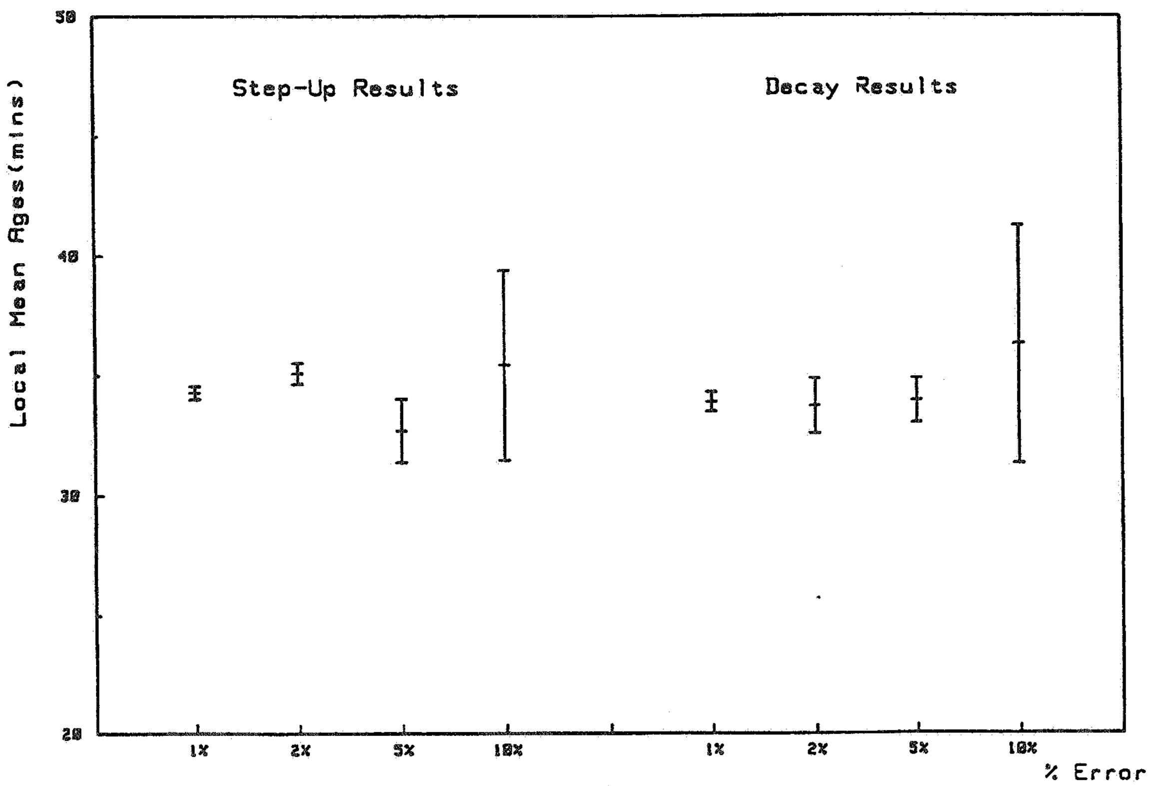

Figure 6. Local mean age(mins), Channel 5

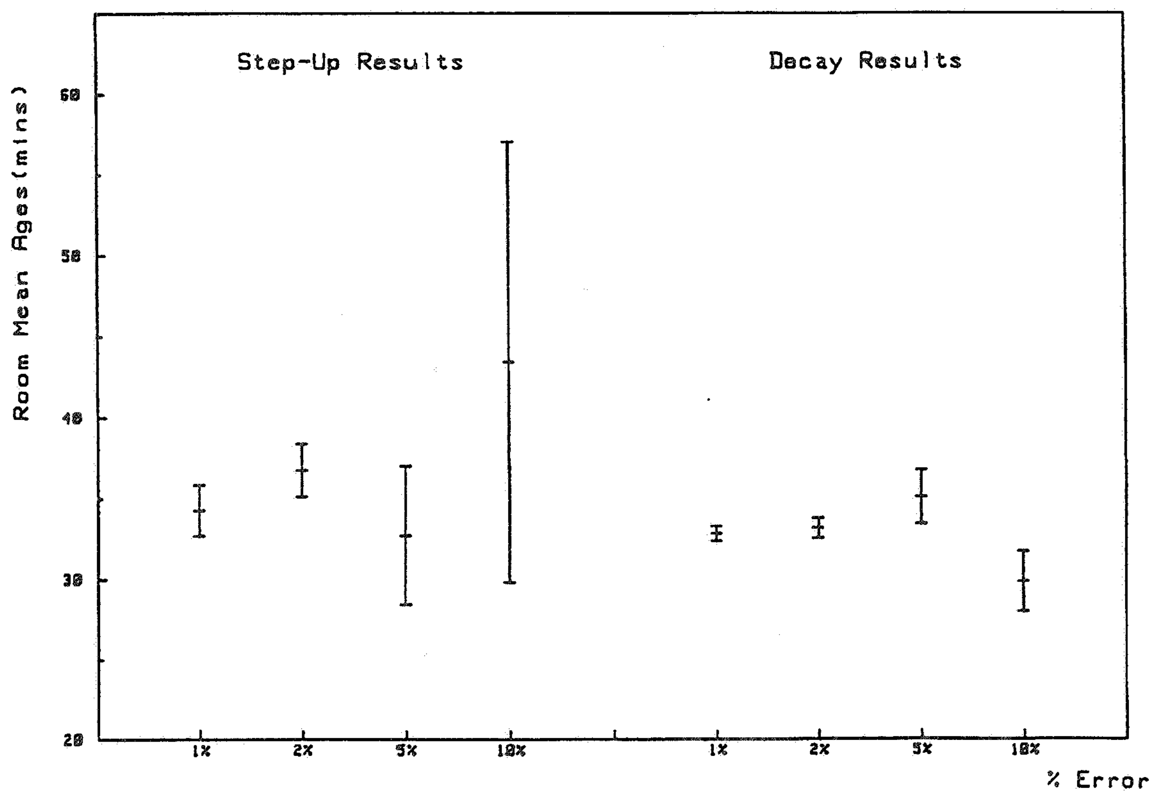

Figure 7. Room mean age(mins)

VENTILATION SYSTEM PERFORMANCE

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FIELD MEASUREMENTS OF VENTILATION AND VENTILATION EFFECTIVENESS IN AN OFFICE/LIBRARY BUILDING

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ABSTRACT

Mechanical ventilation system performance involves the provision of adequate amounts of outdoor air, uniform distribution of ventilation air within the occupied space, and the maintenance of thermal comfort. Standardized measurement techniques exist to evaluate thermal comfort and air exchange rates in mechanically ventilated buildings; field techniques to evaluate air distribution or ventilation effectiveness are still being developed. This paper presents field measurements of air exchange rates and ventilation effectiveness in an office/library building in Washington, DC. The tracer gas decay technique was used to measure whole building air exchange rates. Ventilation effectiveness was investigated at several locations within the building through the measurement of local tracer gas decay rate and mean local age of air. The ventilation effectiveness measurements serve as an investigation of the applicability of the measurement procedures employed, providing insight into the measurement issue of establishing initial conditions, the spatial variation in tests results within a building, and the repeatability between tests.

1. INTRODUCTION

The James Madison Memorial Building of the Library of Congress, located in downtown Washington DC, has had a history of indoor air quality complaints by the building occupants. These complaints include stale, "dead" and uncomfortable air, poor air circulation, warm temperatures, and physical discomfort such as headaches, drowsiness, nausea, and sinus irritation. In an attempt to determine whether these complaints are related to the ventilation system performance and contaminant concentrations in the building, the Center for Building Technology of the National Institute of Standards and Technology (NIST) conducted an evaluation of the building's ventilation and air quality characteristics. This evaluation included measurements of building air exchange rates, ventilation effectiveness, and the concentrations of selected indoor air pollutants.

Indoor air quality complaints have become more common, or at least more publicized, in recent years. Whether or not air quality within office buildings has actually worsened, the awareness of building occupants and managers with respect to these problems has increased. Although there has been insufficient research to establish the causes of many indoor air quality complaints, several have been suggested including the reduction of building air exchange rates. In fact, air exchange rates and other ventilation system performance parameters have not been well characterized in mechanically ventilated office buildings. Therefore, general statements about air exchange rates in buildings, including trends over time, can not be supported [1].

The investigation of the Madison Building conducted by NIST included the evaluation of the ventilation system performance and the measurement of indoor contaminant concentrations. Whole building air exchange rates were measured with the tracer gas decay technique, using an automated measuring system. The assessment of ventilation effectiveness involved the measurement of local tracer gas decay rates and the mean local age of air. In addition, the concentrations of selected pollutants were measured in the building. This report presents the results of the air exchange rate and ventilation effectiveness measurements in the building. In addition, the relationship between the indoor carbon dioxide concentration and the building air exchange rate is examined. Additional information describing the building and preliminary results of the study are available in Reference 2.

2. BUILDING AND VENTILATION SYSTEM DESCRIPTION

The Madison Building was constructed during the 1970s and first occupied in 1979. It is a nine-story building with two basement levels and a ground level that is partially below grade. The building contains primarily office space, along with several other facilities including meeting rooms, auditoriums, library material storage areas, a print shop and preservation laboratory on the ground level, a loading dock on the ground level, and an underground garage. There is a tunnel on the ground floor connecting the Madison Building to the other buildings of the Library of Congress. The total floor area of the building is about 164,400 m^2.

The mechanical ventilation system of the Madison Building consists of 44 air handlers in the penthouse mechanical room and ten additional air handlers in four subbasement mechanical rooms. The portion of the building served by the penthouse air handlers constitutes the bulk of the building volume, while the subbasement air handlers serve only a small fraction of the building volume, most of which is unoccupied. Each floor of the building is divided into eight zones, and each of the eight building zones is associated with a bank of air handlers in the penthouse mechanical room. These zones are not isolated from each other in terms of airflow, with interior air being able to flow between these zones through hallways, from room to room, and within open office spaces that are in more than one zone.

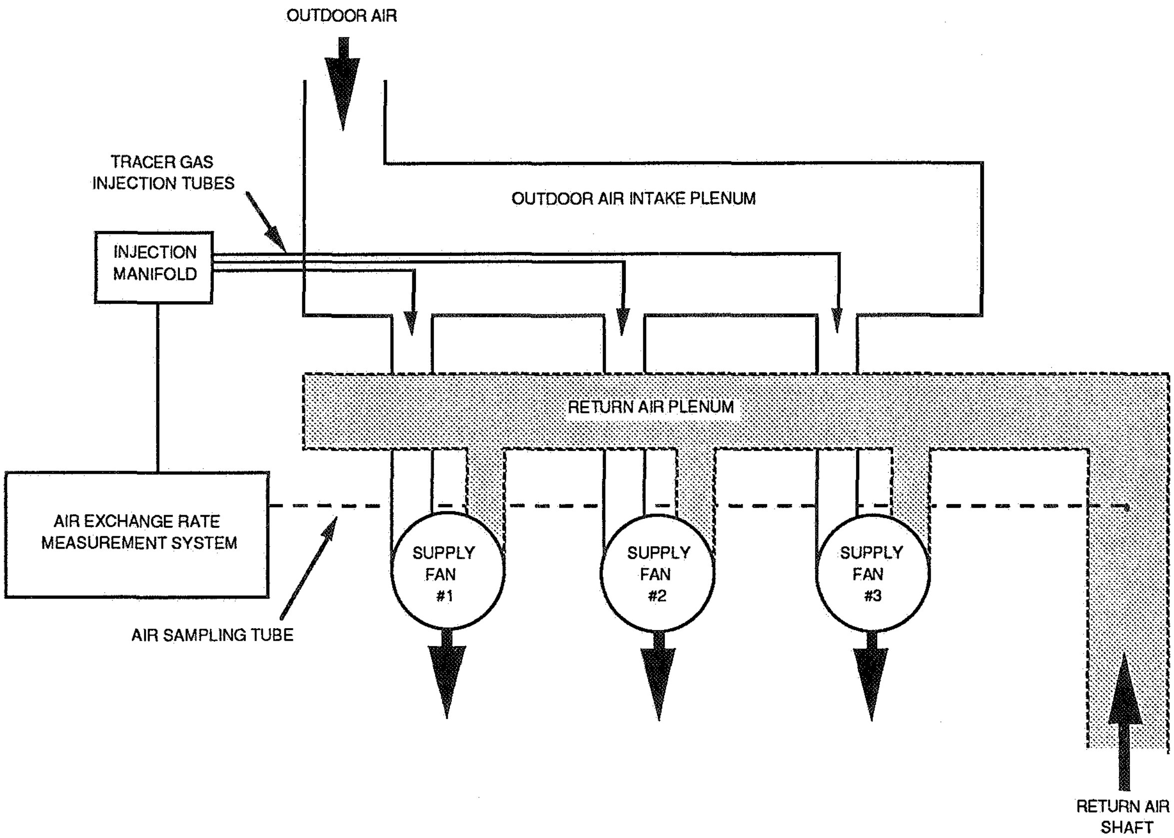

Figure 1 Schematic of Air Handling System

Figure 1 is a schematic of the air handling system for one of the eight building zones. Each zone is associated with an outdoor air intake plenum and a return air shaft. There are four to eight air handlers associated with each zone (only three are shown in the figure), with any given air handler serving from one to nine of the building floors. These air handlers all have variable air volume (VAV) supply fans and maintain constant outdoor air intake rates through the control of dampers in the

3. DESCRIPTION OF MEASUREMENTS

There are many factors related to the performance of building ventilation systems, but standardized measurement techniques and procedures exist to evaluate only some of these performance parameters in the field. Of particular interest is the development of measurement techniques to quantify the uniformity of air distribution or ventilation effectiveness in mechanically ventilated office buildings. Existing techniques to measure ventilation effectiveness have been successfully applied in laboratory facilities and in some field applications, but experience in modern, North American office buildings is limited. In this investigation, whole building air exchange rates were measured using the tracer gas decay technique, a standardized procedure. Ventilation effectiveness was evaluated using the tracer gas decay technique to measure the local tracer gas decay rate and the mean local age of air.

3.1 Whole Building Air Exchange Rates

Whole building air exchange rates were measured in the Madison Building using the tracer gas decay technique. This procedure has been used in thousands of residential buildings and many office buildings [1,4] and is described in ASTM Standard E741 [5]. The tracer gas decay technique is used to determine the rate at which outdoor air enters a building, including both intentional outdoor air intake through the air handling systems and unintentional infiltration through leaks in the building envelope. Regardless of common design expectations of minimal infiltration rates in mechanically ventilated office buildings, these two components of air exchange can be comparable in magnitude [6,7]. The air exchange rate of a building depends on a variety of factors including the design, installation and operation of the mechanical ventilation system and its controls, the airtightness of the building envelope, the interior configuration of the building, and outdoor weather conditions. Due to the complexity of air exchange in mechanically ventilated buildings, it is necessary to make many air exchange rate measurements in order to understand the air exchange characteristics of a building.

The tracer gas measurements of air exchange rates in the Madison Building employed an automated measuring system that enables the collection of large amounts of data under a range of outdoor weather and building operation conditions. The automated measuring system has been used previously to provide continuous measurements of building air exchange rates [4] and employs sulfur hexafluoride (SF_6) as the tracer gas. The microcomputer-based system controls tracer gas injection and air sampling, records SF_6 concentrations, and monitors and records outdoor weather, indoor temperature and fan operation status. A gas chromatograph equipped with an electron capture detector is used to measure SF_6 concentrations in a range of about 5 to 300 parts per billion (ppb) with an accuracy of roughly 1%.

In tracer gas tests, the manner in which the tracer gas is injected into the building and the locations at which the tracer gas concentrations are measured are necessarily based on the layout of the building and its air handling systems. In the Madison Building both the tracer gas injection and the air sampling strategies are based on the division of the building into the eight zones. Tracer gas was injected into the eight outdoor air intake plenums associated with the eight building zones, and the SF_6 concentration was monitored in each of the eight return air shafts and at an outdoor location. Figure 1 depicts the injection and sampling scheme for one building zone. A tracer gas injection tube carries a metered amount of tracer gas from the automated system to each of the eight outdoor air intake plenums, where the injection tube is connected to an injection manifold containing a flowmeter for each air handler in that outdoor air intake ducts of each fan. These intake dampers are modulated based on the output of an airflow monitor in the outdoor air intake duct. Supply air is delivered to the occupied space through ducts that run vertically through ventilation chases and horizontally through the plenums above the suspended ceilings on each floor. The return air from the occupied space flows into the suspended ceiling plenum on each floor through return air openings in the suspended ceiling. This return air then flows through the plenum and into the vertical return air shafts. The return shaft of each zone is connected to a return air plenum in the penthouse that serves the air handlers for that zone, enabling the recirculation of return air. There are no return fans in the building and no provisions for spilling excess return air, therefore all of the return air is recirculated. The air handling systems in the Madison Building operate 24 hours a day, throughout the year. There is a nighttime setback in the supply air static pressure setpoint, but the outdoor air intake rate is constant.

The air handling systems in the Madison Building are different from the systems that are typically employed in modern, North American office buildings. In order to protect library materials from degradation by outdoor air pollutants, the outdoor air brought into the buildings is cleaned and filtered. In order to limit the infiltration of unfiltered outdoor air, the outdoor air intake rate is carefully controlled to maintain the building at a positive pressure relative to the outdoors. The ventilation systems are operated at the same outdoor air intake rate 24 hours a day in order to provide constant protection of the library materials. In typical office building ventilation systems, outdoor air intake rates are not monitored and controlled as carefully as they are in the Madison Building. The ventilation systems in most office buildings are designed to bring in minimum levels of outdoor air during very cold and very hot weather to reduce the space conditioning load. Larger amounts of outdoor air are brought in during mild weather for cooling, employing a so-called economizer cycle. Therefore, in typical office buildings the air exchange rate varies by a factor of 5 or more depending on the outdoor weather, time of day and season of the year. In addition, office building ventilation systems are generally shut down during evenings and weekends.

The supply airflow rate capacity for the Madison Building's air handlers is about 850 m^3/s, and the minimum outdoor air intake rate is 170 m^3/s. These airflow rates can be converted to air changes per hour (ach) by dividing them by the building volume. Based on the gross building volume (the gross floor area of the building multiplied by the ceiling height, including the height of the return air plenum) the supply airflow rate capacity corresponds to about 5 ach, and the design outdoor air intake rate corresponds to 1.05 ach. The actual interior volume of the building is less than the gross volume due to the volume associated with interior partitions, furniture and other items. The volumetric airflow rates should be divided by this lower volume, which will increase the corresponding air change rates. An appropriate factor by which to reduce the gross volume is not available, but the correction to the air change rates is probably no more than 10 or 20%. ASHRAE Standard 62 [3] recommends a minimum ventilation rate in office space of 10 L/s per person. This value can be converted to air changes per hour by dividing by the volume associated with a single person. Assuming an occupant density in office space of 7 people per 100 m^2 (the default value contained in ASHRAE Standard 62) and a ceiling height of 3.5 m, 10 L/s per person converts to 0.72 ach. This conversion should also be corrected to account for the volume occupied by interior furnishings, but as stated above the correction is probably not large. The outdoor air intake rate specified in the mechanical ventilation system design for this building is almost 50% above the recommendation in ASHRAE Standard 62-1989.

zone. An injection line runs from the outlet of each flowmeter to each air handler in the zone. Thus, when tracer gas is injected into one of the eight zones, it is released into all the air handlers of that zone for the same length of time, at flow rates that are based on the volume served by each individual air handler.

Tracer gas was injected into 39 of the the 44 penthouse air handlers every three hours at a rate that was based on achieving an initial concentration of about 150 ppb in the building. After the injection, the tracer gas concentration was monitored at the nine air sample locations, with each location being sampled once every ten minutes. With the building fans operating 24 hours a day, eight tracer gas decay tests were conducted each day. The tracer gas concentration data were analyzed to determine the decay rate for each of the eight returns, and these eight decay rates were averaged to estimate the whole building air exchange rate. The accuracy of this air exchange rate determination depends on the uniformity of the tracer gas concentration within the building. The measurement error is estimated to be about 10%.

3.2 Ventilation Effectiveness

It is valuable to compare measurements of whole building air exchange rates to design values and ventilation standards, but these air exchange rates do not provide an indication of how this ventilation air is distributed within a building. Although the air exchange rate may be adequate on a whole building scale, there may be areas within the building with inadequate outdoor air supply due to nonuniform ventilation air distribution among the spaces within the building or due to poor mixing of this ventilation air within these spaces. Nonuniform air distribution, i.e., the existence of rooms or locations within rooms that are less well ventilated than other portions of the building, have been suspected as being responsible for some air quality complaints. There are no standardized measurement procedures for quantifying the uniformity of air distribution or ventilation effectiveness in mechanically ventilated office buildings. Therefore, the effects of nonuniformities in air distribution on air quality have not been demonstrated. However, based on the potential importance of air distribution, ventilation effectiveness measurement techniques for field application are being developed and studied [8-11].

Ventilation effectiveness was evaluated in the Madison Building in two series of tests, the first presented in Reference 2 and the second described in this paper. These evaluations consisted of measurements of local tracer gas decay rate and mean local age of air. In order to make these measurements, a uniform tracer gas concentration is established throughout the building. The first series of tests employed a pulse injection of tracer gas followed by a mixing period to achieve these initial conditions. In the second series of tests, tracer gas was injected at a constant rate until the concentration in the building attained equilibrium. A series of air samples was then taken at selected locations within the occupied space during the subsequent tracer gas decay. In order to assess local tracer gas decay rates, the tracer gas concentrations at each location are fit to an equation of the form:

$$C_i(t) = C_{oi}e^{-\lambda_i t} \tag{1}$$

$C_i(t)$ is the concentration at location i measured at time t, and C_{oi} and λ_i are determined from the curve fit. C_{oi} is the calculated the tracer gas concentration location i at t=0, though in general it will not equal the measured concentration at t=0 as explained below. In the first series of tests, t=0 corresponds to the time at which the pulse injection is complete. In the second series of tests, t=0 corresponds to the time at which the constant injection of tracer gas is stopped. λ_i is the tracer gas decay rate at the location in units of air changes per hour, although it is not generally equal to the air exchange rate at the location being tested. The value of λ_i is equal to the building air exchange rate only when the tracer gas concentration is uniform throughout the building during the decay. Based on multi-zone building airflow theory, the tracer gas concentration at all locations in the building will decay according to Equation 1 after a sufficient length of time (assuming that none of the locations are in spaces do not exchange air with the rest of the building). Regardless of the degree of mixing within the building, the values of λ_i will eventually be the same throughout the building and the values of C_{oi} will vary within the building [12]. C_{oi} is therefore a calculated concentration that characterizes a particular location and does not correspond to the actual initial concentration, which should be the same throughout the building. The calculated value of C_{oi} will be higher at locations that are less well ventilated and can be considered an indicator of ventilation effectiveness. There are no straightforward relationships between the values of C_{oi} and the airflow rates within a building except in very simple situations [13]. In real buildings, the values of C_{oi} can only be used as a qualitative indicator of ventilation effectiveness, with higher values corresponding to poor ventilation air distribution. There are some practical considerations regarding the use of C_{oi} as a measure of ventilation effectiveness. If one does not wait long enough before fitting the data to Equation (1), the values of λ_i will vary among locations and C_{oi} may not be a useful indicator of ventilation effectiveness. Also, variation in C_{oi} can occur due to a nonuniform distribution of the initial tracer gas injection.

Measurements of the mean local age of air were also made in the building as a measure of ventilation effectiveness. Mean local age has been proposed for quantifying ventilation effectiveness and has provided useful results in both laboratory and field tests [8,10]. However, complications exist when applying the measurement procedures in large, mechanically ventilated office buildings [9], and research is needed to examine the applicability of this approach in the field. The mean local age of air at a location i within a building τ_i is defined as the average amount of time that has elapsed since the air at that location has entered the building. If the ventilation air within a building is uniformly distributed to all spaces and the air within each space is perfectly mixed, then the local age will be the same throughout the building and equal to the inverse of the building air exchange rate. The inverse of the building air exchange rate is defined as the nominal time constant of the building τ_n. If there is nonuniform ventilation air distribution within a building, then those locations with poor ventilation air distribution will have ages of air that are higher than the building average. There are several definitions of ventilation effectiveness based on comparisons of τ_i to τ_n or to the building average age of air. In this paper, the ventilation effectiveness ε_i is defined as τ_n divided by τ_i. When the ventilation air distribution is perfectly uniform, $\varepsilon_i = 1$ at all locations in the building. When there is nonuniform ventilation air distribution, locations in so-called stagnant zones, which are effectively bypassed by the ventilation air, will have local ages of air that are relatively large and values of ε_i significantly less than 1.

To measure the mean local age of air, one establishes a uniform tracer gas concentration within the building and monitors the decay in tracer gas concentration at specific locations. The mean local age of air at location i is then defined as:

$$\tau_i = \frac{1}{C_{oi}} \int_0^{\infty} C_i(t)dt \qquad (2)$$

Two techniques for evaluating this integral were employed in the Madison Building, the first based on an average concentration determined by slowly filling an air sample bag at a constant rate during the test. The integral was also determined numerically from the discrete tracer gas concentration measurements made during the decay.

Reference 2 reports on the first series of measurements of local tracer gas decay rate and mean local age of air at 56 locations in the Madison Building. In these tests the initial conditions of uniform tracer gas concentration were achieved by a pulse injection of tracer gas lasting about 5 minutes followed by a period of about 30 minutes for the tracer gas to mix with the interior air. After the mixing period, five air samples were collected at each test location at roughly 20 minute intervals. Ten tests were conducted, one in the morning and one in the afternoon on five consecutive days. During each test, five to seven locations were monitored. One location was included in eight of the ten tests to provide an indication of the repeatability of the test results.

Another series of ventilation effectiveness measurements was conducted in the building more recently. In these measurements, the initial conditions were achieved by injecting tracer gas into the building supply fans at a constant rate until the concentration within the building attained equilibrium. In these tests the tracer gas injection lasted for at least five hours. After equilibrium was established, six air samples were collected at selected locations within the occupied space at approximately 20 minute intervals, with the first air sample taken before the injection was stopped. These measurements were made at 22 locations within the building, with each location tested three times.

4. RESULTS AND DISCUSSION

4.1 Whole Building Air Exchange Rates

Whole building air exchange rates were measured in the Madison Building from the end of January 1989 through March 1990, with a total of about 1300 individual measurements. Figure 2 is a plot of the building air exchange rates measured during the day versus the indoor-outdoor air temperature difference. These data indicate that the building air exchange rates are essentially constant over a wide range of temperature difference. The mean daytime air exchange rate is 0.82 ach with a standard deviation of 0.05 ach. The nighttime air exchange rates are similarly constant and have a mean of 0.76 ach with the same standard deviation. These standard deviations are less than the measurement uncertainty of 10%. Therefore, the outdoor air intake controls are performing as intended, i.e., the building air exchange rate is constant. There are very small variations over the day that appear to be related to the total supply airflow rate, with higher supply airflow rates corresponding to slightly higher air exchange rates.

Figure 3 is a plot of the daytime air exchange rates against Julian day, showing a slight decrease in air exchange rate over the measurement period. The variation in air exchange rate is not large relative to the measurement uncertainty of roughly 10%. This figure shows the minimum ventilation recommendation in ASHRAE Standard 62-1989, i.e., 10 L/s per person, corresponding to an air exchange rate of 0.72 ach, and the building's design minimum outdoor air intake rate, 1.05 ach. All of the measured air exchange rates are below the design value, and almost all are above the ASHRAE recommendation. These air exchange rates are similar in magnitude to those measured in other U.S. office buildings [1].

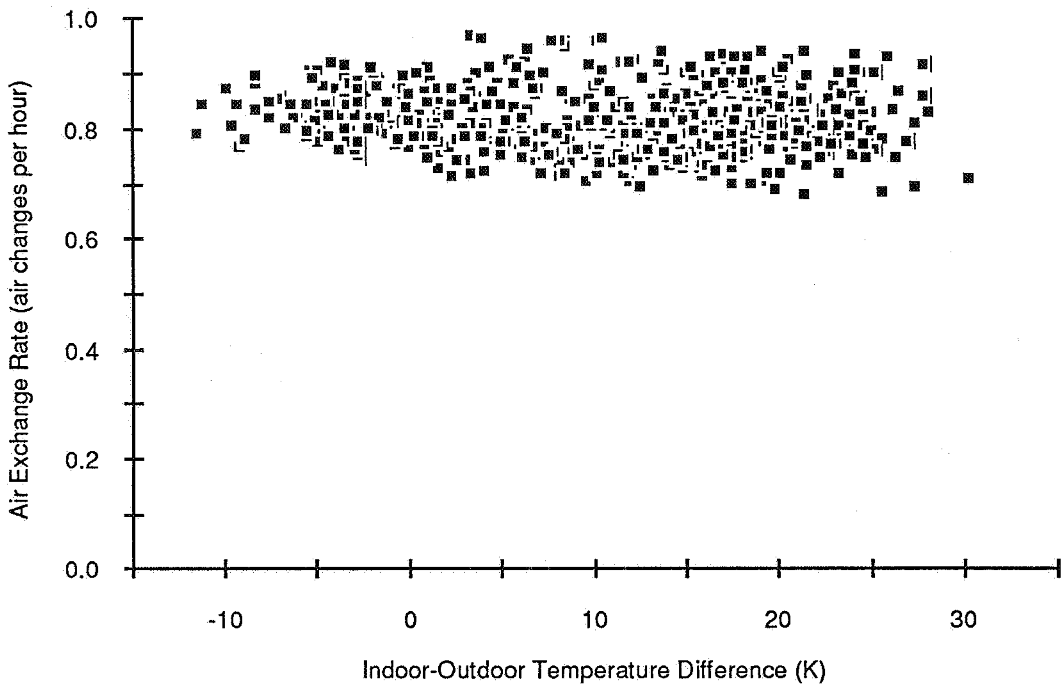

Figure 2 Daytime Air Exchange Rate versus Temperature Difference

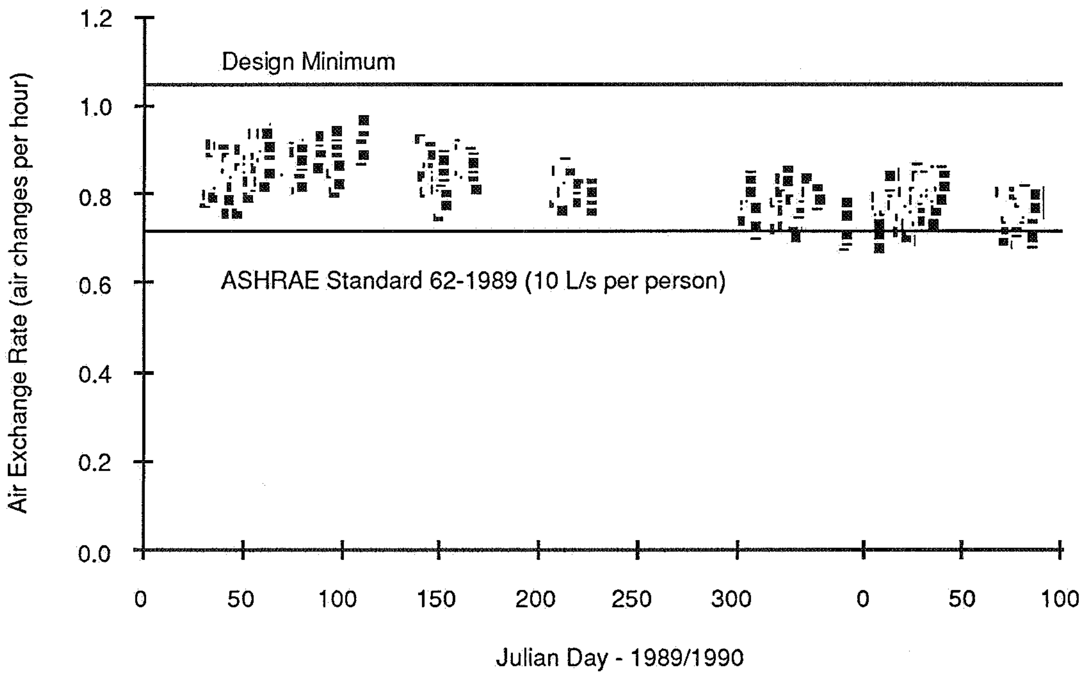

Figure 3 Daytime Air Exchange Rate versus Julian Day

4.2 Ventilation Effectiveness

Given the current status regarding the measurement of ventilation effectiveness in mechanically ventilated buildings, the ventilation effectiveness measurements in the Madison Building must be considered an investigation of the practicality of making these measurements in the field and the usefulness of results. In addition to providing insight into the applicability of the procedures, the measurements in the Madison Building enable the examination of three issues regarding the evaluation of ventilation effectiveness. Another issue addressed by both series of tests is the spatial variation in ventilation effectiveness measurement results, since this variation provides an indication of the uniformity of ventilation air distribution within a building. However, because there have been few field measurements of ventilation effectiveness, it is difficult to relate the magnitude of spatial variation to the uniformity of air distribution. Finally, measurements were made eight times at a single location in the first series of tests and three times at each location in the second series. Therefore, these results provide information on the repeatability of the test results.

4.2.1 Results of Previous Measurements

The results of the first series of ventilation effectiveness measurements are presented in detail in Reference 2. During each test, the local decay rate and the local age of air was measured at five to seven locations within the occupied space. The results of each test include the whole building tracer gas decay rate λ, the calculated value of the initial tracer gas concentration C_o averaged over the eight building returns, and calculated values of the initial concentration C_{oi}, the tracer gas decay rate λ_i, the mean local age of air τ_i, and the ventilation effectiveness ε_i at each location.

The ratio of the local tracer gas decay rate to the whole building decay rate, i.e., λ_i/λ, was calculated for each location. According to multi-zone building airflow theory, this ratio will equal one at all locations in the building after a sufficient length of time [12]. The average of these ratios for all the tests is 0.99 with a standard deviation of 0.05. Given that the local decay rates are basically uniform, a value of the ratio of the local C_{oi} to the whole building C_o that is close to 1.0 indicates good ventilation air distribution at this location. Those locations with poor mixing of the ventilation air will have a value of C_{oi} that is greater than C_o. The average value of C_{oi}/C_o for all the tests is 0.94 with a standard deviation equal to 0.09. Due to a lack of experience with these measurements, it is not possible to relate the magnitude of the variation in C_{oi} within the building to the uniformity of air distribution, but the results are consistent with good ventilation effectiveness. The predominance of values of C_{oi} less than C_o and the variation in C_{oi} throughout the building could also be caused by nonuniformities in the initial tracer gas concentration within the building. The eight tests conducted at the same location in the building provide an indication of the repeatability of the test results. The standard deviation of C_{oi}/C_o for the eight measurements at this location is 5% of the mean. Therefore, the variation in C_{oi}/C_o within the building is somewhat larger than the variation at a single location.

The mean local age of air was determined at the same locations at which the local tracer gas decay rates were measured. Two procedures were used, the first based on the average tracer gas concentration at each test location determined by filling an air sample bag at a constant rate during the test. The second determination was based on a curve fit to the tracer gas concentrations measured every 20 minutes during the test. If the air within the building was perfectly mixed and the measurement results had no errors, then the values of τ_i would be the same throughout the building and their inverses equal to the whole building air exchange rate. The ventilation effectiveness ε_i (equal to τ_n/τ_i) will then equal one throughout the building. Given the simultaneous measurement of τ_i at several locations in the building, the magnitude of the standard deviation of the values of τ_i at the various measurement locations relative to their mean value serves as a measure of the uniformity of ventilation air distribution. For the ten tests, the ratio of the standard deviation of τ_i to its mean ranged from 2% to 7%, with an average of 4%. The ventilation effectiveness values for all the tests range from 0.94 to 1.23, with an average value of 1.01, indicative of uniform ventilation air distribution. The spatial variation in the ventilation effectiveness ε_i is identical to the variation in τ_i. As in the case of the local decay rate measurements discussed above, there is insufficient experience with ventilation effectiveness measurements to determine how these results relate to the uniformity of air distribution within the building. The eight tests conducted at a single location provide an indication of the repeatability of the test results. The standard deviation of ε_i at this location is 5% of the mean value when determined from the average concentrations and 4% when determined numerically. Therefore, the variation in ε_i throughout the building is similar in magnitude to the variation at this single location.

It is difficult to interpret the results of the first series of ventilation effectiveness measurements in the Madison Building due to a lack of measurements in other buildings for comparison. The predominance of values of ε_i close to one is indicative of uniform air distribution in the building. The variation in both C_{oi}/C_o and ε_i throughout the building is similar in magnitude to the variation in repeated measurements at the same location. While no firm statements can be made regarding the uniformity of air distribution, these results are consistent with uniform ventilation air distribution and good air mixing within the space. The uniformity of the tracer gas concentrations during the decays, both within the occupied space and within the return air shafts, is also consistent with good ventilation effectiveness.

4.2.2 Results of Second Series of Measurements

In the second series of tests three measurements were conducted at three groups of locations, and the results of the nine tests are presented in the Table 1. The first column in the table gives the test location with the first number in the location designation indicating the building floor. The next two columns contain the initial concentrations at each location. C'_{oi} is the concentration measured just before the injection was stopped, and C_{oi} is the concentration determined from a curve fit to the five concentrations measured during the tracer gas decay (Equation 1). C_{oi} was evaluated from this curve fit at the time the injection was stopped. The local decay rate λ_i in air changes per hour, given in the fourth column of the table, is also determined from this curve fit. The inverse of λ_i is also included in the table. The local age of air in hours is presented in the last two columns of the table and is determined in two ways. τ'_i is based on the measured value of the initial concentration C'_{oi}, and τ_i is based on the calculated value C_{oi}. The results of each test are summarized for each of these parameters as the mean value for all locations in the test, the standard deviation and the ratio of the standard deviation to the mean. The table also shows the measured whole building air exchange rate for each of the tests.

Table 2 presents the measured values of ventilation effectiveness ε_i for the tests, where ε_i is defined here as τ_n/τ_i. The value of τ_i is used in this definition rather than τ'_i, because C_{oi} provides a more reliable estimate of the initial tracer gas concentration than C'_{oi}. The measured values of ε_i range from 0.85 to 1.13 with an average value of 0.98, indicative of uniform ventilation air distribution. The measured values of ε_i are summarized in Table 2 along with the mean, standard deviation and their ratio for each test. The mean and standard deviation of ε_i are also given for each location.

As discussed above, these experiments enable the examination of the establishment of initial conditions, the spatial variation in the test results and repeatability of the results at a single location. The measurement of local tracer gas decay rates and mean local ages of air using the tracer gas decay technique requires the establishment of a uniform tracer gas concentration throughout the building being tested, and establishing these initial conditions is extremely challenging in the field. The approach taken and the subsequent success will be determined to a large degree by the layout of the building and its air handling systems, the manner in which the ventilation system operates and the experience of the person conducting the test. In the case of the Madison Building, the ability to obtain the desired initial conditions is greatly facilitated by the building air handlers operating 24 hours a day, the air exchange rate being constant, and the recirculation of large volumes of return air.

The spatial uniformity of the initial tracer gas concentration can be characterized by the ratio of the standard deviation of the initial concentrations at the various test locations to the mean concentration. The initial concentrations C'_{oi} can be measured directly at each location or determined by the curve fits to the concentrations C_{oi} measured at each location during the tracer gas decay. In the first series of tests, the concentration that was used for assessing the uniformity of the initial conditions is the concentration calculated at the time of the first air sample of the test. The ratios of the standard deviation to the mean initial concentration for the ten tests are as follows: 0.068, 0.073, 0.072, 0.079, 0.030, 0.102, 0.090, 0.074, 0.112 and 0.085. The average of these ratios is 0.078. In the second series of tests, the concentration used for assessing the uniformity of the initial conditions is that calculated at the time that the tracer gas injection was stopped. These calculated concentrations C_{oi} are in the third column of Table 1. The ratios of the standard deviation to the mean initial concentrations for these nine tests range from 0.075 to 0.139, and the average of these ratios is 0.101. Thus, the initial concentrations are somewhat less uniform for the constant injection tests than for the pulse injection tests. One reason for the difference is that tracer gas was injected into 39 air handlers in both series of tests, and despite much effort the balancing of the tracer gas injections among the air handlers was not perfect. This injection imbalance led to spatial variation in tracer gas concentration in the building after the pulse injection and during the so-called equilibrium conditions of the constant injection tests. About 30 minutes elapsed between the time of the pulse injections and the time of the first air samples, while only 10 to 20 minutes elapsed before the first samples in the constant injection tests. The different amounts of time for tracer gas mixing allowed the spatial differences in concentration to diminish more in the pulse tests than in the constant injection tests.

These results provide some insight into the problem of establishing a uniform tracer gas concentration within a building when conducting ventilation effectiveness measurements. In general, the pulse injection approach takes less time than constant injection. Depending on the mixing characteristics of the building, the pulse injection approach will take less than one hour for injection and mixing. The constant injection approach takes about four times the nominal building time constant τ_n (τ_n equals the inverse of the building air exchange rate) to reach equilibrium. Depending on the building air exchange rate, the time to reach equilibrium can take from two hours to more than twelve hours. The long time period to reach equilibrium will present a problem when the ventilation system operation varies over this time period. Another important factor for either injection procedure is the ability to balance the tracer gas injection among a building's air handlers. Either procedure requires that the tracer gas be injected into each air handler at a rate that results in approximately the same tracer gas concentration throughout the building. Adjusting or balancing the tracer gas injection rates for multiple air handlers can be an extremely difficult and time-consuming task, depending on the number of air handlers and the complexity of the ventilation system zoning in the building. In general, many iterations of injection, assessing the concentration response and adjusting the injection airflow rates will be required in a building before an adequately balanced injection is achieved. The process of setting the injection rates in the Madison Building took several weeks. In either injection procedure, small imbalances in the injection rates can be overcome during a mixing period. If mixing within the building is poor or the ventilation air distribution is nonuniform, the mixing period can make the tracer gas concentration less uniform in either procedure. The degree of mixing within the building is very important in establishing the desired initial conditions. If mixing is rapid and thorough, it will decrease the mixing time required in the pulse injection procedure, and the faster pulse injection approach should be used. If the mixing is slow and incomplete, the pulse injection procedure may be inappropriate and the constant injection procedure should be used with no mixing period.

The evaluation of ventilation effectiveness in a building is based in part on the spatial uniformity of the test results, as measured by the local age of air τ_i and ventilation effectiveness ε_i. Locations with poor ventilation effectiveness will generally be characterized by larger values of τ_i and smaller values of ε_i than locations with good air distribution. The measured values of these two quantities are presented in Tables 1 and 2 for the nine tests conducted in the Madison Building. In the case of the local age of air τ_i, the variation among the measurement locations for each test ranges from 3% to 10% with an average value of 5%. The variation in the ventilation effectiveness ε_i among the test locations is almost identical. Because there have been few field measurements of ventilation effectiveness, the significance of this amount of variation can not be related to the uniformity of air distribution or the extent of mixing within the building. Measurements in additional buildings are required to determine the magnitude of the measurement error associated with these quantities. Because the Madison Building appears to be characterized by good mixing and uniform air distribution, the spatial variation in these results may serve as a useful baseline for the magnitude of variation in the measured values of these quantities.

The repeatability of ventilation effectiveness measurements at any particular location is another important measurement issue. Three ventilation effectiveness measurements were made at each location in the second series of tests in order to provide some insight to this question. The last two columns of Table 2 present the mean and standard deviation of ε_i at each location for the three measurements. The variation among the three measurements at each location, about 5%, is similar in magnitude to the variation among the different locations in each test.

5. VENTILATION AND CARBON DIOXIDE CONCENTRATIONS

The maximum whole building CO_2 concentration was determined for each working day in the Madison Building. The mean daily peak concentration was 512 ppm, the standard deviation was 30 ppm, and the largest daily peak was 605 ppm. ASHRAE Standard 62-1989 recommends that the CO_2 concentration be maintained below 1000 ppm, and therefore the whole building average CO_2 concentrations in the Madison Building are well below the ASHRAE maximum.

In a building with constant occupancy, the daily maximum in the CO_2 concentration is related to the building air exchange rate. The relationship between CO_2 concentration and air exchange rate has been discussed extensively in Reference 14. This reference contains a plot of daily maximum CO_2 concentration versus daily average air exchange rate for three mechanically ventilated office buildings. This plot is reproduced in Figure 4 with the addition of the Madison Building data. The solid line in the figure is the equilibrium CO_2 concentration as a function of air exchange rate based on the following assumptions: an occupant density of seven people per 100 m^2 of floor area, a ceiling height of 3.5 m (including the return air plenum), an outdoor CO_2 concentration of 300 ppm, and a CO_2 generation rate of 5.2×10^{-6} m^3/s per person. The data in the figure deviate from the equilibrium curve because the CO_2 concentrations in these buildings do not attain equilibrium. This is due to the fact that CO_2 generation rates (proportional to occupancy levels) are not constant for sufficient lengths of time to attain equilibrium. The deviation between the measured peak CO_2 concentrations and the calculated equilibrium concentrations is greater at lower air exchange rates because it takes longer to reach equilibrium at lower air exchange rates. These data confirm the inappropriateness of using CO_2 concentrations to determine building air exchange rates. The Madison Building data are clustered closely together in this plot, because the air exchange rates in the building do not vary. The relation between CO_2 concentration and air exchange rate in the Madison Building is similar to that observed in other office buildings.

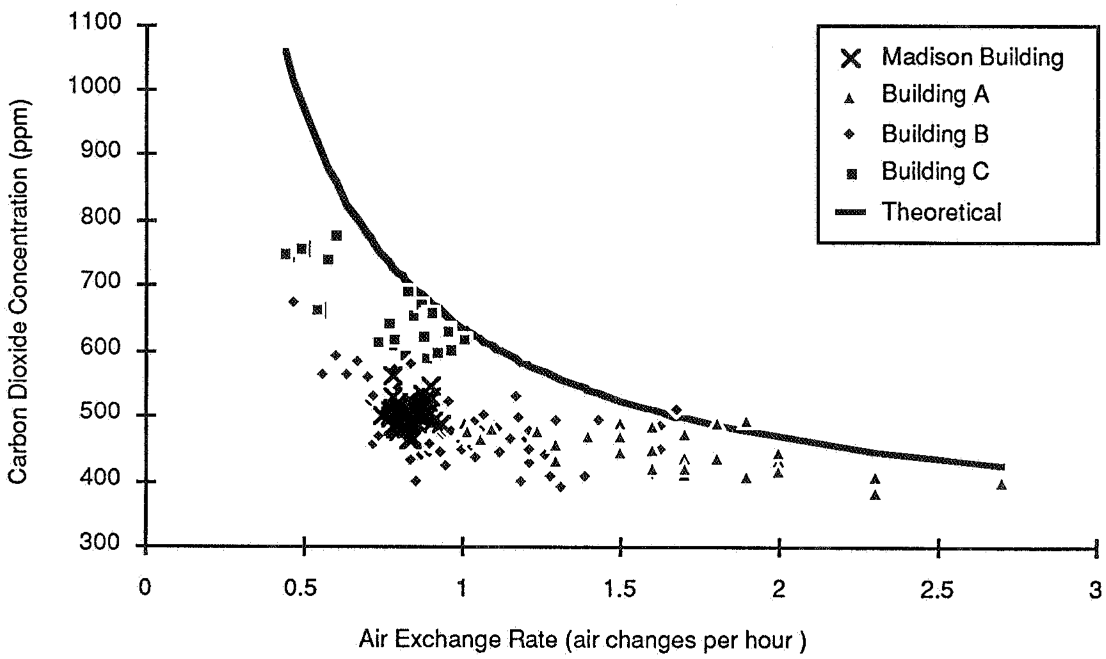

Figure 4 Carbon Dioxide Concentration versus Air Exchange Rate

6. SUMMARY

The ventilation assessment of the Madison Building study included the measurement of whole building air exchange rates for over a year and the application of two techniques to evaluate ventilation effectiveness. The whole building air exchange rates measured during occupied hours had an average value of 0.82 air changes per hour (ach) with little variation over the entire period of measurement. The air exchange rates measured at night were slightly lower. The daytime air exchange rates are generally above the ventilation recommendation in ASHRAE Standard 62-1989 (0.72 ach) and below the design air exchange rate for the building (1.05 ach). The measurements of whole building average CO_2 concentrations yielded an average value of the daily peak concentration of 512 ppm on working days, well below the recommended maximum in ASHRAE Standard 62-1989 of 1000 ppm. The relationship between CO_2 concentration and building air exchange rate is consistent with the relationship seen in other office buildings.

The results of the ventilation effectiveness measurements of local tracer gas decay rate and mean local age of air are consistent with good distribution of the outdoor air by the ventilation system and good mixing within the space. The average of the measured values of ventilation effectiveness is close to 1. The spatial variation in these quantities within the building is about 5%, as is the repeatability of the measurement results at the same location.

Valuable experience with the application of ventilation effectiveness measurement techniques was gained in this building. Even though several characteristics of this particular building made the measurements much easier than they might have been in a more typical office building, the measurements were still very involved and time consuming. The feasibility of measuring ventilation effectiveness and interpreting the results is in general problematic, and may very well be impractical in many buildings. Additional experience with the application of these measurement techniques in mechanically ventilated office buildings is needed in order to develop a reliable procedure for measuring ventilation effectiveness and the ability to interpret the results in terms of the adequacy of ventilation air distribution.

7. ACKNOWLEDGEMENTS

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Table 1 Ventilation Effectiveness Measurement Results

Test	Location	Initial Concentration (ppb) Measured C'_{oi}	Calculated C_{oi}	Local Decay Rate (hr $^{-1}$) λ_i	$1/\lambda_i$	Age of Air (hours) Based on C'_{oi} τ'_i	Based on C_{oi} τ_i
#1	Building Air Exchange Rate = 0.83/hr						
	6U4	54.1	50.5	0.79	1.27	1.19	1.27
	6V14	59.2	63.1	0.70	1.43	1.52	1.43
	6F12	56.8	58.9	0.83	1.21	1.26	1.21
	5D4	79.6	76.5	0.94	1.06	1.02	1.06
	5M8	65.2	64.2	0.82	1.22	1.20	1.22
	5S4	62.7	55.3	0.76	1.31	1.16	1.31
	5R13	58.3	55.7	0.76	1.32	1.27	1.32
Mean		62.2	60.6	0.80	1.26	1.23	1.26
Standard Deviation		7.8	7.8	0.07	0.10	0.14	0.11
Std Dev/Mean		0.13	0.13	0.09	0.08	0.11	0.08
#2	Building Air Exchange Rate = 0.81/hr						
	6U4	62.1	62.7	0.84	1.19	1.20	1.19
	6V14	72.1	71.4	0.78	1.28	1.26	1.27
	6F12	56.0	63.4	0.82	1.22	1.38	1.22
	5D4	74.9	80.9	0.87	1.15	1.24	1.14
	5M8	64.8	61.6	0.86	1.16	1.11	1.17
	5S4	57.8	55.0	0.79	1.27	1.21	1.27
	5R13	53.7	60.4	0.79	1.27	1.42	1.26
Mean		63.1	65.1	0.82	1.22	1.26	1.22
Standard Deviation		7.5	7.9	0.03	0.05	0.10	0.05
Std Dev/Mean		0.12	0.12	0.04	0.04	0.08	0.04
#3	Building Air Exchange Rate = 0.76/hr						
	6U4	72.4	68.1	0.72	1.39	1.31	1.39
	6V14	85.2	84.9	0.73	1.37	1.37	1.37
	6F12	67.1	77.0	0.69	1.45	1.66	1.45
	5D4	89.1	90.9	0.78	1.28	1.31	1.29
	5M8	80.9	67.3	0.77	1.30	1.08	1.30
	5S4	67.5	59.4	0.72	1.39	1.22	1.39
	5R13	64.4	68.3	0.69	1.45	1.55	1.46
Mean		75.2	73.7	0.73	1.38	1.36	1.38
Standard Deviation		9.1	10.3	0.03	0.06	0.18	0.06
Std Dev/Mean		0.12	0.14	0.04	0.04	0.13	0.04
#4	Building Air Exchange Rate = 0.74/hr						
	4X17	92.9	77.8	0.77	1.30	1.09	1.30
	4P12	63.3	66.7	0.72	1.39	1.47	1.39
	4W4	63.2	60.9	0.77	1.30	1.25	1.30
	4E7	77.7	77.5	0.83	1.20	1.20	1.20
	2D4	61.9	64.5	0.75	1.33	1.40	1.34
	2E7	72.5	61.3	0.76	1.32	1.12	1.32
	2G16	67.0	68.4	0.72	1.39	1.43	1.40
	2S16	57.9	61.4	0.70	1.43	1.51	1.42
Mean		69.6	67.3	0.75	1.33	1.31	1.33
Standard Deviation		10.6	6.5	0.04	0.07	0.15	0.07
Std Dev/Mean		0.15	0.10	0.05	0.05	0.12	0.05
#5	Building Air Exchange Rate = 0.80/hr						
	4X17	87.8	69.8	0.79	1.27	1.00	1.26
	4P12	63.9	64.5	0.73	1.37	1.39	1.38
	4W4	70.5	70.3	0.77	1.30	1.30	1.30
	4E7	79.0	81.0	0.83	1.20	1.23	1.20
	2D4	72.3	68.2	0.75	1.33	1.25	1.33
	2E7	57.0	60.6	0.73	1.37	1.45	1.37
	2G16	71.9	77.0	0.71	1.41	1.51	1.41
	2S16	69.8	70.3	0.74	1.35	1.36	1.35
Mean		71.5	70.2	0.76	1.33	1.31	1.33
Standard Deviation		8.6	6.0	0.04	0.06	0.15	0.06
Std Dev/Mean		0.12	0.09	0.05	0.05	0.11	0.05

Table 1 Ventilation Effectiveness Measurement Results (continued)

Test	Location	Initial Concentration (ppb) Measured C'_{oi}	Calculated C_{oi}	Local Decay Rate (hr^{-1}) λ_i	$1/\lambda_i$	Age of Air (hours) Based on C'_{oi} τ'_i	Based on C_{oi} τ_i
#6	Building Air Exchange Rate = 0.76/hr						
	4X17	87.6	76.6	0.84	1.19	1.05	1.20
	4P12	60.9	67.3	0.77	1.30	1.44	1.30
	4W4	68.7	70.0	0.83	1.20	1.23	1.21
	4E7	74.1	74.3	0.76	1.32	1.31	1.31
	2D4	62.5	68.6	0.78	1.28	1.41	1.28
	2E7	61.7	68.8	0.80	1.25	1.40	1.25
	2G16	71.1	84.5	0.84	1.19	1.42	1.19
	2S16	68.3	69.7	0.80	1.25	1.28	1.25
Mean		69.4	72.5	0.80	1.25	1.32	1.25
Standard Deviation		8.2	5.4	0.03	0.05	0.12	0.04
Std Dev/Mean		0.12	0.07	0.04	0.04	0.09	0.03
#7	Building Air Exchange Rate = 0.71/hr						
	3T17	75.6	73.8	0.70	1.43	1.40	1.44
	3G17	68.7	77.8	0.72	1.39	1.58	1.40
	3G11	77.1	79.9	0.71	1.41	1.47	1.42
	3E9	84.2	82.4	0.77	1.30	1.27	1.29
	3G5	77.7	92.7	0.72	1.39	1.67	1.40
	1K4	79.6	78.9	0.70	1.43	1.41	1.42
	1H17	62.8	66.6	0.70	1.43	1.52	1.43
Mean		75.1	78.9	0.72	1.40	1.47	1.40
Standard Deviation		6.6	7.4	0.02	0.04	0.12	0.05
Std Dev/Mean		0.09	0.09	0.03	0.03	0.08	0.03
#8	Building Air Exchange Rate = 0.73/hr						
	3T17	71.2	72.5	0.56	1.79	1.81	1.78
	3G17	78.7	83.9	0.63	1.59	1.70	1.60
	3G11	82.8	89.3	0.74	1.35	1.46	1.35
	3E9	78.6	82.5	0.74	1.35	1.42	1.35
	3G5	84.5	94.9	0.65	1.54	1.73	1.54
	1K4	71.7	81.1	0.72	1.39	1.56	1.38
	1H17	67.0	77.0	0.65	1.54	1.76	1.53
Mean		76.4	83.0	0.67	1.51	1.63	1.50
Standard Deviation		6.0	6.9	0.06	0.15	0.14	0.15
Std Dev/Mean		0.08	0.08	0.09	0.10	0.09	0.10
#9	Building Air Exchange Rate = 0.77/hr						
	3T17	62.0	68.2	0.74	1.35	1.48	1.35
	3G17	67.3	75.6	0.81	1.23	1.39	1.24
	3G11	71.2	75.2	0.79	1.27	1.33	1.26
	3E9	65.6	63.7	0.76	1.32	1.28	1.32
	3G5	68.5	78.2	0.73	1.37	1.56	1.37
	1K4	61.4	63.8	0.74	1.35	1.41	1.36
	1H17	59.1	62.1	0.73	1.37	1.43	1.36
Mean		65.0	69.5	0.76	1.32	1.41	1.32
Standard Deviation		4.0	6.2	0.03	0.05	0.09	0.05
Std Dev/Mean		0.06	0.09	0.04	0.04	0.06	0.04

Table 2 Summary of Ventilation Effectiveness Measurements

Location	Ventilation Effectiveness, ε_I			Mean	Standard Deviation
	Test #1	Test #2	Test #3		
6U4	0.95	1.04	0.95	0.98	0.05
6V14	0.85	0.97	0.96	0.93	0.07
6F12	1.00	1.01	0.91	0.97	0.06
5D4	1.13	1.08	1.02	1.08	0.06
5M8	0.99	1.06	1.01	1.02	0.03
5S4	0.92	0.97	0.95	0.95	0.03
5R13	0.91	0.98	0.90	0.93	0.04
Mean	0.96	1.02	0.96		
Standard Deviation	0.09	0.04	0.05		
Std Dev/Mean	0.09	0.04	0.05		
	Test #4	Test #5	Test #6		
4X17	1.04	0.99	1.10	1.04	0.05
4P12	0.97	0.91	1.01	0.96	0.05
4W4	1.04	0.96	1.09	1.03	0.06
4E7	1.13	1.04	1.00	1.06	0.06
2D4	1.01	0.94	1.03	0.99	0.05
2E7	1.02	0.91	1.05	1.00	0.07
2G16	0.97	0.89	1.11	0.99	0.11
2S16	0.95	0.93	1.05	0.98	0.07
Mean	1.01	0.94	1.05		
Standard Deviation	0.06	0.05	0.04		
Std Dev/Mean	0.06	0.05	0.04		
	Test #7	Test #8	Test #9		
3T17	0.98	0.77	0.96	0.90	0.12
3G17	1.01	0.86	1.05	0.97	0.10
3G11	0.99	1.01	1.03	1.01	0.02
3E9	1.09	1.01	0.98	1.03	0.06
3G5	1.01	0.89	0.95	0.95	0.06
1K4	0.99	0.99	0.95	0.98	0.02
1H17	0.98	0.90	0.95	0.95	0.05
Mean	1.01	0.92	0.98		
Standard Deviation	0.04	0.09	0.04		
Std Dev/Mean	0.04	0.10	0.04		

Discussion

Paper 43

D.Harrje (DTH Consultant, USA)

Since the constant injection at 39 air handlers had 5 hours to equilibrate shouldn't this approach be better at achieving the uniform concentration desired?

A.Persily (NIST, USA)

In order for the constant injection approach to result in a uniform tracer gas concentration throughout the building, the injection rate into each air handler must be proportional to the volume served by that air handler. In practice this is very difficult to establish. Any difference in the ratio of injection rate to space volume among the air handlers will result in non-uniform tracer gas concentrations regardless of how long one waits.

J.Axley (MIT, USA)

I would like to reiterate the sources of error in determining vent effectiveness in the field that you mention and establish an additional source. You noted that the reliability of the test depends on one's success in establishing initial conditions which in turn depend on a) the building flow/air distribution character and b) the injection strategy used to establish these initial conditions. An additional source of error is related to the skill of the experimentalist - to establish some sense of this aspect I only remind the attendees here today that you were dealing with 39 air handlers and would ask you to simply describe the size of these air handlers and the building volume they served so that scale of the experimental challenge can be established.

A.Persily (NIST, USA)

I regret having not communicated the difficulty associated with making those measurements, which were in part due to the complexity of the air handling systems. The air handlers were indeed quite large; most of them had supply air capacities greater than 20 m3/s (40,000 cfm) and served volumes larger than 14,000 m^3*. In fact there are several aspects of this building that facilitated the measurements: 24 hour fan operation constant outdoor air intake, fairly uniform ventilation air distribution to the spaces, and good mixing in the spaces. Even under these fortunate circumstances, the measurements were still quite difficult. In other buildings the measurements will generally be even more difficult if they are indeed at all practical.*

Earle Perera (BRE, UK)

Is this the experience with the "pulse decay" technique in real buildings?

A.Persily (NIST USA)

We have made measurements in a mechanically ventilated office building and they were presented in a paper by James Axley and myself at the 9th AIVC Conference. In these measurements we idealized the building as three zones and learned several lessons regarding practical measurement issues. First, one can start the integration period somewhat after the tracer gas injection and avoid some problems associated with the mixing of the tracer. Also, the integration can end before the concentrations get too low and accuracy becomes a problem. In the zone of injection one can determine the integral by filling an air sample container at a constant rate thereby avoid ing measurement inaccuracies associated with determining very high concentrations.

J Van der Maas (LESO, Switzerland)

The ventilation effectiveness seems to vary only 10-20%, how do you expect this to explain the complaints from occupants?

A Persily (NIST, USA)

The ventilation effectiveness measurement results do not explain the occupant complaints nor do the measured ventilation rates. Other factors must be causing the occupants' dissatisfaction with the environment. Additional measurements have been made in the building which may explain the complaints and these will be reported on in the future.

VENTILATION SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
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Paper 44

USING PRESSURISATION MEASUREMENTS TO PREDICT VENTILATION PERFORMANCE AND HEATING ENERGY REQUIREMENTS OF A LARGE INDUSTRIAL BUILDING

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SUMMARY

A single whole building pressurisation test using robust and easy to use equipment can, in a very short time, quantify the air-leakiness of the building envelope. However, such measurements do not give a direct measure of the ventilation characteristics of the building which normally requires time-consuming and specialist tracer gas tests.

This paper provides a model which makes the link between leakage measurements and ventilation characteristics and applies it to a large, industrial building constructed according to 1979 UK Building Regulations. Air leakage measurements with the building 'as found' and then with its loading doors sealed showed a 14% reduction at an inside/outside pressure differential of 25 Pa. Using these leakage characteristics, the model predicted ventilation rates which corresponded well with measured values.

Meteorological data at the site for the heating season were combined with the ventilation characteristics of the building (given by the model) to predict the ventilation performance of the building over that period. The results indicated that the building 'as-found' would have, on average, an air change rate of 0.5 h^{-1} during the heating season. Sealing the loading doors would reduce this rate by 24%, i.e. to 0.38 h^{-1}.

This paper shows that space heating energy requirements for the heating season can be assessed using either the combined hourly-predicted ventilation rates and meteorological data for the site or using the mean predicted ventilation rate with existing design guidance. The two approaches agreed to within 10% of each other. The results also indicated that ventilation heat losses accounted for 44% of the total required energy. Calculations also show that replacing the loading doors with ones which are more air tight and better insulated will reduce by 25% the energy required for ventilation losses and by 5% the losses through the building fabric. This results in an overall reduction of 14% of the total requirement.

1. INTRODUCTION

A major factor in the ventilation of buildings and their energy performance is the leakiness of the building envelope. While adequate ventilation is essential for the health, safety and comfort of the occupants, the adventitious ingress (infiltration) of air through the building envelope can be a source of excessive ventilation and can lead to energy waste and, in some cases, to discomfort.

A single whole building pressurisation test[1] using robust and easy to use equipment can, in a very short time, quantify the air-leakiness of a buildings's external envelope. However, such a measurement does not give a direct measure of the ventilation characteristics of the building. This is usually assessed by tracer gas techniques[2] using sophisticated equipment and specialist expertise. It would also be time-consuming if a proper assessment of the ventilation performance of the building is needed, i.e. an assessment of how often specific ventilation rates occur over a specified period. This is because, at any moment, the ventilation rate of a naturally-ventilated building depends (among other factors) on the prevailing meteorological conditions (i.e. wind speed and direction and outside air temperature) and a proper assessment would require many tracer gas measurements to be carried out over a wide range of weather conditions.

A building pressurisation test is normally used to;

(a) compare the air tightness of different buildings and
(b) identify and quantify rates of leakage through different paths in the same building.

However, the value of a pressurisation test would be considerably enhanced if the results can be linked through a simple procedure to the ventilation performance of the building, bypassing ventilation measurements. This would then make it possible to assess whether there is adequate ventilation within the building and, if excessive, to;
(c) assess the potential for reducing air infiltration and,
(d) determine the cost-effectiveness of retrofit measures in reducing the energy used for the space heating of the building by reducing its ventilation heat loss.

The use of a simple correlation becomes even more attractive in the case of naturally-ventilated and large non-domestic buildings where tracer gas measurements can be fraught with difficulties[2]. As a starting point to provide similar correlations for more complex building-types like offices, this paper describes a preliminary attempt to make this link for a large single-space industrial building.

As said above, such a link could result in the identification of energy conserving opportunities. At the moment, UK industry as a whole consumes about 2600 PJ of primary energy annually[3] accounting for about 30% of the UK total. Of this, the 0.5 million or so industrial premises account for some 600 PJ (at a cost of £1400 million) for space and water heating and lighting with space heating being by far the largest component. A recent study[4] has shown that one of the most important factors in any energy savings in this type of industrial building is the impact of loading doors on air infiltration rates.

This paper covers a three-stage process to relate air leakage measurements to ventilation performance and energy use for a naturally-ventilated large single-space industrial building. The effect of sealing the loading doors is considered. In the first stage, a simple theoretical model is derived to predict the ventilation characteristics of the building from its measured whole building air leakage characteristics. The predicted values are then compared with field measurements (in which tracer gas techniques were used).

Given a specific wind direction, wind speed and an external air temperature, the ventilation rate of the building can be predicted from its ventilation characteristics. However, the ventilation characteristics of the building makes no reference to the local climatic conditions expected at the site. This is achieved in the second part of this paper by combining the ventilation characteristics of the industrial building with the local meteorological data to predict the ventilation performance of the building, i.e. how often various ventilation rates would be expected to occur. In this paper, the assessment was constrained to the ventilation performance expected over a typical heating season between the beginning of October until the end of March.

Finally, the effect of sealing the loading door on the heating energy requirements of the industrial building is assessed using the predicted ventilation performance. The energy requirements obtained are also compared against current guidance procedures[5].

2. PREDICTING VENTILATION CHARACTERISTICS FROM PRESSURISATION MEASUREMENTS

2.1. Theoretical Model

In a naturally-ventilated building, air enters either through design (e.g. through purposely provided openings like windows) or adventitiously by uncontrolled leakage (infiltration) through cracks and gaps in the building envelope. The air is driven through the openings by the pressure difference between inside and outside set up by the combined influence of wind and inside/outside temperature difference.

The pressure difference ΔP (Pa) across the opening can be expressed as[6],

$$\Delta P = (\frac{1}{2}\rho_o U^2 C_p + P_o - \rho_o g z) - (P_i - \rho_i g z) \qquad .... (1)$$

where C_p is the wind pressure coefficient referenced to a freestream wind speed U (m/s) measured at a reference height,
ρ_o is the outside (external) air density (kg/m^3),
P_o and P_i are the outside and inside static pressures (Pa) at height z = 0 m and
g is the gravitational constant (= 9.80665 m/s^2).

For large openings like windows, the air flow rate Q (m^3/s) through that component is given by the simple formula

$$Q = A\, C_d \sqrt{\frac{2\, \Delta P}{\rho}} \qquad .... (2)$$

where A is the area (m^2) of the opening, C_d is the discharge coefficient (usually given a value of 0.61) and ρ is the density (kg/m^3) of air flowing through the opening.

Using a derivation similar to that of Warren and Webb[6], the adventitious leakage Q (m^3/s) through any single element of the building facade, i.e. a roof or wall element, for a pressure differential ΔP can be obtained from a whole-building leakage measurement. For brevity, this is not derived here but it can be shown that this relationship is given by

$$Q = \rho_o \left[\frac{k \rho_{pr}}{g(n+1)}\right] \left(\frac{A}{A_T}\right) \left(\frac{1}{h_U - h_L}\right) \left[|a - bh_L|^{n+1} - |a - bh_U|^{n+1}\right] \qquad .... (3)$$

where k and n are respectively the coefficient and exponent obtained from a pressurisation test and related by an equation of the form, $Q = k.\,\Delta P^n$,
ρ_{pr} is the outside air density during the pressurisation test.
A (m^2) is the area of the facade element,
A_T (m^2) is the total external permeable area of the building and
h_L and h_U are the lower and upper heights of the facade element above ground level.

Here, $a = 1/2\ \rho_o\ U^2\ C_p - p_i$ where p_i (= $P_I - P_e$) is the inside pressure and
$b = g\ (\rho_o - \rho_i)$ in which ρ_i is the air density inside.

The main assumption[6] inherent in deriving the above equation is that the pressure generated by the wind and the air leakage through the envelope is uniformly distributed across each surface. However, a weighting factor (totalling 1.0 over all surfaces) can be ascribed, if necessary, to each of the surfaces to take into account variations in air permeability.

A computer program BREAIR, similar to the domestic model BREVENT[7], was written to solve the above equations. The inside air pressure is the unknown parameter to be found. The program uses an iterative procedure to calculate this pressure for which there is mass balance. Volume flows are then calculated using the appropriate air densities.

2.2. Field Measurements

Whole building pressurisation tests and tracer gas measurements of ventilation flow rates were carried out in an industrial building in Cwmbran, Wales. Measurements in this building formed part of a larger programme of seven similar studies undertaken in a variety of single-celled industrial buildings to provide, amongst other things, data sets for validating prediction procedures. In each building, measurements were made before and after sealing various identifiable air leakage paths.

Industrial building

This detached single-cell building (Fig. 1) is located in the Ty-Coch Industrial Estate, Cwmbran in Wales. Figures 2 is a site plan showing the location of the building relative to its neighbours and showing that the factory faces east and is oriented on a north-south axis.

The volume of the building was estimated as 15063 m^3 with a production floor area of about 1950 m^2. The walls and roofs were built to Part FF standard of the 1979 UK Building Regulations which required wall and roof U-values of 0.7 W/m^2K. The wall comprised of a metal clad outer leaf with a fibre board inner leaf containing 60 mm of glass fibre insulation with a band of single glazing at high-level. On the east wall, there were two standard roller-shutter loading doors, each with an area of 18 m^2.

There were two working areas or bays with duo-pitch metal clad roofs with an asbestos panelled inner leaf containing 60 mm of glass fibre insulation. There was approximately 5% roof lighting but no roof ventilators.

Whole-building pressurisation tests

A pressurisation rig consisting of four large fans was used to pressure test the building. Two tests were carried out in calm conditions; one with the building 'as-found' and the other with both loading doors sealed with polythene sheeting. The outside air temperature during the test was about 8°C with the inside temperature ranging between 8 and 9°C.

The generally accepted relationship

$$Q = k \cdot \Delta P^n \qquad \text{.... (4)}$$

between air leakage Q (m^3/s) and inside/outside static pressure difference ΔP (Pa) was fitted to the measured data. Best-fit regression lines (with correlation, $r^2 > 0.99$) on the measured data gave the following results:

Condition of Building	Coefficient, k $m^3/s.Pa^n$	Exponent, n
'as-found'	4.162	0.497
loading doors sealed	3.300	0.521

For large buildings, it is usual[1] to consider the air leakage rate at 25 Pa. Calculations at this pressure show that the air leakage rate is reduced by 14% when the loading doors were sealed.

Ventilation rate measurements

An automated tracer gas system[8] was used to measure the ventilation rates within the building as-found and with the loading doors sealed. The measurements were carried at various intervals over a period of one month to cover varying outside weather conditions. Both 'decay' and 'constant-concentration' measurements[2] were made using nitrous oxide as the tracer. However, only a portion of this data - sufficient to evaluate the predictions - is used in this paper. An analysis of the full data set will be the subject of a future report.

Inspection of the full data set showed that there was sufficient information available from the constant-concentration tests to compare the performance of the factory as-found and also with loading doors sealed for winds blowing from the south. Figure 3 shows the data corresponding to this direction. The outside air temperature for this set averaged at 8°C, while the inside temperature averaged around 11°C for the building as-found and 12°C with the loading doors sealed.

Wind pressure coefficients for predicting ventilation

Wind pressure coefficients are necessary to carry out any prediction. In the absence of specific wind-tunnel measurements, it is sometimes possible to use published wind tunnel data which may have been gathered for other purposes, such as for wind loading calculations when the measurements have been made on isolated buildings.

However, it is known[9] that such pressures are substantially reduced when the building is sheltered or surrounded by buildings of similar height and previous work[6,9,10] have shown that these coefficients could be halved for predicting ventilation in buildings which are sheltered by others of similar height. In this paper, wind pressure coefficient for this type of industrial buildings (multi-bays with 5° duo-pitch roofs) were obtained from published wind-loading data[11] and then modified. These amended values, used in the present prediction, are given in Table 1.

2.3. Comparing Predicted with Measured Ventilation Rates

The input data for BREAIR comprised of the estimated wind pressure coefficients for southerly winds, the measured envelope air leakage data and the individual permeable areas of the roof and wall areas (estimated from Figure 1). The model was run through a range of wind speeds using air temperatures appropriate to the measured data (shown in Fig 3).

In Figure 3, the predicted curves are superimposed over the measured values for both the building 'as-found' and also with the loading doors sealed and shows good comparison between measured and predicted. The predictions appear to identify correctly not only the magnitude of the rates but also the point at which the ventilation switches over from being buoyancy-induced (as a consequence of the inside/outside temperature difference) to wind-induced.

Ad-hoc spot checks were also carried out on the rest of the measured data corresponding to other wind directions. In all, the results were as good as for the specific wind direction considered here. This gives confidence in using the prediction model in this instance. Further comparisons will need to be carried out later on other industrial buildings to confirm the general validity of the model.

3. ASSESSING VENTILATION PERFORMANCE

These ventilation predictions, although specific to the building, make no reference to local climatic conditions at the site. To assess the ventilation performance and to predict how often various levels of ventilation could be expected, the predicted ventilation characteristics of the building have to be combined with the weather frequency distribution, i.e. the joint occurrence of wind direction, wind speed and outside air temperature.

3.1. Compressing Ventilation Flows

To assess the overall ventilation performance of the industrial building, it is necessary to compute the ventilation flow Q for winds blowing from each direction ϕ for a range of wind speeds u and air temperature differences $\Delta\theta$ between inside and outside. To avoid generating a large and unmanageable set of data, the results can be presented in a compressed form, in which the temperature dependence has been removed, without any loss of generality.

Previous work[12] has shown that, for each wind direction ϕ, all the predicted or measured air flow rates $Q(u,\Delta\theta,\phi)$ can be collapsed into such a form by scaling both the ventilation flow and the wind speed by the factor $1/\Delta\theta^{1/2}$. Using predicted values from BREAIR, this was carried out for the industrial building with and without the loading door sealed. Dividing the flow rates by the volume (V) of the building, the air change rates I (per hour) were calculated and scaled accordingly and Figure 4 shows the results for, in this instance, a temperature difference of 1°C. Note that all directions are not represented because of symmetry (about the north-south and east-west axis) for the building 'as found' and (along the east-west axis) with the loading doors sealed.

This compressed form makes the data more manageable when used as input to assess the ventilation performance of the building. This aspect is further enhanced by being able to represent them with polynomial expressions[12] of the form,

$$I(\phi) = a_0(\phi) + a_1(\phi)\overline{u} + a_2(\phi)\overline{u}^2 + a_3(\phi)\overline{u}^3 + a_4(\phi)\overline{u}^4 \qquad \text{.... (5)}$$

where

$$I(\phi) = \frac{Q(u,\Delta\theta,\phi)}{V\sqrt{\Delta\theta}}$$

and,

$$\bar{u} = \frac{u}{\sqrt{\Delta\theta}}$$

Using regression analysis, best-fit (correlation $r^2 > 0.99$) of the above form were fitted to the predicted curves (Fig 4). The polynomial coefficients obtained are tabulated in Table 2.

3.2. Meteorological Data

A record of local weather conditions was obtained from the nearby meteorological station at Cilfynydd, Wales. Although not the nearest, it was located in a terrain similar to the building site. Calculations[13] to take account of change in site and height (required since the pressure coefficients were referenced to the eave height of the building) of the meteorological anemometer showed that the wind speeds at the site had to be reduced to 56% of that measured at the station.

The meteorological data (collected over the period October 1982 to March 1990) was constrained in the analysis to the winter heating season, i.e. beginning of October to the end of March and to the time period between 0600 and 1800 GMT when the building would be heated. Since measurements were collected only at three-hourly intervals, some refinement of the data was carried out to translate them into representative hourly values. No attempt was made to further segregate the weather data to exclude week-ends since the standard occupancy pattern of a full week is used to calculate energy demands in factories[5].

Figure 5 shows the weather conditions monitored at the meteorological station for this constrained period. Winds from the north predominate even though there is some substantial occurrence from the south-west. The mean wind speed and the outside air temperature exceeded for 50% of the time are about 3.7 m/s and 7.5°C respectively.

3.3. Statistical Assessment of the Ventilation Performance

The design or required inside temperature was taken as 19°C, a value normally used[5] to assess heating energy requirements. Using the polynomial expressions given in Equation 5, the air change rates were determined for combinations of wind direction, speed and outside air temperature. The number of hours that these combinations occurred were then read from joint frequency tables (of wind speed, outside air temperature and wind direction) and placed in 'bins' corresponding to various intervals of ventilation flows to build a frequency distribution. Figure 6 shows the evaluated percentage frequency of the ventilation air change rate within the building as-found. Figure 7 shows this translated to a frequency of exceedance. Figure 7 also shows the corresponding information for the case when the doors were sealed.

Figure 7 shows that the mean air change rate within the building as-found is 0.5 h^{-1} corresponding to the recommended[14] design value for calculating energy demands. Figure 7 also shows quite clearly that sealing the loading doors reduces this mean (50% exceedance) rate to 0.38 h^{-1}, i.e. a 24% reduction in the mean ventilation rate.

4. PREDICTING HEATING ENERGY REQUIREMENTS

4.1. Using Predicted Ventilation Rates

The heating energy E_V (GJ) required for losses by natural ventilation over the heating season is given by,

$$E_V = \rho_r \, c \sum_{heating\ period} Q \, (\theta_r - \theta_o) \, 10^{-6} \, \Delta t \qquad .... (6.a)$$

where c (= 0.988 kJ/kg.K) is the specific heat of air,
θ_r and θ_e are the required (design) inside and outside air temperatures respectively,
ρ_r is the air density corresponding to the required air temperature,
Q (m^3/h) is the ventilation rate predicted for a particular wind speed, wind direction and outside temperature, and
Δt is the individual time period (h) during which this occurs.
This calculation is carried out only when the inside air temperature is greater than that outside.

The energy required E_F (GJ) for fabric heat losses over the heating season is similarly given by,

$$E_F = 3.6 \; 10^{-6} \sum_{heating\ period} (\Sigma AU) \, (\theta_r - \theta_o) \, \Delta t \qquad .... (6.b)$$

where Σ(AU) is the product of all areas A (m^2) of surfaces separating the heated space from the outside and their U-values (W/m^2K). The areas and the U-values for each major component of the industrial building are given in Table 3. Note that the U-value for the floor relates[14] to the outside air temperature and not to the surface ground temperature.

Using the meterological data, the energy requirements given in Equations 6 were evaluated for the building as-found and with the loading doors replaced with higher performance doors which are more air tight and better insulated. The results were obtained for each of the major building components and are given in Table 4.

4.2. Compensating the Predicted Heating Energy Requirement for Intermittent Heating

According to design guide[5], when a reduction in the required temperature is allowed at night, the mean inside temperature, θ_{im}, can be taken as the mean temperature calculated for an intermittently heated building. This mean temperature is a function of:

(i) The required inside temperature, θ_r, during the heating period.
(ii) The average daily heating period, including pre-heating, H.
(iii) The response factor, f, of the building.

The equations relating these[5], re-written here for completeness, are as follows:

$$f = \frac{\Sigma(AY) + \rho c Q}{\Sigma(AU) + \rho c Q} \qquad \text{.... (7.a)}$$

and.

$$\theta_{im} = \theta_{om} + \frac{H f (\theta_r - \theta_{om})}{H f + (24 - H)} \qquad \text{.... (7.b)}$$

where Q is the design ventilation rate (m^3/s),
$\Sigma(AY)$ is the sum of products of areas of all exposed surfaces and their appropriate thermal admittances (Table 3) and
θ_{om} is the 24-hour mean outside air temperature (°C)
and all other symbols as previously defined.

A mean outside air temperature θ_{om} of 7°C was estimated for Cwmbran from design guidance[5], a value close to that identified from the meteorological data (Section 3.2). Using air change rates of 0.5 (building as-found) and 0.35 h^{-1} (loading doors sealed), f was evaluated from Equation 7.a. as about 3. This value for the response factor is typical[14] for thermally lightweight buildings of this type. A more massive building will have a higher response factor. Using Equation 7.b., θ_{im} was evaluated as 15.9°C.

The total energy requirement E_H (comprising of requirements for both ventilation and fabric losses) evaluated in previous Section 4.1. covered only the heating period H from 0600 to 1800 hrs. It is therefore necessary to correct to a 24-hour requirement, $E_{24,}$ to compensate for thermal storage effects of the construction and loss during the unheated overnight period. This can be obtained from the equation (Harrington-Lynn, private communication),

$$\frac{E_H}{E_{24}} = \frac{H (\theta_r - \theta_{om})}{24 (\theta_{im} - \theta_{om})} \qquad \text{.... (8)}$$

Substitution of the appropriate values, evaluated for the present case, in the above equation shows that 33% of the energy input during the heating period is carried over due to thermal storage and lost during the unheated overnight period. This 33% carry-over means that the estimates of energy use predicted earlier need to be increased by 50% resulting in the following revised estimate:

Component	HEATING ENERGY REQUIREMENTS (GJ)	
	building as-found	with higher-performance loading doors
FABRIC	496	469
VENTILATION	392	295
TOTAL	888	764

This shows that approximately 44% of the heating energy required for this building is for ventilation losses with over half lost through the fabric. Even though sealing the loading door reduces the losses through ventilation by 25%, the reduction through the fabric is only 5% (as a consequence of reducing the U-value of the loading doors) but overall there is a 14% reduction in the total energy requirement.

The effect of sealing the loading doors in this factory, built to present UK Standards, can be compared with a low energy factory building (with much higher levels of thermal insulation) where a nominally similar measure[4] was also carried out. In this instance, it was estimated that ventilation losses were reduced by 52%, fabric losses by 21% and the total by 32%.

4.3. Using Design Guidance

As a check against the above predicted values, a standard design procedure[5] was used to calculate the energy demand for space heating of the industrial building. Using the transmittance and admittance values given in Table 3, design air change rates of 0.5 and 0.38 h^{-1} were used in calculations for the building as-found and with the leading doors sealed and the results are summarised below:

Component	HEATING ENERGY REQUIREMENTS (GJ)	
	building as-found	with higher-performance loading doors
FABRIC	465	451
VENTILATION	346	270
TOTAL	811	721

The comparison between these design values using a ventilation rate averaged over the heating season and those obtained from the hourly predictions are within 10% of each other with the hourly predictions over-estimating the energy requirements. These comparisons give confidence, at least for this building, that predicted energy demands for space heating could be estimated from whole-building air leakage measurements or obtained from existing design guides provided the air change rate for the building is correctly identified.

5. CONCLUSIONS

The whole building pressurisation tests on the industrial building showed that the air leakiness of the external building fabric was reduced by 14% (at a 25 Pa pressure difference between inside and out) when both loading doors were sealed.

A simple ventilation prediction model which used the building's leakage characteristics was shown to compare well with measured (using tracer gas methods) ventilation data. The predictions were combined with the meterological weather conditions expected during the heating season to predict the ventilation performance of the building. Results indicate that the mean (exceeding for 50% of the time) ventilation rate was 0.5 air changes per hour (ach) with the building 'as-found'. This rate was reduced to 0.38 ach when the loading doors were sealed, i.e. a reduction of 24%.

The ventilation model was also used to predict (on an hourly basis) the space heating energy requirements for the heating season. These values were within 10% of that given by standard design procedures using the predicted ventilation rates averaged over the heating season. The predictions showed that replacing the loading doors with higher-performance doors would reduce the energy requirement for ventilation from 392 to 295 GJ (reduction of 25%), those due to fabric losses from 496 to 469 GJ (a 5% reduction) and the total from 888 to 764 GJ (i.e. reduced by 14%).

Using a database of field measurements, additional work will be carried out to validate further this approach of using measured external wall leakage characteristics to estimate the ventilation performance of naturally ventilated, large non-domestic buildings and to predict their space heating energy demands.

ACKNOWLEDGEMENTS

Thanks are due to Andrew Cripps for help with the BREAIR algorithms. The considerable help and guidance given by John Harrington-Lynn in the energy prediction aspect of this work is gratefully acknowledged. The field measurements described in this paper were carried out by the Welsh School of Architecture under contract to the Building Research Establishment. The work described here has been carried out as part of the research programme of the Building Research Establishment of the Department of the Environment and this paper is published by permission of the Chief Executive.

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TABLE 1 - PRESSURE COEFFICIENTS, C_P

	Wall area				Roof area			
Wind direction (°N)	North-facing	East-facing	South-facing	West-facing	North-facing outer	South-facing inner	North-facing inner	South-facing outer
0	0.43	-0.21	-0.11	-0.21	-0.29	0.00	-0.23	0.00
30	0.35	0.20	-0.13	-0.23	-0.30	0.00	-0.24	0.00
60	0.20	0.35	-0.23	-0.13	-0.21	0.00	-0.17	0.00
90	-0.21	0.43	-0.21	-0.11	0.00	0.00	0.00	0.00
120	-0.23	0.35	0.20	-0.13	0.00	-0.17	0.00	-0.21
150	-0.13	0.20	0.35	-0.23	0.00	-0.24	0.00	-0.30
180	-0.11	-0.21	0.43	-0.21	0.00	-0.23	0.00	-0.29
210	-0.13	-0.23	0.35	0.20	0.00	-0.24	0.00	-0.30
240	-0.23	-0.13	0.20	0.35	0.00	-0.17	0.00	-0.21
270	-0.21	-0.11	-0.21	0.43	0.00	0.00	0.00	0.00
300	0.20	-0.13	-0.23	0.35	-0.21	0.00	-0.17	0.00
330	0.35	-0.23	-0.13	0.20	-0.30	0.00	-0.24	0.00

All coefficients referenced to freestream dynamic pressure measured at eave height

TABLE 2 - POLYNOMIAL COEFFICIENTS

Factory - as found:

Wind direction (°N)	a_0	a_1	a_2	a_3	a_4
0	0.1381	-0.0707	0.0747	-0.0114	0.0006
30	0.1303	-0.0089	0.0422	-0.0053	0.0002
60	0.1207	0.0296	0.0131	-0.0004	0.0000
90	0.1256	-0.0121	0.0230	-0.0033	0.0002
120	0.1207	0.0296	0.0131	-0.0004	0.0000
150	0.1303	-0.0089	0.0422	-0.0053	0.0002
180	0.1381	-0.0707	0.0747	-0.0114	0.0006
210	0.1303	-0.0089	0.0422	-0.0053	0.0002
240	0.1207	0.0296	0.0131	-0.0004	0.0000
270	0.1256	-0.0121	0.0230	-0.0033	0.0002
300	0.1207	0.0296	0.0131	-0.0004	0.0000
330	0.1303	-0.0089	0.0422	-0.0053	0.0002

Loading doors sealed:

Wind direction (°N)	a_0	a_1	a_2	a_3	a_4
0	0.1032	-0.0530	0.0578	-0.0086	0.0004
30	0.0969	-0.0080	0.0348	-0.0044	0.0002
60	0.0910	0.0192	0.0117	-0.0003	0.0000
90	0.0942	-0.0120	0.0186	-0.0026	0.0001
120	0.0910	0.0192	0.0117	-0.0003	0.0000
150	0.0969	-0.0080	0.0348	-0.0044	0.0002
180	0.1032	-0.0530	0.0578	-0.0086	0.0004
210	0.0973	-0.0039	0.0319	-0.0037	0.0002
240	0.0900	0.0198	0.0161	-0.0017	0.0001
270	0.0938	-0.0095	0.0192	-0.0026	0.0001
300	0.0900	0.0198	0.0161	-0.0017	0.0001
330	0.0973	-0.0039	0.0319	-0.0037	0.0002

TABLE 3 - THERMAL TRANSMITTANCE AND ADMITTANCE PROPERTIES

COMPONENT	AREA, A m^2	TRANSMITTANCE, U $W/m^2 K$	ADMITTANCE, Y $W/m^2 K$
Wall	1283.8	0.70	0.75
Wall glazing	44.6	5.60	5.60
Loading door	36.0	5.60 (**0.70**)	5.60 (**0.75**)
Roof	1851.4	0.70	0.75
Roof lights	97.4	3.50	3.50
Floor	1830.6	0.21	6.00

(Note: Figures in bold for the loading doors give relevant values when the doors are sealed)

TABLE 4 - ENERGY REQUIREMENTS WITHOUT COMPENSATING FOR INTERMITTENT HEATING

COMPONENT	HEATING ENERGY REQUIREMENTS (GJ)	
	building as-found	loading doors sealed
Wall	89	89
Wall glazing	25	25
Loading door	20	2
Roof	129	129
Roof lights	34	34
Floor	38	38
VENTILATION	264	199
TOTAL	599	516

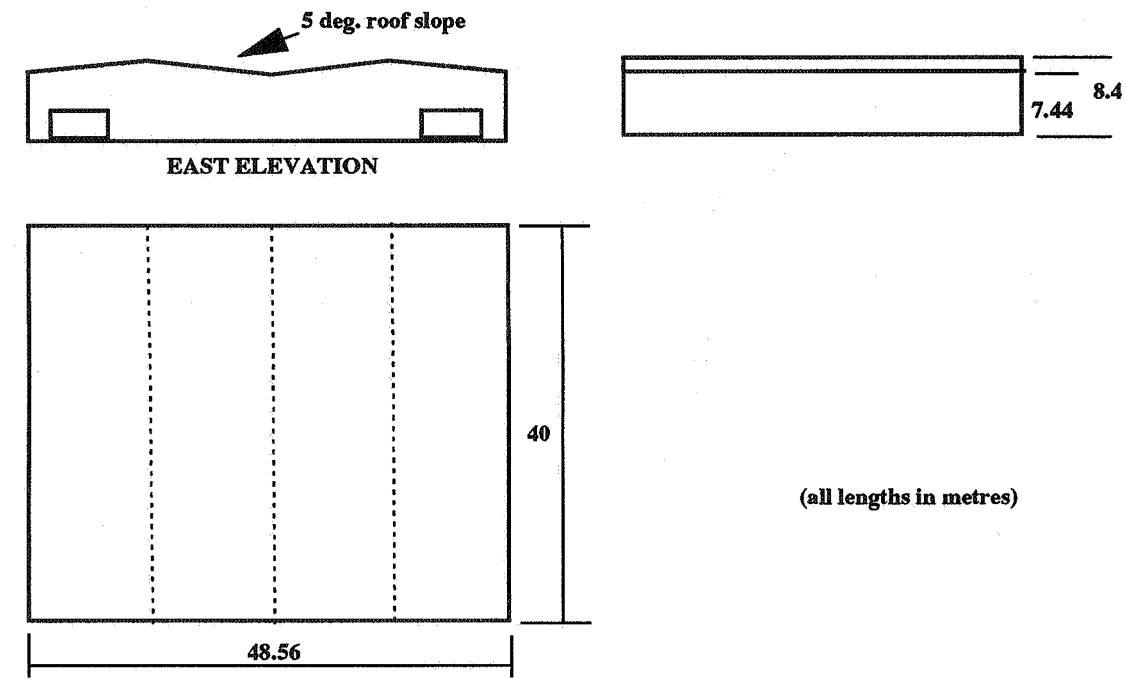


FIGURE 1 - PLAN AND ELEVATIONS OF THE INDUSTRIAL BUILDING

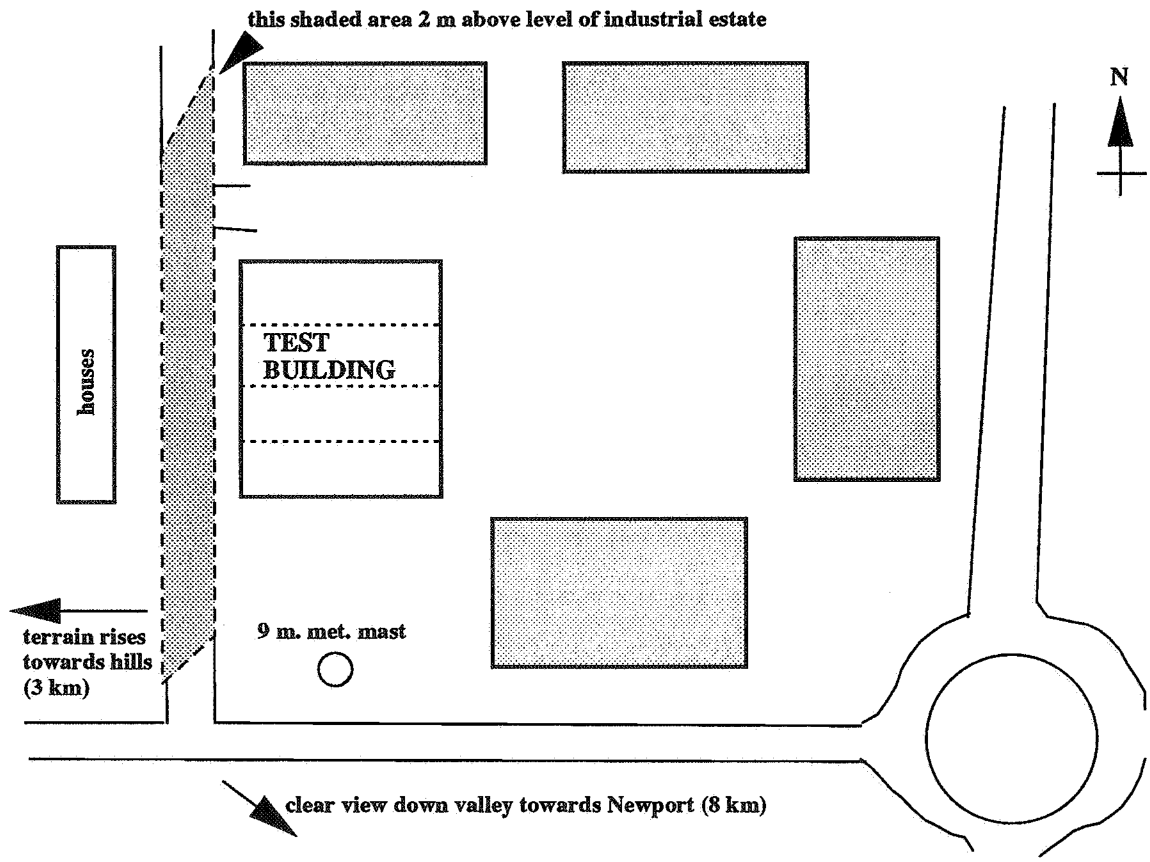


FIGURE 2 - SITE PLAN OF TEST INDUSTRIAL BUILDING

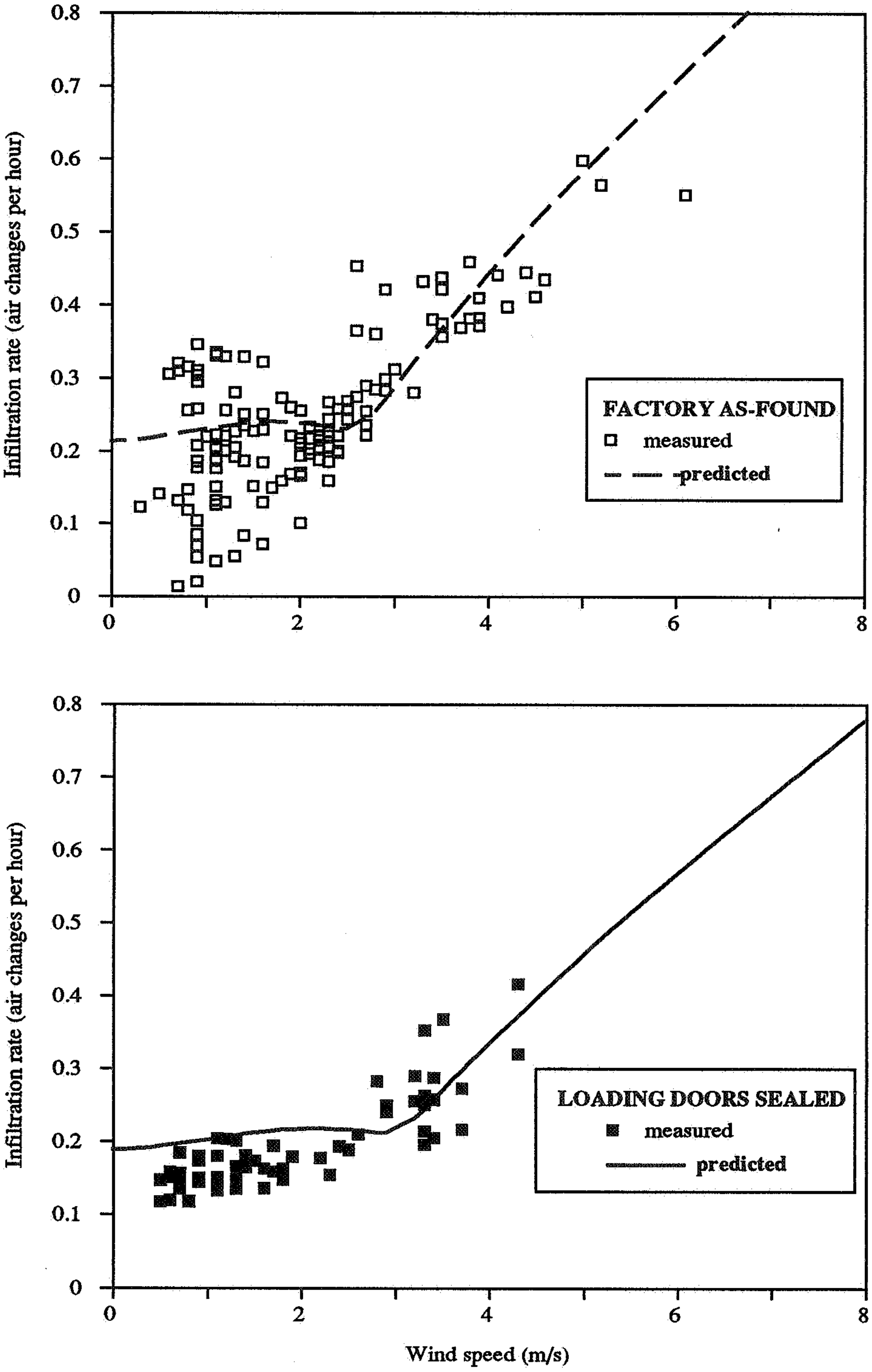


FIGURE 3 - VENTILATION RATES FOR SOUTH WINDS

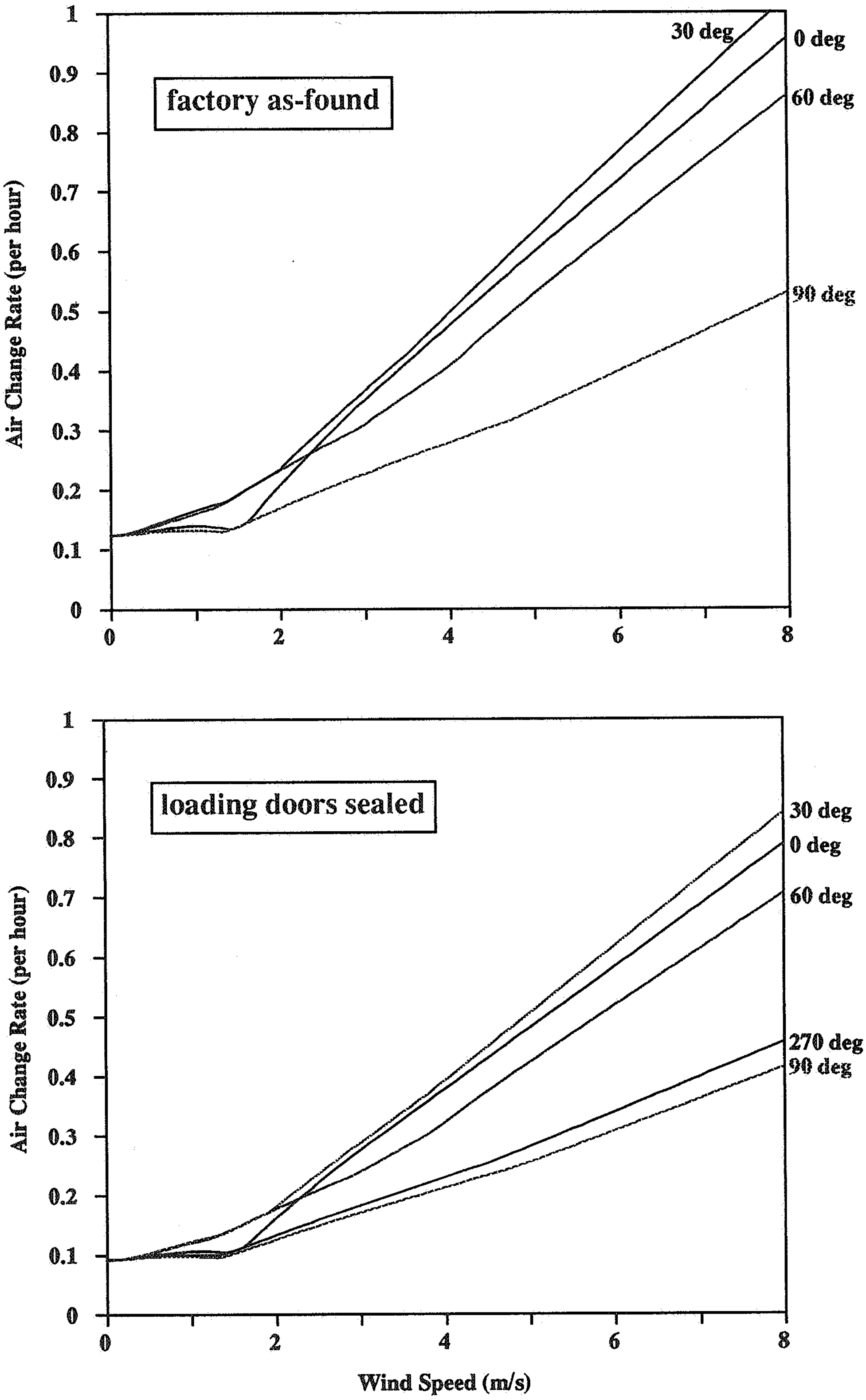
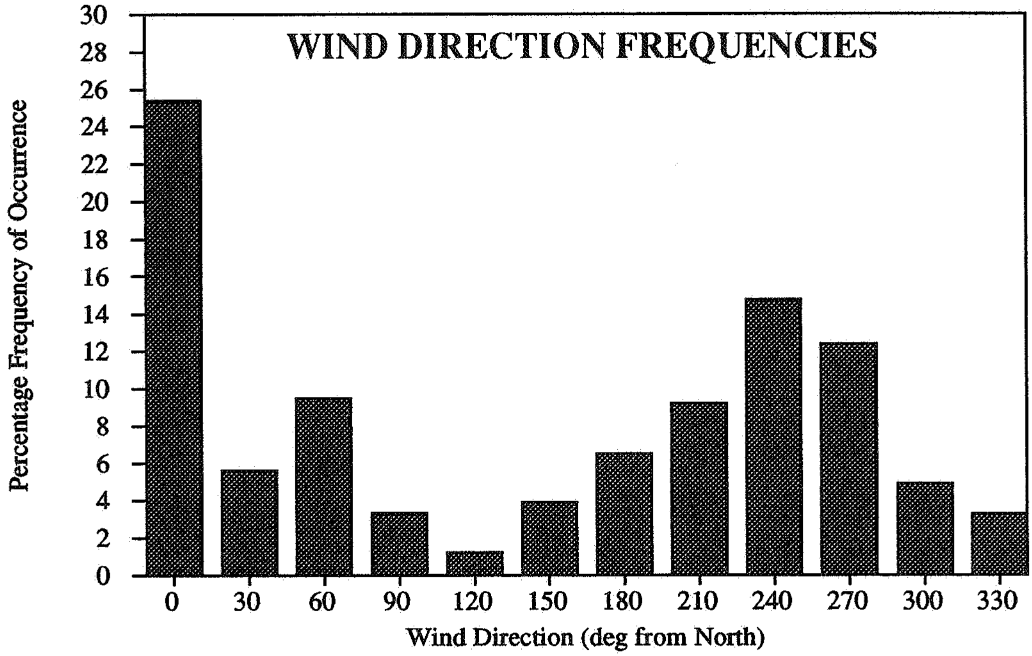


FIGURE 4 - PREDICTED VENTILATION RATES



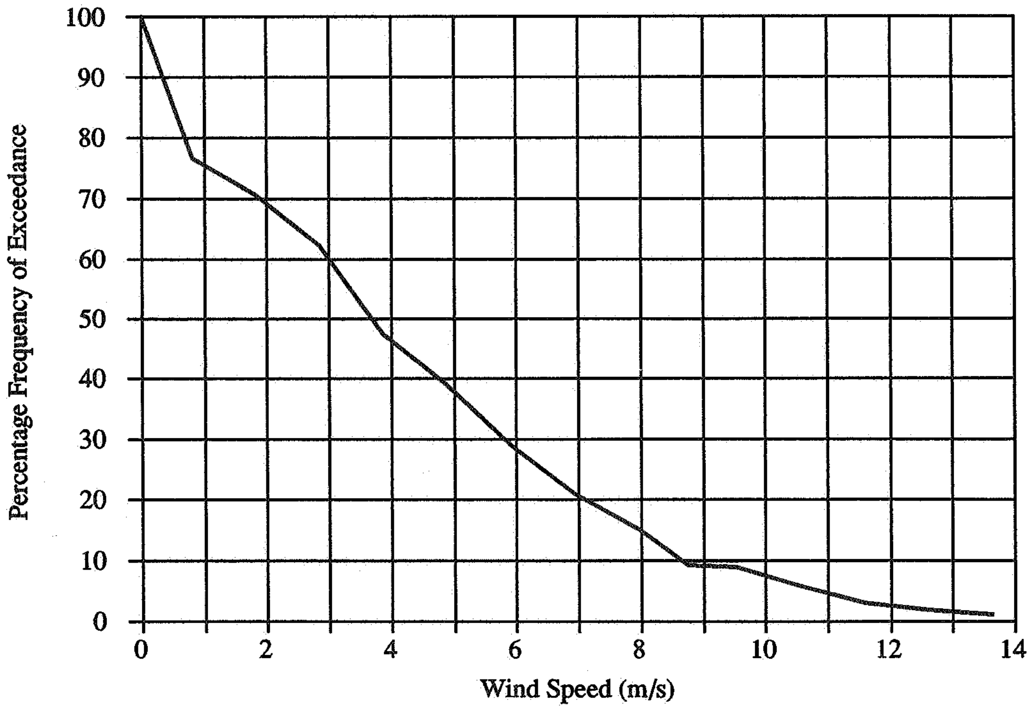


FIGURE 5.a. - WIND CONDITIONS AT THE MET. STATION

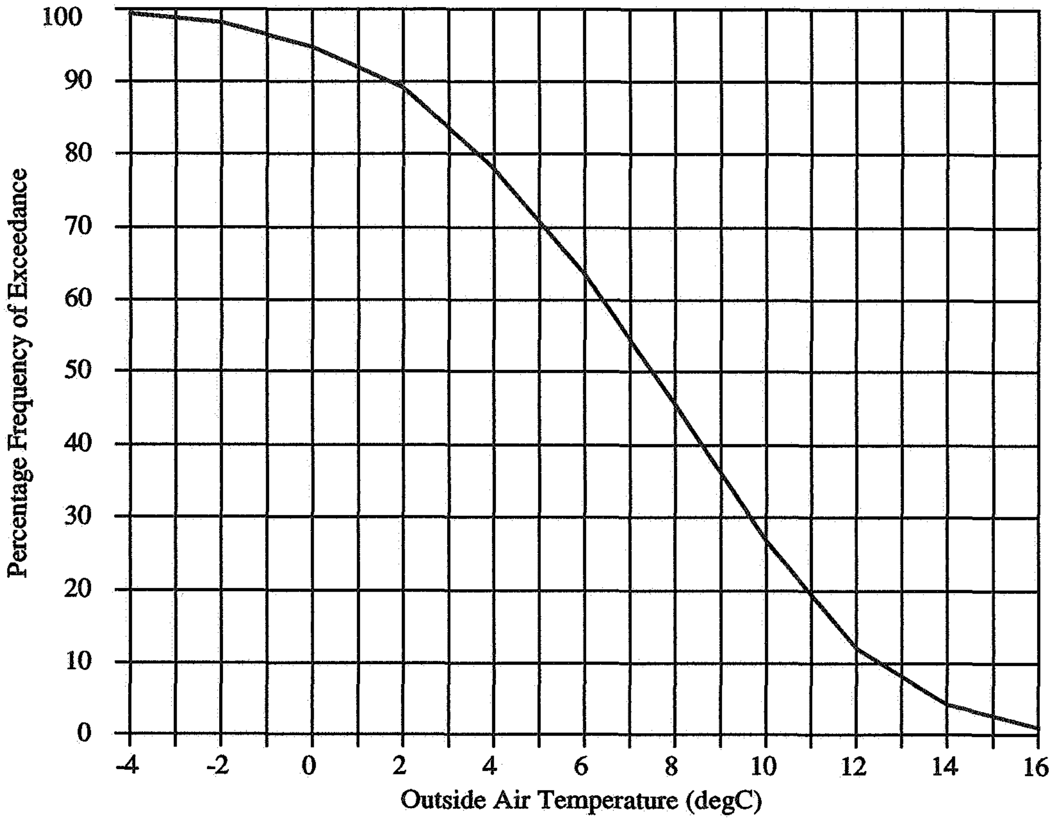


FIGURE 5.b. - OUTSIDE AIR TEMPERATURE AT MET. STATION

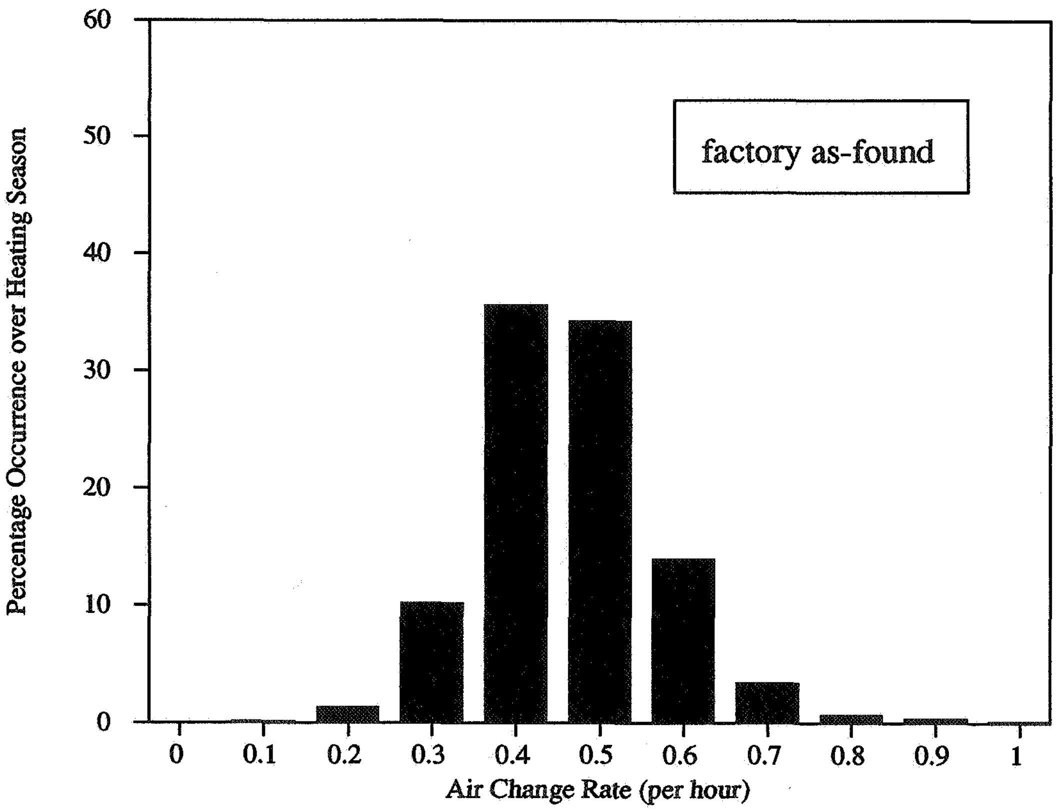


FIGURE 6 - PREDICTED VENTILATION PERFORMANCE

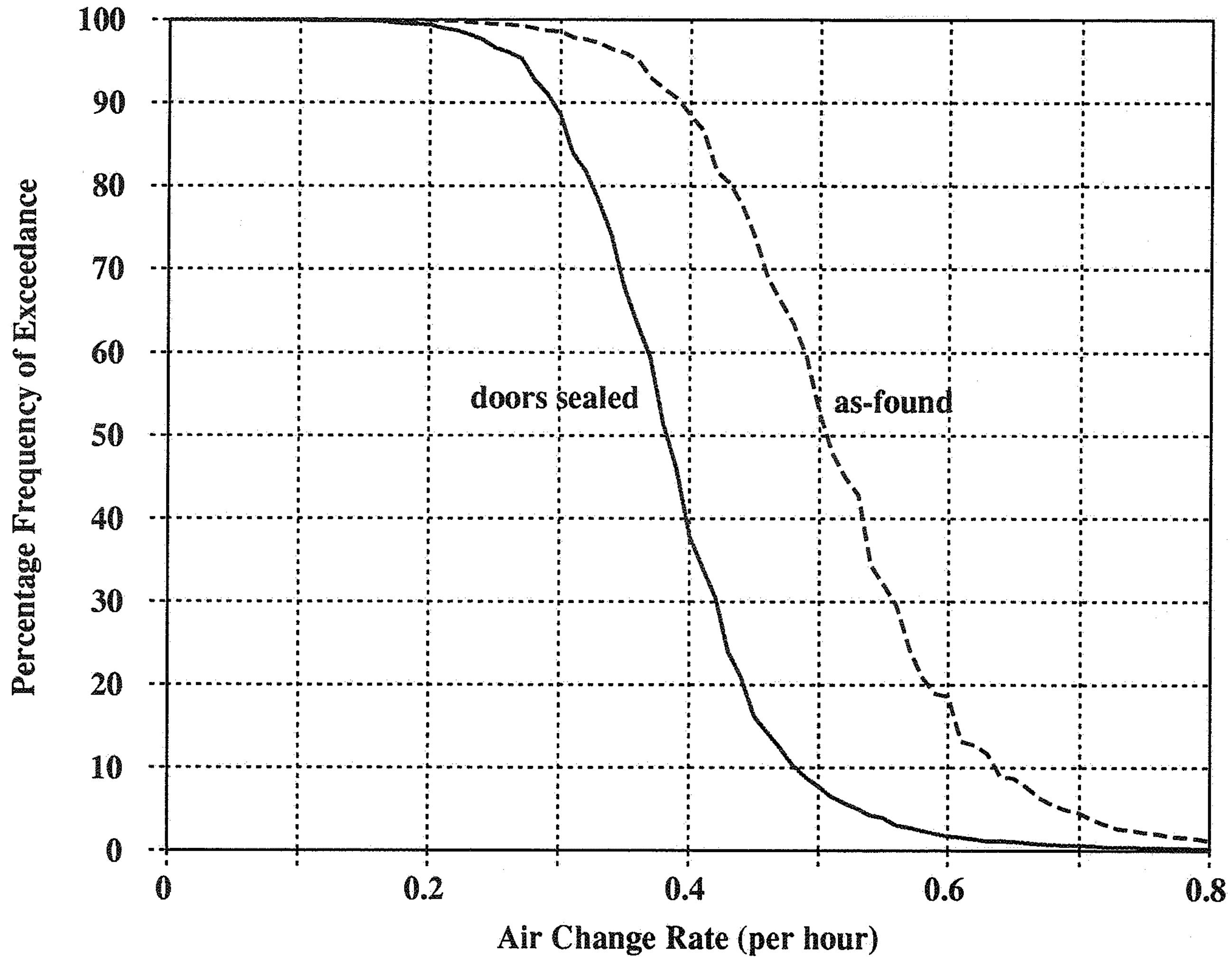


FIGURE 7 - EFFECT OF SEALING DOORS ON VENTILATION PERFORMANCE

Discussion

Paper 44

J Van Der Maas, LESO, Switzerland

What is the relative influence of wind and stack effect in your calculation? Can you estimate the effect of shielding by trees e.g.?

Earle Perera, BRE, UK

The relative effects are not explicitly identified within the paper. However, it is comparatively easy to modify the prediction programme to identify these individual contributions if it is so required. Shielding by trees will reduce the localised pressure coefficients on the building fabric and hence the flow through discreet openings, and the computer model can account for this. Background leakage calculation, which requires an area-weighted average pressure coefficient, would be affected only by a substantial amount of tree shading.

Bas Knoll, TNO, The Netherlands

Did you account for a vertical temperature-gradient in the building because it may influence energy-demand if the warmest air at top is exhausted?

Earle Perera, BRE, UK.

No, we supposed a uniform distributed temperature and leakage. However, it is not a problem to modify Equation (1) and (3) in the paper to account for this effect.

VENTILATION SYSTEM PERFORMANCE

11th AIVC Conference, Belgirate, Italy
18-21 September, 1990

Paper 45

AN APPROACH TO THE SIMULATION OF COUPLED HEAT AND MASS FLOWS IN BUILDINGS

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SYNOPSIS

This paper describes the techniques used within the ESP system to represent and solve the heat and mass conservation equations relating to combined building and plant systems. In particular, it describes the equation-sets used to represent inter-zonal (building) and inter-component (plant) fluid flow and the method used for the integration of the non-linear heat and mass flow equations. By means of a case study, the application in a real design context is demonstrated.

1 INTRODUCTION

In building systems, and the HVAC plant which services them, fluid flow phenomena are encountered in four principle areas:

- air flow through cracks and various openings in the building envelope and interior partitions in the form of infiltration and natural ventilation;
- the flow of air through the distribution networks which exist to satisfy the heating/ cooling demands and ventilation requirements;
- the flow of heating/ cooling fluids through the plant system network;
- and the convective fluid flows within interior building spaces and plant components.

In building design, some knowledge of the magnitude of these flows is a necessary prerequisite of load and energy calculations, system control analysis, thermal comfort assessment and contaminant/ moisture dispersal estimation. Although fluid flow is demonstrably an important aspect of building/ plant performance assessment, the sophistication of its treatment in many modelling systems has tended to lag the treatment applied to the other important flow paths such as the shortwave and longwave processes and conduction. The principal reason for this would appear to be the inherent computational difficulties and the uncertainty associated with the estimation of the parameters of air flow problems.

In recent times more emphasis has been placed on fluid flow simulation with two approaches extant:

Computational Fluid Dynamics (CFD)

A method in which the conservation equations for mass, momentum and thermal energy are solved for all nodal points of a two- or three-dimensional grid inside or around the object under investigation. A well known example of a CFD model is PHOENICS (Spalding 1981).
While in theory the CFD approach is applicable to any thermo-fluid phenomenon, in practice, and in the building physics domain in particular, there are a several problematic issues of which the amount of necessary computing power (Chen 1988), the nature of the flow fields and the assessment of the complex, occupant-dependent boundary conditions are the most problematic. This has often led to CFD applications being restricted to the steady-state case which, in many building performance contexts, is atypical. Application examples in the field of building energy simulation are the prediction of temperature and velocity fields inside large or technically complex enclosures such as atria and television studios (Markatos 1984) and the prediction of the pressure field around a building (Haggkvist et al 1989).

The Zonal Method

A method in which a building and its plant are treated as a collection of nodes representing rooms, parts of rooms and plant components, with inter-nodal connections representing the distributed flow paths associated with cracks, doors, ducts and the like. The assumption is made that there is a simple, non-linear relationship between the flow through a connection and the pressure difference across it. Conservation of mass for the flows into and out of each node leads to a set of simultaneous, non-linear equations which can be integrated over time to characterise the flow domain.

In the context of combined heat and mass flow simulation it is the zonal method which has proved (for the present at least) to be most commensurate with the modelling approach adopted by the ESP system. The reasons for this are threefold. Firstly, the number of nodes involved - say some 100-200 in a moderately sized building - is considerably less than employed in a CFD approach and so the additional CPU burden is acceptable. Secondly, there is a strong relationship between the nodal networks which represent the fluid regime and the thermal counterpart. This means that the information demands of the energy balance formulations can be directly satisfied. And, finally, the technique can be readily applied to combined multi-zone buildings and multi-component, multi-network plant systems.

It is the zonal method then which has been employed for several years as the basis of the air flow element of the ESP system and which underpins recent developments which have led to an improved equation solver and extensions of the technique to plant systems in general. Within ESP these developments are made available to a user via ESPmfs for use in cases where buoyancy effects are time-invariant, and as an integral encapsulation within ESPbps, the main simulation module, for use in cases where buoyancy has a strong temporal dimension.

This paper describes the theoretical basis of the ESP approach to fluid flow simulation in terms of the flow equation types offered and the underlying numerical solution strategy. The integration of the algorithm within the ESPbps numerical processing scheme is then described to demonstrate the technique employed to achieve combined heat and fluid flow in building/ plant systems. Finally, and briefly, the paper illustrates the application of the approach in practice.

2 THE APPROACH IN OUTLINE

Within the ESP approach, during each simulation time step, the problem is constrained to the steady flow (possibly bi-directional) of an incompressible fluid along the connections which represent the building/ plant mass flow network when subjected to certain boundary conditions regarding pressure and/ or flow. The problem reduces therefore to the calculation of fluid flow through these connections with the nodes of the network representing either internal or boundary pressures. This is achieved by an iterative mass balance approach in which nodal pressures are adjusted until the mass residual of each internal node satisfies some user-specified criterion. The flow network may consist of several sub-networks and is not restricted to one type of fluid. However, all nodes and components within a sub-network must relate to the same fluid type.

Information on potential mass flows is given by a user in terms of node descriptions, fluid types, flow component types, interconnections and boundary conditions. In this way a nodal network (or perhaps several decoupled sub-networks) of connecting resistances is constructed. This may then be attached, at its boundaries, to known pressures or to pressure coefficient sets which represent the relationship between free-stream wind vectors and the building external surface pressures to result.

Nodes may represent rooms, parts of rooms, plant components, connection points in a duct or in a pipe, ambient conditions and so on. Fluid flow components correspond to discrete fluid flow passages such as doorways, construction cracks, ducts, pipes, fans, pumps, etc. As an example Figure 1 shows a sketch of a part of a building consisting of two rooms, some connections between the rooms, a radiator heating system connected to one zone and an air heating system connected to the other zone. In this case the building and plant configuration contains at least two mass flow networks - one for the air and one for the water. One possibility with respect to the translation of this configuration into a nodal scheme is indicated by the dots.

2.1 Node Definition

Nodes are characterised by several data items including an identifier, the fluid type, the node type, the height above some arbitrary datum, temperature (for use only in ESPmfs) and several supplementary parameters dependent on the node type (see later). At the present time only two

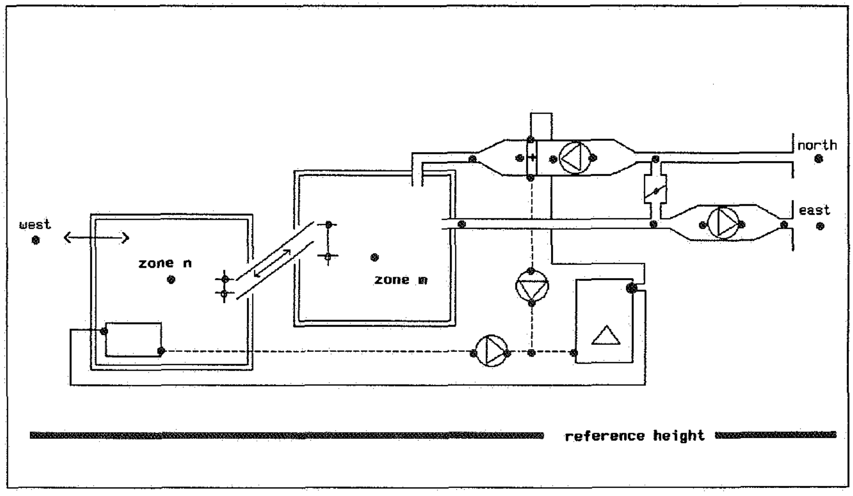

Figure 1 Example building and plant schematic

fluid types are supported - air and water - although additional types will be developed as required. The nodal types currently on offer are summarized in Table 1.

	Type	Supplementary data
0	Internal; unknown pressure	None
1	Internal; known pressure	total pressure (Pa)
2	Boundary; known pressure	1) total pressure (Pa) 2) node temperature flag, indicating 0 : temperature is some constant 1 : temperature equals outside air temperature
3	Boundary; wind pressure* temperature = outside air	1) wind pressure coefficients index 2) surface azimuth (° clockwise from North)

Table 1 Mass flow network node types

The nodes of the network represent either internal or boundary pressures. The difference is that only internal nodes are subjected to the mass balance tracking. Note that in the present context 'internal' is not necessary equivalent to 'inside' nor does 'boundary' necessarily equate to 'outside'. Usually the pressure at an internal node is unknown, although it can be treated as a known parameter as would be required, for example, in the case of an expansion vessel in a hydronic radiator system. An interesting possibility is that this node type can be used in an air infiltration problem to mimic a pressurization test, in order to compare the overall leakage characteristic with measurements.

2.1.1 Wind Induced Pressures

Pressures at boundary nodes can be specified or they can be declared to be wind induced. In the latter case a reference is made to an appropriate pressure coefficient set as held in ESP's pressure coefficients database. At run-time the pressure coefficient, appropriate to the prevailing wind direction, is used to generate the surface pressure due to the wind:

$$P_i = C_{p,i,d}\ 1/2 \rho V_{rd}^2 \quad (Pa) \tag{1}$$

where $C_{p,i,d}$ is the pressure coefficient for a surface location i corresponding to wind from direction d, ρ is the air density (kgm^{-3}) and V_{rd} is the local wind speed (ms^{-1}) from direction d at some reference level r (usually equal to the building height). The ratio between the local wind speed and the wind speed as read from the climate file, is termed the wind speed reduction factor. This reduction factor accounts for any difference between measurement height and building height and for intervening terrain roughness. Besides direct numerical input, ESP offers several user selectable wind profiles for evaluation of the wind speed reduction factor:

- power law wind profile (AIVC 1986):

$$\frac{U_l}{U_{10}} = K\, z_l^{\,a} \quad (-) \tag{2}$$

  where U_l is the local wind speed at a height z_l m above the ground (ms^{-1}), U_{10} is the wind speed measured in open countryside (ms^{-1}) at a standard height of 10 m and K, a are terrain dependent constants (see Table 2)

- logarithmic wind profile (Simiu & Scanlan 1986):

$$\frac{U_l}{U_m} = \frac{U_{*,l}}{U_{*,m}} \left[\ln\frac{z_l - d_l}{z_{0,l}} \Big/ \ln\frac{z_m - d_m}{z_{0,m}} \right] \quad (-) \tag{3}$$

$$\frac{U_{*,l}}{U_{*,m}} \approx \left[\frac{z_{0,l}}{z_{0,m}} \right]^{0.1} \quad (-) \tag{4}$$

  where U_m is the wind speed measured at the meteo site (ms^{-1}) at a height of z_m m above the ground, U_* is the atmospheric friction speed (ms^{-1}), z_0 is the terrain dependent roughness length (m) and d is the terrain dependent displacement length (m) (see Table 2)

- Lawrence Berkeley Laboratory (LBL) model wind profile (AIVC 1986):

$$\frac{U_l}{U_m} = \frac{\alpha_l\, (z_l/10)^{\gamma_l}}{\alpha_m\, (z_m/10)^{\gamma_m}} \quad (-) \tag{5}$$

  where α, γ are terrain dependent constants (see Table 2).

Terrain	K	a	z_0	d	α	γ
Open flat country	0.68	0.17	0.03	0.0	1.00	0.15
Country with scattered wind breaks	0.52	0.20	0.1	0.0		
Rural			0.5	0.7 h	0.85	0.20
Urban	0.35	0.25	1.0	0.8 h	0.67	0.25
City	0.21	0.33	> 2.0	0.8 h	0.47	0.35

Table 2 Typical values terrain dependent parameters (h = building height)

Compared with both the power law profile and the LBL wind profile, the logarithmic wind profile is to be preferred because it is based on physical laws rather than on an empirical formulation. It should be noted however that all the wind profiles above are actually only valid for heights over $(20 * z_0 + d)$ and lower than 60 ... 100 m. The layer below $(20 * z_0 + d)$ is often referred to as the urban canopy. Here the wind speed and direction is strongly influenced by individual obstacles and can only be predicted through wind tunnel experiments or simulations undertaken by a CFD model. If these are not available, great care must be taken although, depending on the problem on hand, a high or low estimate of the wind speed reduction factor may be made to characterise the problem. For example, in the case of an energy consumption and infiltration problem it is safer to use a high estimate of the wind speed reduction factor (for example wind speed evaluated at a height of $(20 * z_0 + d)$). Alternatively, in the case of an air quality or overheating and ventilation problem it is probably safer to use a low estimate (for example wind speed evaluated at the actual building height).
To give a numerical example: assume a building with a height of $7.5\,m$ located in an urban area (say $z_0 = 1.0\,m$ and $d = 6\,m$; ie. urban canopy thickness $\approx 26\,m$), and that the wind speed was measured at a height of $10\,m$ in an open flat country. Then the following local wind speed reduction factors at building height will result:

power law: 0.58 (-)

logarithmic law:	0.10 ... 0.73	(-)	lower/upper estimate as indicated above
LBL profile:	0.62	(-)	

2.1.2 Stack Effect

Within ESP each node is assigned a reference height. The reference height defines the mean height of the associated building zone or plant component. The node reference height may be expressed relative to any arbitrary datum level as long as this datum level is the same for all nodes in the network. The reference height is then used in the calculation of buoyancy driven flows (stack effect) in a manner similar to the approach suggested by Walton (1988).

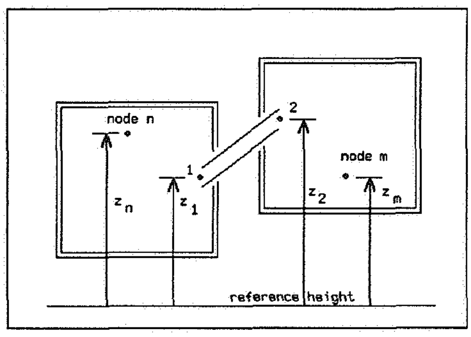

Figure 2 An example two zone connected system

Consider Figure 2 which shows two zones connected by some fluid flow component. It is assumed that each volume can be characterised by a single temperature and a single static pressure at some height relative to a common data plane. The inlet and outlet of the connecting component are at different heights relative to each other and relative to the nodes representing the volumes. Analysis of the fluid flow through a component i is based on Bernoulli's equation for one-dimensional steady flow of an incompressible Newtonian fluid including a loss term:

$$\Delta P_i = (p_1 + \rho V_1^2/2) - (p_2 + \rho V_2^2/2) + \rho g (z_1 - z_2) \quad (Pa) \tag{6}$$

where ΔP_i is the sum of all friction and dynamic losses (Pa), p_1, p_2 are entry and exit static pressures (Pa), V_1, V_2 are entry and exit velocities (ms^{-1}), ρ is the density of the fluid flowing through the component (kgm^{-3}), g is the acceleration of gravity (ms^{-2}) and z_1, z_2 are the entry and exit elevations (m). This equation defines a sign convention for the direction of flow: positive from point 1 to point 2 (or n to m).

Equation 6 can be simplified by combining related terms. Dynamic pressures are the $\rho V^2/2$ terms, and total pressure is defined to be the sum of static pressure and dynamic pressure - that is $P = p + \rho V^2/2$. If nodes n and m represent large volumes (for example a room), the dynamic pressures are effectively zero. If the nodes represent some point in a duct or pipe network, there will be a positive dynamic pressure. The pressures at the inlet and outlet of the flow component can be related to the node pressures by the hydrostatic law:

$$P_1 = P_n + \rho_n g (z_n - z_1) = P_n - \rho_n g h_1 \quad (Pa) \quad \text{where } h_1 = z_1 - z_n \quad (m) \tag{7}$$

$$P_2 = P_m + \rho_m g (z_m - z_2) = P_m - \rho_m g h_2 \quad (Pa) \quad \text{where } h_2 = z_2 - z_m \quad (m) \tag{8}$$

The relative heights, h_1 and h_2, are a convenient way of expressing the flow component inlet and outlet heights. For example, it is quite common for flow components in the building fabric to only differ with respect to inlet and outlet heights relative to the zone heights. On the other hand, if the flow component is part of a duct or pipe network, the relative heights will be zero. Equation 6 can thus be reduced to:

$$\Delta P_i = P_n - P_m + \rho g(z_n + h_1 - z_m - h_2) - \rho_n g h_1 + \rho_m g h_2 \quad (Pa) \tag{9}$$

The terms $[\rho g(z_n + h_1 - z_m - h_2) - \rho_n g h_1 + \rho_m g h_2]$ can be collectively called the stack pressure, PS_i, acting on component i:

$$PS_i = \rho_n g(z_n - z_m) + h_2 g(\rho_m - \rho_n) \quad (Pa) \quad \text{(flow in positive direction)} \tag{10}$$

$$PS_i = \rho_m g(z_n - z_m) + h_1 g(\rho_m - \rho_n) \quad (Pa) \quad \text{(flow in negative direction)} \tag{11}$$

2.2 Components Definition

A flow component is characterised by an identifier, a type code (indicating duct, pipe, pump, crack, doorway, etc.) and a number of supplementary data items defining parameters associated with that component type. When a certain flow component is repetitively present in the network, it need only be defined once. The currently supported fluid flow component types are summarized in Table 3. Detailed information can be found elsewhere (Hensen 1990).

Code	Type	Parameters
10	Power law volume flow resistance element	3
15	Power law mass flow resistance element (definition 1.)	3
17	Power law mass flow resistance element (definition 2.)	3
20	Quadratic law volume flow resistance element	3
25	Quadratic law mass flow resistance element	3
30	Constant volume flow rate element	2
35	Constant mass flow rate element	2
40	Common orifice flow element	3
50	Laminar pipe flow element	3
110	Specific air flow opening	1
120	Specific air flow crack	2
130	Specific air flow door	4
210	General flow conduit (duct or pipe)	6
220	Conduit ending in converging 3-leg junction & C = f(q/qc)	13
230	Conduit starting in diverging 3-leg junction & C = f(q/qc)	13
240	Conduit ending in converging 4-leg junction & C = f(q/qc)	17
250	Conduit starting in diverging 4-leg junction & C = f(q/qc)	17
310	General flow inducer (fan or duct)	7
410	General flow corrector (damper or valve)	17
420	Flow corrector with polynomial local loss factor	16
450	Ideal (frictionless) flow controller	8

Table 3 Currently available fluid flow component types

Within ESP each flow component has a subroutine counterpart which is used to generate the flow and flow derivative at each iteration. As an example, consider the type 420 component: this is a special case of a valve/damper which is described in terms of a variable dynamic local loss factor C - that is, the valve/damper is approached as if it were a conduit with local dynamic losses dependent on the correctors relative position (ie. valve stem displacement or damper blade angle). The mass flow rate $\dot{m}$ is calculated from:

$$\dot{m} = A \left[\frac{2\rho\Delta P}{C} \right]^{1/2} \quad (kg/s) \tag{12}$$

$$\text{with } C = a_0 + a_1 H/H_{100} + a_2(H/H_{100})^2 + a_3(H/H_{100})^3 \quad (-) \tag{13}$$

where A is the cross-sectional area containing the corrector (m^2), C is a factor representing local dynamic losses (-), H/H_{100} is the relative valve/damper position (-), and a_i are fit coefficients (-).

Within ESP this equation will be used to determine the related branch flow at each iteration step. As will be explained later the ESP solver also requires the partial derivative which for the case of Equation 12 is given by:

$$\frac{\partial \dot{m}}{\partial \Delta P} = \frac{.5\,\dot{m}}{\Delta P} \quad (kg/s/Pa) \tag{14}$$

If ΔP becomes too small, and in the case of flow components for which an analytical expression for the derivative does not exist, ESP will determine the value of the derivative by numerical approximation:

$$\frac{\partial \dot{m}}{\partial \Delta P} \approx \frac{\dot{m} - \dot{m}^{\%}}{\Delta P - \Delta P^{\%}} \quad (kg/s\ Pa) \tag{15}$$

where % denotes the previous iteration step value.

To be able to use this flow component as the actuator of a flow control mechanism, necessitates that there is also a control signal and some control law which translates sensor output into actuator input. The control law is described by the user definable parameters: day type index, start hour, finish hour, valve position outside control period, signal lower limit, relative position at low signal, signal upper limit, relative position at high signal, hysteresis, and sensed property index indicating one of: any building/plant control signal generated by ESPbps (only available in the encapsulated version of ESPmfs), nodal temperature, signed or absolute nodal temperature difference, nodal pressure, signed or absolute nodal pressure difference, signed or absolute mass flow rate, wind speed, wind direction, diffuse or direct solar radiation, or relative humidity of outdoor air.

2.3 Defining Networks

The connections data defines the flow network. Each connection is described in terms of the name of the node on its (arbitrarily declared) positive side, the height of the positive linkage point relative to the node on the positive side, the name of the node on the (arbitrarily declared) negative side of the connection, the height of the negative linkage point relative to the node on the negative side, the name of the connecting flow component and supplementary data which depends on the flow component selected. Note that more than one connection may exist between two nodes so that a connection joining node A to B is different from one joining B to A. The concept of a connection having a positive side and a negative side is used to keep track of the direction of fluid flow. For most mass flow component types, uni-directional fluid flow will result (in either direction). However, some component types may represent bi-directional fluid movement - for example in the case of a doorway where, due to the action of small density variations, over the height, bi-directional flow may exist.

3 NETWORK SOLUTION

Each fluid flow component, i, relates the mass flow rate, $\dot{m}_i$, through the component to the pressure drop, ΔP_i, across it. Conservation of mass at each internal node is equivalent to the mathematical statement that the sum of the mass flows must equal zero at such a node. Because these flows are non-linearly related to the connection pressure difference, solution requires the iterative processing of a set of simultaneous non-linear equations subjected to a given set of boundary conditions. The technique employed by ESP is to assign an arbitrary pressure to each internal node to enable the calculation of each connection flow from the appropriate connection equation. The internal node mass flow residuals are then computed from:

$$R_i = \sum_{k=1}^{K_{i,i}} \dot{m}_k \quad (kg/s) \tag{16}$$

where R_i is the node i mass flow residual for the current iteration (kgs^{-1}), $\dot{m}_k$ is the mass flow rate along the kth connection to the node i (kgs^{-1}) and $K_{i,i}$ is the total number of connections linked to node i.

The nodal pressures are then iteratively corrected and the mass balance at each internal node is re-evaluated until some convergence criterion is met. The method used in ESP is based on an approach suggested by Walton (1988). This approach was implemented and tested in an earlier version of ESP and shown to result in considerable speed improvements as evidenced in table 4 (Clarke & Hensen 1988).

Problem	Original Solver		New Solver		24 hr Iteration Ratio	CPU Ratio
	CPU Seconds	Iterations 1st hr - 24 hrs	CPU Seconds	Iterations 1st hr - 24 hrs		
1. atria	3087	6363 - 152117	55	137 - 522	291	56
2. house 1	377	374 - 27863	17	29 - 459	60	21
3. house 2	48	146 - 2510	23.2	11 - 105	23	2
4. 2 zone	9	309 - 2376	3.6	16 - 287	8	2
5. 3 zone	3	27 - 358	2.5	4 - 90	3	1
6. Trombe	2168	14009 - 122754	50.2	29 - 474	258	43
		1st hr - 2nd hr		**1st hr - 2nd hr**	**1st - 2nd hr**	
7. large		13270 - 25318		24 - 1	552 - 25318	

Table 4 Bench-mark results. All runs were performed on a SUN 3/50 and correspond to a one day (24 hour) simulation

The latest ESP model has a further enhanced solver which has resulted in additional iteration reductions. However, at the time of writing no bench-mark results were available. The solution method is based on a Newton-Raphson technique applied to the set of simultaneous nonlinear equations (for example see Conte and De Boor 1972). With this technique a new estimate of the vector of all node pressures, $\mathbf{P}^*$, is computed from the current pressure field, $\mathbf{P}$, via:

$$\mathbf{P}^* = \mathbf{P} - \mathbf{C} \tag{17}$$

where the node pressure correction vector, $\mathbf{C}$, is determined on the basis of a simultaneous solution of a Jacobian matrix which represents the nodal pressure corrections in terms of all branch flow partial derivatives. The pressure corrections vector $\mathbf{C}$ is given by:

$$\mathbf{C} = \mathbf{R}\,\mathbf{J}^{-1} \tag{18}$$

where $\mathbf{R}$ is the vector of nodal mass flow residuals and $\mathbf{J}^{-1}$ is the inverse of the square Jacobian matrix (N*N for a network of N nodes) whose diagonal elements are given by:

$$J_{n,n} = \sum_{k=1}^{K_{n,n}} \left[\frac{\partial \dot{m}}{\partial \Delta P} \right]_k \quad (kg/s\,Pa) \tag{19}$$

where $K_{n,n}$ is the total number of connections linked to node n and ΔP_k is the pressure difference across the kth link. The off-diagonal elements of $\mathbf{J}$ are given by:

$$J_{n,m} = \sum_{k=1}^{K_{n,m}} - \left[\frac{\partial \dot{m}}{\partial \Delta P} \right]_k \quad (kg/s\,Pa) \tag{20}$$

where $K_{n,m}$ is the number of connections between node n and node m. This means that - for internal nodes - the summation of the terms comprising each row of the Jacobian matrix are identically zero.

ESP currently uses LU decomposition with implicit pivoting (also known as Crout's method with partial pivoting) for solution of the matrix equation **J C** = **R** for the unknown pressure correction vector **C**. The implementation in use by ESP originates from an algorithm by Press (et al. 1986). In this case the matrix **J** is decomposed to a lower triangular matrix **L** and an upper triangular matrix **U**, such that **L U**=**J**. This decomposition is used to solve the linear set:

$$\mathbf{J}\,\mathbf{C} = (\mathbf{L}\,\mathbf{U})\,\mathbf{C} = \mathbf{L}\,(\mathbf{U}\,\mathbf{C}) = \mathbf{R} \tag{21}$$

by first solving, by forward substitution, for the vector **Y** such that **L Y** = **R** and then solving (by back substitution) **U C** = **Y**. The advantage is that both substitutions are quite trivial. Pivoting is used to make the method numerically stable.

As a future possibility, sparse matrix methods could be used to reduce further the storage and execution time requirements.

It should be noted that it is quite easy to define a mass flow network which has no unique solution. One requirement for solution is that at least one of the node pressures is known. A second requirement is that all nodes must be linked, through some path, to a known pressure node.

Conservation of mass at each internal node provides the convergence criterion. That is, if $\sum \dot{m}_k = 0$ for all internal nodes for the current system pressure estimate, the exact solution has been found. In practice, iteration stops when all internal node mass flow residuals are below one of two user definable thresholds: ERRMAX the largest percentage residual flow error, or FLOMAX the largest absolute residual flow error.

In some cases, large corrections for the successive pressure correction applied to any node during the iteration process may cause a numerical instable situation. Therefore, ESP offers PMAX a user definable maximum pressure correction applied to any node during the iteration process.

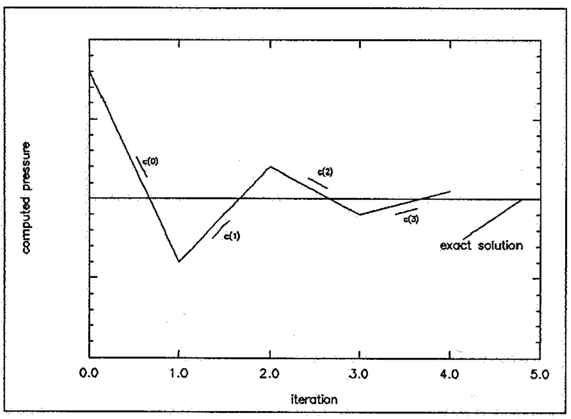

Figure 3 Example of successive computed values of pressure and oscillating pressure correction at a single node

As noted by Walton (1988), there may be occasional instances of low convergence with oscillating pressure corrections on successive iterations at a single node. In the case shown in Figure 3, each successive pressure correction is a constant ratio of the previous correction - that is $C_i = -0.5 * C_i^{\%}$ ($^{\%}$ denotes the previous iteration step). In a number of tests the observed oscillating corrections came close to such a pattern. By assuming a constant ratio, it is simple to extrapolate to the 'final solution':

$$P_i^* = P_i - C_i \,/\, (1-r) \quad (Pa) \tag{22}$$

where r is the ratio of C_i for the current iteration to its value in the previous iteration. The factor $1/(1-r)$ is called a relaxation factor. The extrapolated value of node pressure can be used in the next iteration. If it is used, then r is not evaluated for that node in the following iteration but only in the one thereafter. In this way, r is only evaluated with unrelaxed pressure correction values. This process is similar to Steffensen iteration (Conte and De Boor 1972) which may be used with a fixed point iteration method for individual non-linear equations. The iteration correction method presented above gives a variable and node dependent relaxation factor. When the solution is close to convergence, Newton-Raphson iteration converges quadratically. By limiting the application of the relaxation factor to cases where r is less than some value (ESP's user definable parameter STEFFR) such as -0.5, it will not interfere with the rapid convergence.

4 COMBINED HEAT AND MASS FLOW SIMULATION

Within the ESP system, a fluid flow simulation may be initiated independently of the main energy simulation or pursued in tandem. In the former case the assumption is made that the flows are predominantly pressure driven and that buoyancy effects, although included, are time invariant (or user specifiable). In the latter case ESP must establish and solve the coupled, matrix equations corresponding to the heat and fluid flows within the multi-zone building and the multi-component plant. The ESP scheme, which is reported fully elsewhere (Clarke 1990), can be summarised, for any computational time-step, as follows.

- the energy balance, state-space equations corresponding to the finite volumes which represent the plant-side discretised components and distribution network are established on the basis of the latest values of the building-side state variables and plant component/ network mass flows
- this plant matrix equation is then solved by a sparse matrix technique taking into account any defined control action
- the energy balance, state-space equations corresponding to the finite volumes which represent the building-side discretised constructions, surfaces and air volumes are then established on the basis of the latest values of the plant flux inputs and building-side air flows
- this building matrix equation is then solved by a customised matrix inversion technique which employs a partitioning and ordering technique which ensures that only non-zero matrix entries are processed and which integrates control system characteristics within the solution process
- the whole-system, fluid flow equations are then solved, iteratively, by the technique described earlier in this paper, utilising the newly established building and plant-side state variables to estimate the buoyancy effects
- if required, time-step control can be activated to prevent the evolution of time in cases where the newly computed state-variables differ markedly from the latest values assumed when the matrix equations were established
- finally, the simulation clock is incremented and the process repeats

Since the time constant associated with the state-space equations representing plant-side finite volumes are often an order of magnitude smaller than their building-side counterpart, a facility is provided to allow the plant system equations to be established and solved at a greater frequency than the building system equations.

5 A CASE STUDY

In a paper by Emslie (1990) four case studies are presented involving air infiltration modelling in buildings with ESP. These case studies were selected from a portfolio of computer modelling projects carried out by the Energy Design Advisory Service (Emslie & Chalmers 1988).

One of the case studies was concerned with thermal upgrading of Ladywell high rise, a tower block in Glasgow comprising a number of similar 1 and 2 bedroom flats. A detailed computer

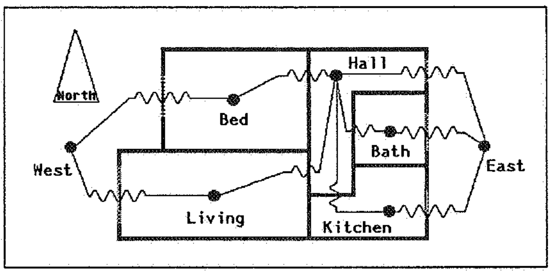

Figure 4 Plan view and air leakage network of 1 bedroom flat

simulation study was commissioned to predict the most cost-effective options for upgrading in terms of occupant comfort, condensation risk, heating capacity required and energy consumption & cost. As part of the study it was necessary to predict 'design' air infiltration rates for all rooms in two cases: (i) leaving existing ill fitting single glazing units unmodified or, (ii) replacing with new double glazed units incorporating trickle ventilators. Given the site microclimate details, it was decided that infiltration rates be predicted for two wind directions.

flat	glazing	wind	living	kitchen	hall	bath	bed 1	bed 2
1	single	west	2.3				1.6	
		east		3.1	1.8	3.9		
	double	west	0.6				0.4	
		east		0.4	1.2	0.5		
2	single	west	2.0	2.5			0.1	0.2
		east	0.1		2.0	5.1	0.3	0.5
	double	west	0.5	0.5			0.1	0.2
		east	0.0		1.6	1.1	0.0	0.1

Table 5 Ladywell high rise: predicted 'design day' average air infiltration rates (air changes per hour) for 1 and 2 bedroom flats

Figure 4 shows a plan view of the 1 bedroom flat with the distributed air leakage network (consisting of windows, cracks, doors) superimposed. Design day simulation using representative hourly weather data for Glasgow gave hourly nodal pressure distribution and leakage path's air flow rates. ESP offers facilities to retrieve this data both tabular and graphical. It is also possible to reduce this data to predicted average air infiltration rates for each flat type as listed in Table 5.

These simulations demonstrated the effect on air infiltration to be expected from window replacement. With regard to the global study, which involved detailed thermal modelling as well as air flow modelling, it was concluded that in this case window replacement outperformed proposed wall insulation upgrading as a means of energy conservation.

6 CONCLUSIONS

This paper has demonstrated how a mass flow network method can be used to provide a unified model of major building and plant fluid flows. It was shown how this model can be used for the simulation of coupled heat and mass flows in buildings. The performance of the model indicates that it is practical to solve the building/ plant heat and mass flow network in detail. Solution of complex fluid flow networks for problems involving many time steps is now feasible on current small computers.

It is felt that the model reflects the current state of the art in the field of network modelling approach to simulation of coupled heat and mass flows in buildings. Development of the model did reveal however that research is still needed in several areas. These include development of additional fluid flow component models (especially improved large opening models), modelling of intrazone effects by simplified methods and by integration with CFD modelling methods, expansion of the wind pressure database, expansion of the actual building and plant components 'database', and experimental validation of the simplifying assumptions in the flow component models and the network method.

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The authors are indebted to George Walton who so willingly shared his theoretical approach to the simulation of air flow in buildings.

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Discussion

Paper 45

J.Axley (MIT, USA)

The source reference for Figure 2 in the paper is erroneous.

J.Hensen (Technische Universiteit, Eindhoven, Netherlands)

In our paper we reference the AIRNET report by G. Walton as the source for the figure and for the way in which stack effects are handled. Since the report in question did not state otherwise, we naturally assumed this to be the original source. From Walton's final paper on AIRNET at the 1989 ASHRAE summer meeting, it is now clear that a number of the ideas used in that model also originate from Dr. J. Axley.

J.Axley (MIT, USA)

There is a need to provide linear flow equations for low flow ranges of components to avoid near singularity of the system Jacobian thereby avoiding convergence problems for low flow cases.

J.Hensen (Technische Universiteit, Eindhoven, Netherlands)

in ESPmfs, low flow cases do not impose problems. Even zero flow cases are handled correctly. Therefore we do not think that it is necessary to provide separate expressions for low flow cases. Furthermore, linear equations can only be provided for a number of component types (it is for instance impossible in case of a black box (perhaps look-up table) component.

J.Axley (MIT, USA)

It can be proven that the system of non-linear flow equations results in a positive definite nonsingular matrix. The solution method will be numerically stable, regardless of the fact whether pivoting is used or not.

J.Hensen (Technische Universiteit, Eindhoven, Netherlands)

We agree on the first part of the statement. Regarding the necessity of (partial) pivoting we do not agree, and refer to the following quote from Press et al (1986) from their Section 2.3 on the LU decomposition method: "Pivoting is absolutely essential for the stability of Crout's method. Only partial pivoting (interchange of rows) can be implemented efficiently. However this is enough to make the method stable"...

J.Axley (MIT, USA)

Linear expressions used for the low-flow range provide a convenient means to initialize solution strategy.

J.Hensen (Technische Universiteit, Eindhoven, Netherlands)

See question two above. Furthermore, we found that in ESPmfs's case of successive time steps the pressures for the previous time step usually give a pretty good initial pressure vector for the next step.

J.Axley (MIT, USA)

Did you have a look at the paper/poster by R.A. Grot and J.Axley on "Structure of models for the prediction of air flow and contaminant dispersal in buildings"?

J.Hensen (Technische Universiteit, Eindhoven, Netherlands)

Because the paper by Axley and Grot was not included in the Conference preprints and at the time of presentation it was very crowded at the poster session, I did not have any real opportunity to study the paper/poster in great detail. Furthermore, I lost the point which Dr.Axley was trying to make with his comment. From another recent paper by Axley and Grot (1989) I think it might have been the fact that we do not use an integral element formulation approach to coupled airflow analysis and thermal analysis which is described in that paper. Although I think that what is described is a very interesting approach - especially in view of current software engineering developments like the object orientated approach - we did not consider a similar approach at this point in time. The main reasons being: - although the emphasis in our paper was on simulation of air flow, ESPmfs is really developed for a wider field of interest including different fluids and a broad spectrum of plant components. I am not sure whether it will be easy or even possible to establish in all cases the kind of element equations which Axley and Grot describe (e.g. in case of a component which is only described in terms of certain input/output relationships) - ESP uses 5 basic *matrix equations: for building side energy, plant side energy, plant side 1st phase mass flow rate, plant side 2nd phase mass flow rate and for the combined building and plant mass flow network, respectively. Combining these matrix equations into 1 is particularly useful if it is possible to use a direct solution method. In case of non-linear relationships between state-variables or non-linear control strategies - like we have - a direct method is not possible. If you use an indirect (say iterative) method, it does not seem to make much difference whether you use one combined or several separate matrix equations. On the other hand, if you do use the separate matrix approach it is very easy to introduce mixed-frequency and variable time stepping schemes. In that case it is also quite easy to exclude certain matrix equations if they are of no interest in a certain problem context (e.g. the mass flow network matrix in case the flows are constant) - introducing the method described by Axley and Grot would involve complete restructuring of the main simulation engine, while incorporating the method as described in our paper was relatively straight forward.*

B A Fleury, ENTPE-LASH, France

I wonder about the difference between ESPair, the air flow network model already present in the ESP system, and the model described in the current paper, ESPmfs. Also, the coupling between energy and air flow.

J Hensen (Technische Universiteit Eindhoven, Holland)

Some of the differences between ESPair and ESPmfs are already indicated above. ESPmfs is a completely new development, which is set up in a highly modular fashion to facilitate easy introduction of new flow component models, alternative matrix solvers, etc. When compared to ESPair, the main differences, in terms of functionality are: ESPmfs is a general building and plant fluid mass flow network solver, whereas ESPair is restricted to air flow through a very limited number of building opening type air leakage models. ESPmfs offers more node types and eg. alternative theories with respect to wind pressure calculation. The stack effect calculation is quite different. The coupled heat and mass flow simulation is now extended to the plant side. In terms of solution method the main difference is that ESPair uses a node wise Newton-Raphson method, while ESPmfs uses a whole network NR approach. In terms of performance the main differences are illustrated by the speed improvements as indicated in Table 4. The coupling between energy flow and air flow is basically established by substituting the appropriate fluid mass flows in the whole building energy matrix coefficients with the mass flows calculated by the in ESPbps incorporated version of ESPmfs at each finite calculation time step.