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Using CFD Techniques to Evaluate Wind Pressure Distribution for Air Infiltration Analysis

by Alison Cooper, Guest Researcher at the AIVC, Summer 1992

Summary

This report describes the work carried out at the Air Infiltration and Ventilation Centre (AIVC) by Alison Cooper during July and August 1992 as part of the University of Warwick Science Park project scheme.

Pressure distributions around buildings are important factors affecting the air infiltration and ventilation of a building and consequently energy.

Existing methods of determining pressure coefficients are costly both in terms of time and resources. This report aims to show the benefits of using a computational fluid dynamics (CFD) program in this field.

Work was carried out to predict the pressure distribution around a sheltered building. The CFD program was used to investigate how pressure coefficients vary with building separation and the degree of shelter offered by an upwind building.

It is hoped to show that this method of calculation will prove to be more time efficient and economical than those in preferred use at the present time.

Introduction

Airflow patterns and the corresponding pressure distributions are important factors which need to be considered in predicting the ventilation characteristics of a building at the design stage. Such pressure distributions, usually expressed in the form of pressure coefficient, have a direct effect on ventilation and air infiltration and consequently on the movement of any pollutants in the surrounding air and the energy efficiency of the building.

Surface pressure coefficients are defined as:

$$C_p = \frac{P_s}{P_f}$$

where Ps is the pressure measured on the building surface taking into account reference static pressure at this point measured in the absence of any obstacles. P_f is the free stream pressure at building height.

High building pressures are undesirable due to their effect on the air leakage of a building and subsequently the increased transfer of pollutants and

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reduction in efficiency of ventilation and heating systems. The effects of wind on a building therefore need to be considered carefully in order to be able to create a design which will provide the best conditions for the occupants and be energy efficient.

Pressure coefficients are known to vary with building size and shape, wind direction and the presence of any nearby obstacles.

At present any methods used to obtain such information involve either actual measurement on full scale buildings or wind tunnel test on scale models.

In the former case the obvious disadvantage is that construction has to be complete before any form of testing can take place. Also experimental measurements of this kind are affected by the continuous fluctuation in weather conditions. This will tend to give a variation in results and reduces the possibility of being able to accurately relate and compare results from a number of investigations. As a consequence such forms of testing tend to be both very costly and time consuming.

In the case of wind tunnel testing, restrictions are placed on the model size that can be used and this contributes to the measurement process being very repetitive, time consuming and therefore costly.

However at present wind tunnel tests are still considered to be the best method for predicting C_p values due to their relative convenience and the lack of a more convenient alternative.

The emergence of computational fluid dynamics (CFD) programs has led to a new area of research into building design. This method of calculation uses mathematics alone to solve the equations for energy and momentum for a given problem.

A previous study on numerical simulation of pressure fields around buildings was undertaken by Häggkvist and Taesler, BFR, Sweden, 1987. The authors of the report on this work looked at simulations using PHOENICS and compared their results with some measurements from wind tunnel studies. The main conclusions formed were that this sort of simulation gave a promising new approach to looking at building pressures. More specifically the numerical simulations used appeared to give shorter recirculation zones on the leeward side of buildings than those observed in the wind tunnel and that deviations in pressure coefficient were greater when considering houses in a group rather than for an exposed building.

This project attempts to use one particular CFD program to investigate the changes in pressure coefficients due to the sheltering effects of an upwind obstacle.

Project Description

The aim of this project is to use the mathematical modelling techniques of a CFD program to predict the pressure coefficients on all faces of a sheltered building.

This method was employed to show how such pressure coefficients vary with:

a) distance from an upwind building,

b) form of the upwind obstacle.

These main objectives were chosen since values of pressure coefficient over the walls of a building are known to depend on a number of different factors including shelter from nearby buildings. Different separation distances were studied in order to gain information indicating when separation distance becomes important in contributing to undesirably high C_p values. By considering changes in the form of the upwind obstacle it is intended to show that CFD techniques are able to predict changes in value and distribution of pressure coefficients when a building is only partially sheltered.

It was intended to show that with just basic knowledge of airflow and computer handling experience quite complex flow and pressure fields could be predicted for many different building configurations in a relatively short period of time.

The first part of the project was designed to test the given program for a number of different building

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arrangements in two dimensions and to show the various forms of information that could be gained from any one problem.

This involved investigating what boundary conditions needed to be imposed in order to create expected results and then to show how dimension and shape affected the flow patterns.

Once satisfied with these results the problem was extended to the more realistic three dimensional case with the purpose of calculating pressure coefficients from the output data.

Description Of Mathematical Modelling

Computer model.

The CFD program used throughout this project was EXACT3 developed at NIST (Fang and Grot, 1990).

EXACT, standing for "Explicit Time Marching Algorithm for Continuous Thermal Fluid Flow" describes a numerical method for calculating airflows within a specified domain. This numerical method is based on a finite difference technique and treats three dimensional non-isothermal turbulent flows using a k-ε turbulence method. It solves the resulting non-linear system of momentum, energy and turbulence equations by an explicit time marching technique to obtain a solution for either steady or transient flow.

The model is able to handle a number of flow conditions by specifying inflow and outflow boundary conditions and an arbitrary number of obstacles can be placed within this flow domain.

The actual working space is created using a grid system where the flow domain is subdivided into a large number of cells by a series of orthogonal lines in each of the three coordinate directions. Each of these lines is assigned a number which means any cell within the total volume can be individually identified.

The total working volume is restricted by the available computer memory size. In this case the total parameter size was confined to 32000 giving:

$NX \times NY \times NZ < 32000$

where NX is the number of cells in the x-direction,

NY is the number of cells in the y-direction,

and NZ is the number of cells in the z-direction.

Further restriction is placed on the actual working space by the need to include three false planes in each of the three directions.

Velocity components and pressures are defined at each cell location according to the prescribed boundary conditions. Figure 1 shows the staggered mesh system where velocity components are defined on the cell boundary normal to the cell face and pressures are defined at the centre of each cell.

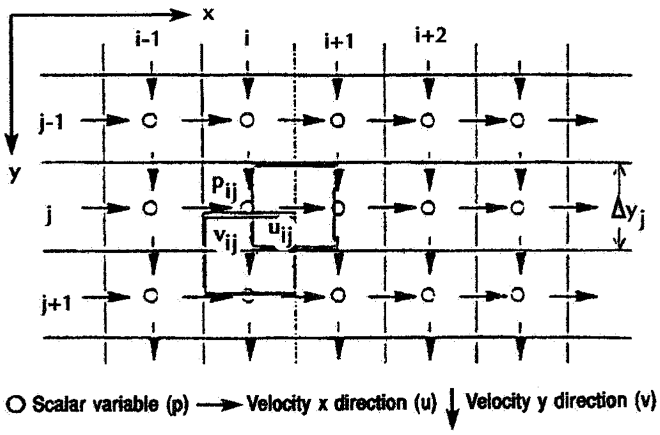


Figure 1: Staggered mesh system

Obstacles were created by specifying planes within the working domain in order to construct simulated buildings within the airflow. These planes effectively create a volume within the total working space which the program interprets as a number of dummy cells meaning variables such as velocity and pressure are assigned the value of zero within these cells.

In order to maximize the available working volume symmetric boundary conditions were imposed where it is possible to enter only data for half of the obstacle assuming that results will be symmetric about a specified boundary.

Care was taken in order to create suitable grid spacings since the lines may be spaced non-uniformly so that attention can be concentrated on regions of interest such as the edges of buildings. For regions of less importance grid spacing can be increased.

In both two dimensional and three dimensional cases it was considered satisfactory to have uniform close spacing in both y and z directions but the x spacing was increased above any obstacle height. This resulted in a suitable mesh representation on the screen.

The following indicates the parameter values and line spacings that were used:

Two dimensions
NX = 28 (7 x 0.2 , 21 x 0.1)
NY = 285 (285 x 0.1)
NZ = 4 (4 x 0.2)

Three dimensions
NX = 28 (7 x 0.2 , 21 x 0.1)
NY = 95 (95 x 0.1)
NZ = 12 (12 x 0.1)

Figure 2 shows the three dimensional flow domain and specifies planes which will create a cube as an obstacle.

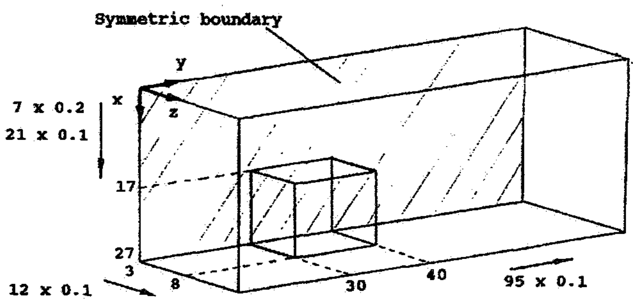


Figure 2: Three dimensional grid layout

Equations and boundary conditions.

Inflow and outflow velocities were defined by imposing the following power law velocity profile to simulate airflow in the negative y direction:

$$U_x = U_o \left(\frac{x}{\delta}\right)^{1/6}$$

where U_x = velocity at height x above boundary,

U_o = free steam velocity,

δ = boundary layer thickness,

x = height above boundary.

To simulate a fully developed atmospheric turbulent boundary layer turbulence energies (k) and the dissipation rate of turbulence energy(ε) need to be specified at the inflow point. These are defined as follows:

$$k = 0.17\, U_x^2$$

$$\varepsilon = 0.05\, k$$

Cubes were used to simulate buildings placed in the airflow field. For each building and the boundaries of the computational domain suitable velocity and wall boundary conditions had to be imposed. Flow next to such boundaries must obey boundary layer theory but some problems were encountered in trying to find which conditions were essential and those which were not.

Figure 3 shows the resulting flow pattern for buildings placed in the flow after a calculation of fifty iterations. The input data for this case only specified the presence of obstacles and although no signs of recirculation are present it was unclear whether this was due to the lack of boundary conditions or due to the short time the program was run for.

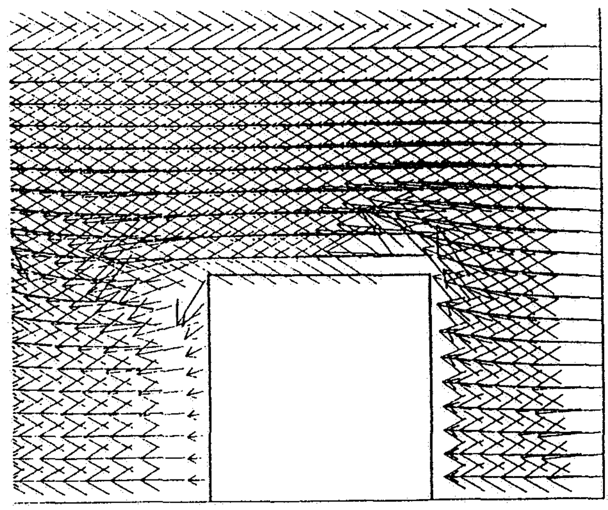


Figure 3: Flow pattern after fifty iterations - specification of obstacles only.

However after imposing more conditions on the obstacles marked differences could be seen even after only fifty iterations. Figure 4 shows the flow pattern obtained when specifying no tangential or perpendicular velocity components on each surface and wall boundary conditions following a sixth power law in accordance with the velocity profile used.

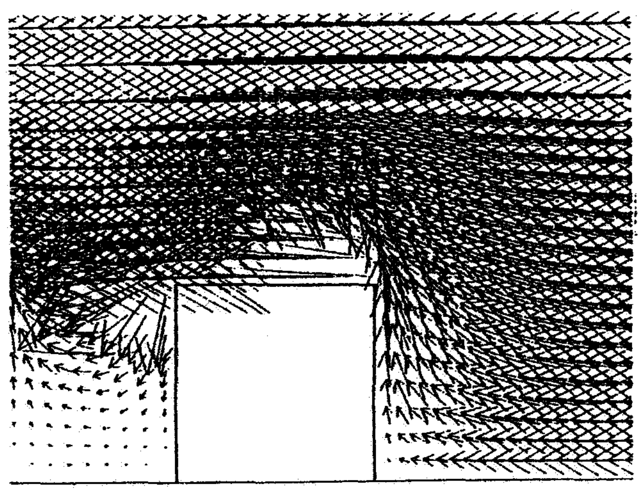


Figure 4: Flow pattern after fifty iterations-specification of velocity and wall boundary conditions.

The base of the computational domain was specified by obstacle data and conditions of no tangential or perpendicular velocity components were applied. All other boundary conditions on the extremities of the domain were defined according to the velocity profile used. This meant, for example, that the whole of the upper surface of the domain was assigned the value of free stream velocity.

These conditions were then used throughout the rest of the project after good results were achieved when the program was run for a much longer period of time.

Temperature was fixed at a constant value throughout.

One simulation in two dimensions involved the modelling of a building with a sloping roof. In order to represent inclined surfaces in the Cartesian grid system it was necessary to build the roof section from planes in a stepwise fashion.

Figure 5 indicates how such a roof section was constructed.

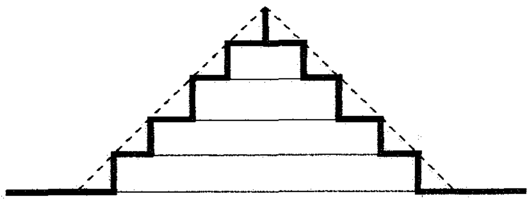


Figure 5: Construction of inclined roof section. Bold print indicates the imposition of wall boundary conditions.

On all planes used to create the roof the usual conditions of no tangential or perpendicular velocity components were defined. The external horizontal and vertical sections were defined as wall boundaries.

In order to study the effect of partial shelter on a building in three dimensions the sheltering obstacle needed to be modified. Two rectangular sections from both planes normal to the airflow were removed. Taking into consideration the symmetry conditions used this created a sheltering building with four openings in the two walls normal to the airflow.

Figure 6 indicates the location of the openings.

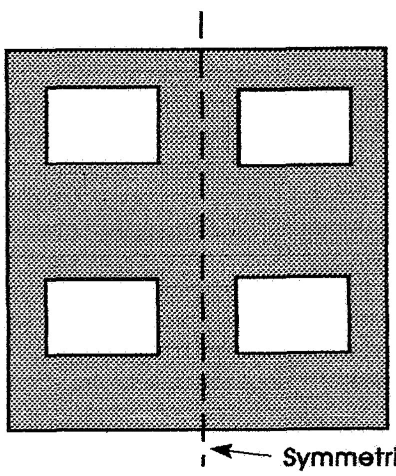


Figure 6: Positions of opening on faces normal to the airflow of the sheltering building.

The walls surrounding these openings were subject to the same boundary conditions used previously but in addition these conditions were also applied to the interior of the building. This resulted in the edges of the openings being subject to no-slip velocity conditions.

Calculating pressure coefficients.

EXACT3 produces a long output file which lists all boundary and obstacle conditions as well as standard values such as volumetric mean pressure and volumetric mean energy calculated at the end of each iteration. It also produces velocity and pressure values at a specified data output point. By entering the data output point to coincide with desired positions on the buildings it is possible to calculate pressure coefficients.

Free stream pressure is extracted from this data with the value taken away from any obstacles at approximately building height.

Results

Realistic flow patterns which compare favourably with those resulting from tests in wind tunnels and previous numerical simulations were obtained if the program was left to run for a long enough period of time. Acceptable results for two dimensions and two cubes in three dimensions were obtained after sixteen hours (overnight run). Anything more complex in three dimensions was found to require in the region of forty to fifty hours calculating time (weekend run).

Two dimensions.

A sample of results in two dimensions is shown in Appendix 1. These show velocity vectors and the associated streamlines for two building configurations. It is also possible to produce pressure and velocity contours for these examples.

From these two cases it can be seen that relative size and shape of the buildings are factors which greatly affect the airflow patterns.

Behind each building a zone of recirculating air forms and the zone behind the sheltered building is seen to extend approximately three building heights downstream.

Small recirculation zones at ground level are also shown suggesting that in a full scale construction these would be the regions where any litter would collect. The larger recirculation zones could result in the transport of any airborne pollutants into the buildings.

The second building configuration shows a particularly large recirculation zone which would be a problem in, say, a shopping centre complex. This is because the recirculating air produces regions of acceleration in both x and y directions, which can be seen from plots of velocity contours, and which would be undesirable for any shoppers entering and leaving the building.

It is shown that information can be gained which is not immediately obvious and indicates some of the problems associated with building design.

Comparison between two and three dimensions.

Flow patterns resulting from various building separations in three dimensions are shown in Appendix 2.

By comparing the two and three dimensional cases of two cubes with a 2 x building height separation it can be seen that disturbances above the buildings are generally less for the three dimensional example. This is due to the fact that air is now able to flow around the sides of the building whereas in the two dimensional case the air is confined to just one plane and when confronted with an obstacle it is forced to flow over that obstacle.

As a result recirculation zones tend not to extend so far downstream when modelling the airflow in three dimensions and recirculation also now occurs in all directions.

Comparison between these two cases exposes some of the difficulties in restricting problems to just two dimensions.

Three dimensions.

Using the results presented in Appendix 2 it is possible to see how the stagnation point on the windward face of the sheltered building is located in the upper twenty to thirty per cent of the face and as building separation increases this point moves downward.

The windward face of the sheltering building takes the full force of the airflow and has a stagnation point located approximately central.

It will be shown later how these observations relate to actual pressure distributions.

Again if an airborne pollutant is present upstream of the two buildings then it is likely to enter the sheltering building at all levels. However, for the second building the reduction in wind speed and the change in position of the stagnation point will mean the point of entry is confined to the top of the building and the likelihood of this is reduced as building separation is reduced.

Vertical distributions of wind pressure coefficients.

Appendix 3 shows the vertical distribution of pressure coefficients calculated up the centre of each face for a building in the shelter of an identical building at three different separations and compares these distributions with those calculated for an isolated building of the same type.

The main problem in determining these pressure coefficients is that the EXACT3 program calculates pressures at the centre of each cell. Therefore the results do not give pressure coefficients directly on the building surface but if the grid spacing in this region is sufficiently close then the results are good enough to compare with those taken by other forms of measurement.

Values of pressure coefficient calculated are high. For the windward face of the isolated building. But show trends which relate to those taken by wind tunnel testing.

The large peak in pressure coefficient values at the top of the windward face of the isolated building is due to high values of air velocity. For the sheltered buildings pressure coefficients are much reduced. For a separation of three building heights a peak near the top of the building is still obvious but this disappears as separation is reduced.

Leeward faces for the sheltered building show little variation from the isolated case.

Appendix 3 also shows the change in sheltering effect for a separation of two building heights when the program is used to simulate a sheltering building with the windows open. These results show the difference between full and partial shelter.

For the windward face the peak in pressure coefficients near the top of the building returns and values at all points are increased when the windows of the upstream building are opened.

Leeward faces show a slight variation in the shape of the pressure distributions but the values are very similar. This suggests the form of the sheltering building has little effect on the pressure felt on the leeward face.

The side faces are also similar in shape but values for the partially sheltered building are slightly higher than for the fully sheltered case.

Comparison with existing wind tunnel results

In wind tunnel measurements carried out by Bowen (2) a model surrounded by a high density urban area was considered. This was simulated by blocks of the same plane and attitude as the building model and of half the height. These blocks were placed around each side of the model with gaps of two building heights separating them.

These results are compared to data calculated here for separation distances of two and three building heights where obstacle and building are of the same dimension.

The profiles for the windward face, although different in terms of numerical value, do show similar characteristics. On the upper half of the building face Cp values increase and achieve a maximum at about 85-90% building height and begin to decrease as the top of the building is approached.

Values on the remaining faces again differ in value but the variation in C_p values for these faces is slight and display no significant features in either case.

Face 1

Height	x=H	x=2H	x=3H
1.OH	0.082	0.184	0.267
0.9	0.038	0.166	0.354
0.8	-0.029	0.177	0.314
0.7	-0.076	0.078	0.255
0.6	-0.103	0.0496	0.208
0.5	-0.114	0.0294	0.175
0.4	-0.112	0.016	0.157
0.3	-0.101	0.007	0.149
0.2	-0.088	0.003	0.147
0.1	-0.081	0.003	0.151

Face 3

Height	x=H	x=2H	x=3H
1.OH	-0.475	-0.447	-0.402
0.9	-0.466	-0.435	-0.405
0.8	-0.468	-0.436	-0.412
0.7	-0.467	-0.438	-0.420
0.6	-0.461	-0.438	-0.426
0.5	-0.450	-0.433	-0.428
0.4	-0.431	-0.422	-0.423
0.3	-0.408	-0.407	-0.413
0.2	-0.392	-0.395	-0.402
0.1 -	0.391	-0.393	-0.399

Face 2

Height	x=H	x=2H	x=3H
1.OH	-0.365	-0.376	-0.317
0.9	-0.368	-0.379	-0.319
0.8	-0.370	-0.380	-0.324
0.7	-0.372	-0.380	-0.326
0.6	-0.374	-0.380	-0.327
0.5	-0.375	-0.380	-0.335
0.4	-0.376	-0.379	-0.337
0.3	-0.377	-0.378	-0.328
0.2	-0.376	-0.376	-0.322
0.1	-0.375	-0.376	-0.317

Numerical results for these CFD simulations.

Conclusions

The CFD technique is not only able to predict pressure coefficients but can also give detailed airflow patterns in the form of velocity vectors and pressure and velocity contours.

Although the actual values of pressure coefficient calculated differ numerically the variation in these pressure coefficients over the building faces are in fairly good agreement with the trends shown in previous wind tunnel tests and indicate that shelter from an upstream obstacle has the most effect on windward faces.

This method of calculation has many advantages over alternative techniques. One being that experiments can be performed without the expense of undertaking construction of full scale buildings or scale models and the need to take many time consuming measurements. Conditions imposed are always fixed allowing direct comparison between different configurations without the need to take into account fluctuations in the weather and the environment.

To reduce computer processing time it is possible to confine a problem to two dimensions but results obtained show the dangers of using such a restriction. It can be assumed that two dimensional examples are sufficient to model a situation but if actual flow and pressure distributions are required then data modelling three dimensions must be used. This also has the added advantage of being able to obtain pressure distributions at different locations on all faces of a building.

One disadvantage is that more computer processing time is required as problems become more complex but compared to other methods of calculation CFD work should prove to be much less time consuming and consequently very much more economical.

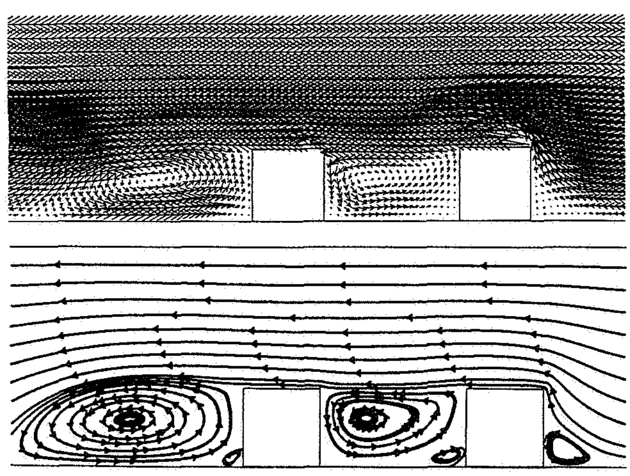
However, cooperation between building engineers and mathematicians is still needed to avoid being led by computer predictions since any flow patterns obtained may require some form of verification.

Recommendations

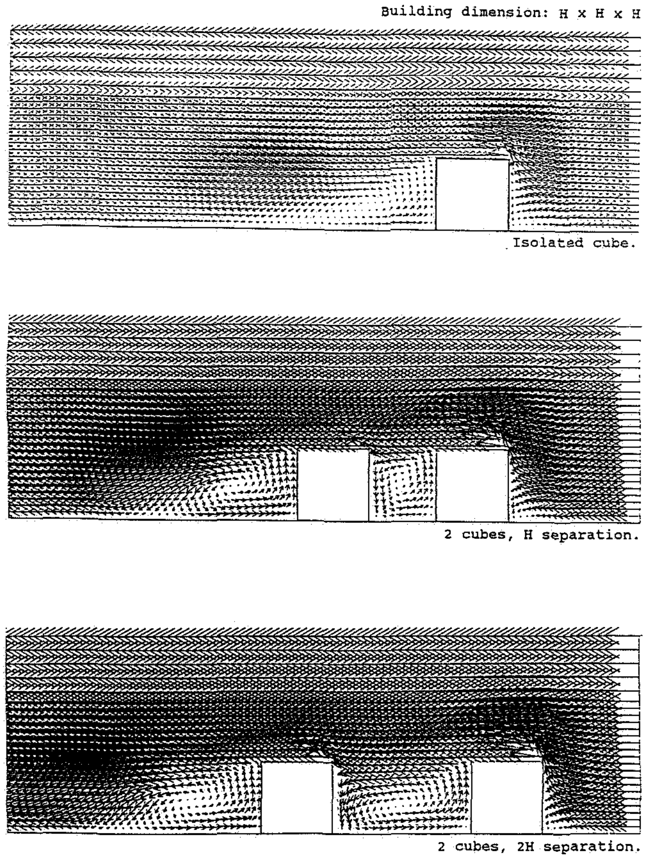
Since pressures are calculated at the centre of each cell it is important that the possible grid dependence of these results is assessed. This could be investigated by taking finer and finer spacing next to the obstacle walls in order to calculate pressures as close to the actual boundary as possible.

Once confident with the results for airflow normal to building faces it may be possible to investigate how pressure distributions change with wind angle. This was in fact attempted but was found to require more computer memory size and processing time than was available.

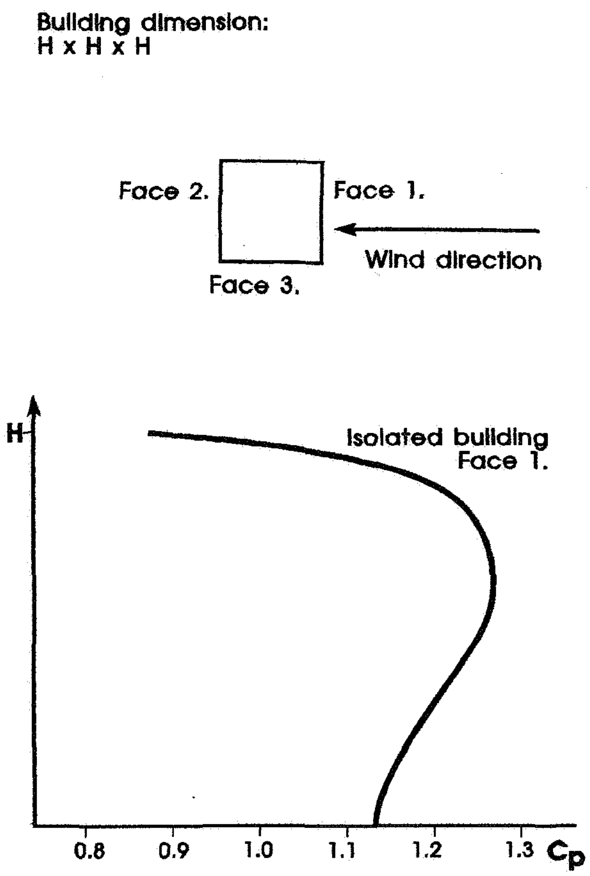
Further applications of this CFD technique might be to study the transport of contaminants and also to see how temperature differences affect the air infiltration of a building.



Appendix 1, Flow Patterns (two Dimensions)



Appendix 2, Flow Patterns (three Dimensions)



Appendix 3, Pressure Coefficients

References

1 Grot R A, Fang J B, "A Numerical Method for Calculating Indoor Airflows Using a Turbulence Model." U.S. National Institute of Standards and Technology, Jan 1990.

2 Bowen J J, "A wind tunnel investigation using simple building models to obtain mean surface wind pressure coefficients for air infiltration estimates." National Research Council, Canada, 1976.

BOWEN

Model O
Side

Height	A	B	C	D
0.71	-0.39	-0.12	-0.30	0.95H
0.88	-0.29	-0.12	-0.35	0.85
0.61	-0.21	-0.14	-0.34	0.75
0.42	-0.14	-0.14	-0.31	0.65
0.32	-0.22	-0.14	-0.38	0.55
0.24	-0.27	-0.26	-0.30	0.45
0.29	-0.24	-0.23	-0.21	0.25
0.59	-0.24	-0.23	-0.16	0.05

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Do Filters Pollute or Clean the Air?

Dr. Philomena M. Bluyssen, TNO-Building and Construction Research, Department of Indoor Environment, Building Physics and Systems, Delft, The Netherlands

Abstract

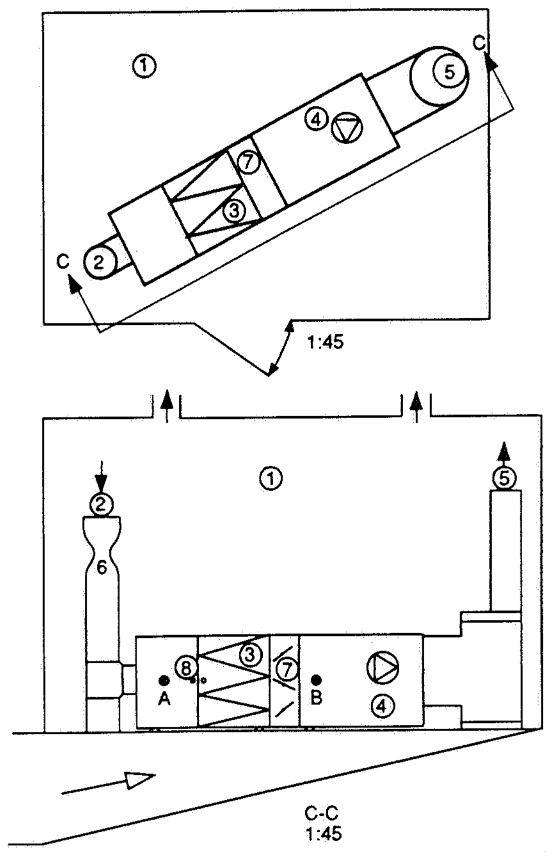
The contribution of new and used fine bagfilters (EU7) to indoor air pollution has been determined in a laboratory study by a trained panel as well as by TVOC measurements. The used filters were all taken out of air handling systems which run with full outdoor air supply. The time that a used filter had been situated in the ventilation system until it was taken out and was studied in the laboratory varied from two to ten months. The new filters did not pollute significantly in comparison to the used filters. However, all used filters polluted the air instead of cleaning it.

Introduction

Ventilation systems may in some cases be a major contributor to indoor air pollution. In several studies it was shown that work-related symptoms of employees occur more frequently in buildings with a ventilation system than in buildings without a ventilation system (1,2,3,4,5). Using a naive panel of judges, Fanger et.al. (6) showed that ventilation systems frequently contribute to the pollution of indoor air. Pejtersen et.al. (7) went a step further and located where pollution sources may be situated in a ventilation system by using a trained panel (8). The pollution in the investigated ventilation systems came mainly from the filters. The filters mostly found in ventilation systems of offices in Denmark are fine bag filters which follow the EUROVENT Standard EU7 (9). Therefore, fine bag filters following EU7 were studied. How much does a filter, new or used, contribute to indoor air pollution was the principal question to be answered. A trained panel was used as well as TVOC (Total Volatile Organic Compounds) measurements, to investigate the contribution of several filters to indoor air pollution.

Procedure

Test system
To study different filters in the laboratory, a test system was built in one of the twin climate chambers of the Laboratory of Heating and Air Conditioning (Figure 1) (10). This test system comprised a filterbox and a fan. The motor of the fan was speed controlled, so the airflow through the system could be varied. The airflow through the test sytem was measured by a thermo-anemometer at the inlet of the test system (point 6 in Figure 1). The temperature of the air in the climate chamber was kept at 22°C. The outdoor air supply to the climate chamber could be varied from 5 to 426 l/s and the humidity was measured by a $LiCl,H_2O$-sensor.



1: climate chamber
2: inkt test system
3: filter
4: fan
5: outlet test system
6: measuring point airflow
7: damper (not in use)
8: pressure drop measuring points
A: connection point diffuser A
B: connection point diffuser B

Figure 1: Set-up of test system in climate chamber

Test method
A panel of ten subjects was trained for ten days during three weeks, 30 minutes per day, to evaluate perceived air quality in decipol (8).

One decipol is defined as the perceived air quality in an unpolluted space, ventilated with 10 l/s of "fresh" air, where a standard person (1 olf) is present. One olf is defined as the bioeffluents of a standard person (11). A standard person is a thermal comfortable average adult who perfoms sedentary office-work and has a hygienic standard of 0.7 baths per day. Any other pollution source can be quantified by the number of standard persons (olfs) that are required to cause the same dissatisfaction as the actual pollution source. A perceived air quality of 1.4 decipol in a space means that 20% of the visitors to that space will find the air quality unacceptable (11). Air samples were exhausted before and after the filter tested (Figure 2), by small fans via polyethylene tubes. The airflow coming out of the diffusers (A and B in Figure 2), located at the end of each tube, was regulated by speed control of the small fans at approximately 1 l/s. This was determined by measuring the air velocity in the top of the diffusers by a Lambrecht anemometer. The outdoor air supply to the climate chamber was kept at 426 l/s and the airflow through the filter was set at a level between 10 and 230 l/s. At least for one hour the airflow was kept constant before the trained panel evaluated the quality of the air coming out of the two diffusers in decipol. The TVOC levels of the air coming out of the diffusers were measured for half an hour in each diffuser by a gas monitor (B&K type 1302). The TVOC level was an equivalent of methane, i.e. the gasmonitor was calibrated with methane and registered all other VOC which were in the same measuring range (detection limit: 0.1 ppm).

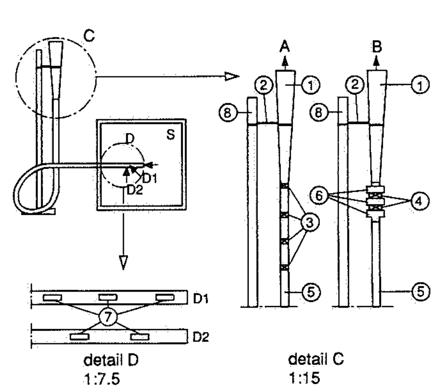


Figure 2: Test method

By calculating the difference in decipol between the air leaving the two diffusers and measuring the airflow through the filter, the source strength in olf of the studied filter can be calculated by the following equation:

$$S_{fs} = 0.1x(C_b - C_a)xQ_t \text{ (olf)} \qquad (1)$$

where:

S_{fs} = subjectively determined source strength of the filter (olf)

C_a,C_b = perceived air quality before (A) and after (B) filter (decipol)

Q_t = airflow through filter (l/s)

The source strength in μg/s TVOC of the studied filter can be calculated in a similar way:

$$S_{fc} = (C'_b - C'_a)xQ_tx663x10^{-6} \ (\mu/s) \qquad (2)$$

where:

S_{fc} = chemically determined source strength of filter (μg/s)

C'_a,C'_b = concentration of TVOC before and after filter (ppb)

$663x10^{-6}$ = conversion factor from ppb to μg/l, based on the molecular weigth of methane

Six glassfiber filters (F85, EU7), two new and four used, were selected for testing. Information on the selected filters is presented in Table 1 and Figure 3. The used filters were taken out of systems which were situated in buildings located on the country side of Copenhagen, and which ran approximately 12 hours a day with full outdoor air supply.

The pressure drop over each filter was measured by a debro water-pressure-based instrument for two different airflows. Since the openings between the fibers of the filters are small, a laminar flow through the filter occurs (Reynolds number is small) and the pressure drop over the filter is therefore linearly related to the airflow. The mass of each filter was measured before and after the tests by an electronic balance. The mean mass and the pressure drops are given in Table 1.

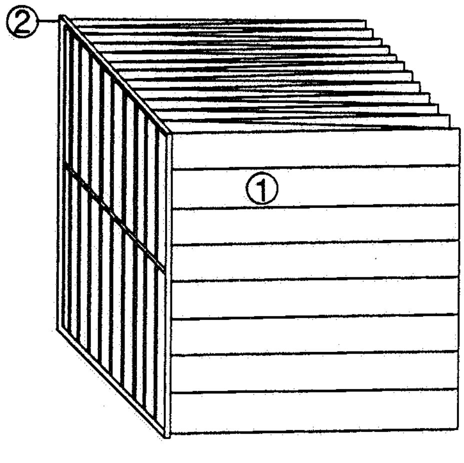


Figure 3: Tested glassfiber filter (F85, EU7)

filter no.[1]	age[2] (months)	measured pressure drop over filter in test system (Pa) with airflow (l/s)		calculated pressure drop[3] (Pa/(l/s))	mass (gram/m²)[4]
$N1_{a,b}$	.	4.1 (94)	8.1 (184)	0.044	6887
N2	.	7.2 (86)	12.2 (184)	0.075	6861
U1	2	9.1 (102)	18.2 (221)	0.088	6974
U2	6	49.5 (102)	79 (211)	0.433	8774
U3	2	5.0 (94)	8.2 (193)	0.047	5368
U4	10	6.0 (94)	9.1 (184)	0.060	8713

Table 1: Information on the tested new and used filters

1: N stands for new filter; U stands for used filter
2: age stands for the time that a used filter has been situated in the ventilation system until it was taken out and was studied in the laboratory
3: assuming a linear relation between pressure drop and arflow
4: this unit is expressed per m^2 filter cross area; cross section of the filters is 0.56x0.56 m^2

Measurements

New filters
Three glassfiber filters (F85, EU7) (Figure 3), manufactured at different times were tested. The filters (filters N1a,b and N2 in Table 1) were tested three and a half months and two months after manufacture, respectively. During those periods the filters were stored in cardboard boxes. To test whether the source strength of a new filter decreases with time, filter N1a was tested four times: after no air had passed the filter, and after a total of 65,000 m^3, 138,000 m^3 and 274,000 m^3 had passed the filter. Filter N2 was tested three times: after no air had passed, and after a total of 2,300 m^3 and 100,000 m^3 had passed. During the measurement the airflow through the filter was set at a value between 11 and 45 l/s. After each measurement the airflow through the filter was set at approximately 200 l/s until a certain amount of air had passed the filter. The airflow was then decreased, and 24 hours later the measurements were made (stationary conditions were reached).

The test conditions are shown in Table 2. The source strength of each filter for each test condition was calculated by using Equations 1 and 2. Figure 4 shows the relation between the total passed air and the source strength of the filter in olf. To study whether the source strength of a new filter depends on the airflow through the filter, a third filter, filter N1b, was tested for three different airflows through the filter, after a total of 120,000 m^3 of air had passed that filter (Table 2, Figure 5).

filter no.	total air volume passed filter (m^3)	airflow through filter (l/s)	perceived air quality[1] (decipol)		measured TVOC[1] (ppb)		source strength (olf)[2]	(µg/s)[3]
			A	B	A	B		
$N1_a$	0	13.9	1.8	8.6	.	.	9.5	.
	65000	15.4	2.1	7.6	.	.	8.5	.
	138000	11.4	2.2	8.8	.	.	7.5	.
	274000	18.5	2.7	6.8	.	.	7.6	.
$N1_b$	120000	15.4	1.7	6.2	.	.	6.9	.
	120000	28.2	2.4	4.7	.	.	6.5	.
	120000	64.6	1.9	3.0	.	.	7.1	.
N2	0	18.5	2.1	4.0	148	374	3.5	2.8
	2300	44.4	1.8	2.7	148	274	4.0	3.7
	100000	15.2	2.9	5.7	285	662	4.3	3.8

Table 2: Test conditions and measurements of new filters

1: A and B refer to the locations in Figure 1
2: the source strength of the filter in olf was calculated by Equation 1
3: the source strength of the filter in g/s was calculated by Equation 2

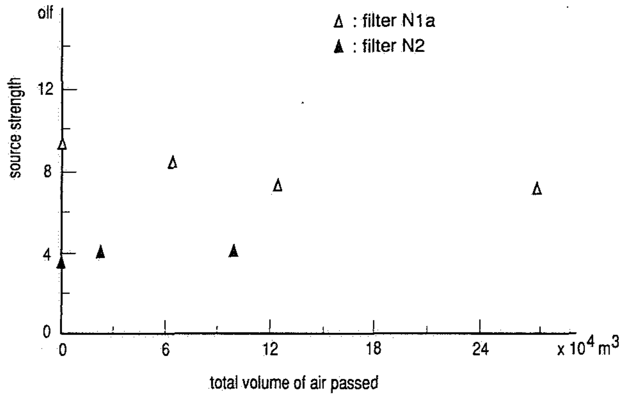


Figure 4: Source strength (in olf) of new filter versus total volume of air passed

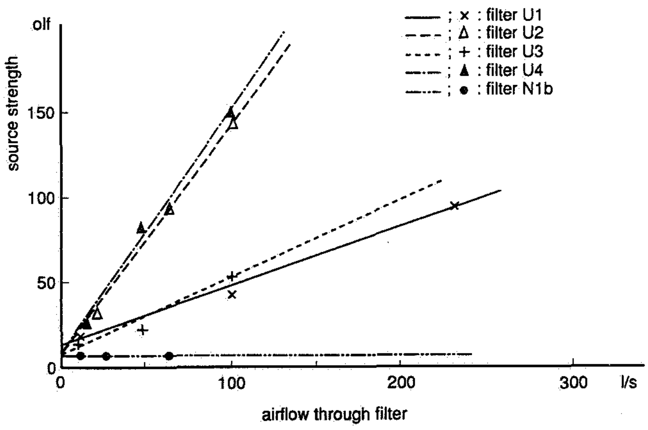


Figure 5: Source strength (in olf) of used and new filters versus airflow through filter tested

Used filters
A series of tests with four used filters were performed. Information on the four selected filters (U1, U2, U3 and U4) is given in Table 1. The used filters were all made of glassfiber and were of the type F85, EU7. To study whether the source strength of a used filter depends on the air flow through that filter, each used filter was evaluated with three different air flows. The test procedure was identical to the test procedure of the new filters. The test conditions and measurement results are presented in Table 3.

filter no.	airflow through filter (l/s)	perceived air quality[1] (decipol)		measured TVOC[1] (ppb)		relative humidity (%)	source strength (olf)[2]	(µg/s)[3]
		A	B	A	B			
U1	13.6	3.4	16.0	562	603	31	17.1	0.4
	101	2.9	7.0	394	478	31	41.1	5.6
	228	4.3	8.5	415	539	37	95.8	18.6
U2	22.2	4.7	19.5	384	537	27	32.9	2.3
	63.9	2.9	17.6	267	347	28	93.9	3.4
	101	4.4	18.6	471	502	28	143	1.1
U3	13.6	2.6	13.6	554	561	36	15.0	0.05
	48.8	2.5	6.7	481	590	33	20.6	3.5
	101	1.9	7.6	537	718	35	57.6	12.1
U4	18.6	2.6	17.7	385	802	21	28.0	5.2
	48.8	2.7	19.6	479	636	32	82.5	5.0
	101	3.7	18.4	553	553	31	149	0.0

Table 3: Test conditions and measurements of the used filters

1: A and B refer to the locations in Figure 1
2: the source strength of the filter in olf was calculated by Equation 1
3: the source strength of the filter in µg/s was calculated by Equation 2

The source strength of the filters were determined by Equations 1 and 2. Figures 5 and 6 present the relation between the air flow through the filter tested and the source strength of the filters respectively in olf and µg/s.

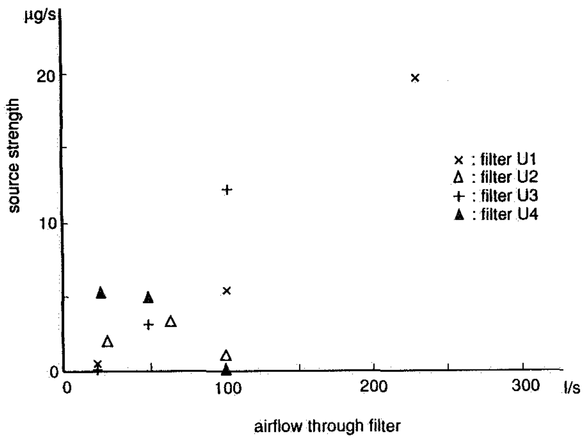


Figure 6: Source strength (in µg/s) of used filters versus airflow through filter tested

Discussion

The perceived air quality in decipol and the TVOC-concentration in ppb were applied to determine the pollution strength of two new and four used fine bag filters (EU7). After unpacking and placing the new filters in the test system, the source strength of filter N2 in olf increased negligibly with the increasing total passed air, while the source strength of filter N1a decreased circa 20% after approximately 120,000 m^3 air had passed (Figure 4). The results showed that the source strength of a new filter did not depend on the airflow through the filter. For three different airflows through filter N1b (15, 28 and 65 l/s) the source strength was around 7 olf (Table 2, Figure 5).

The source strength of used filters in olf seemed however strongly related to the airflow through the filter (Figure 5). The source strength in olf increases with increasing airflow. This relation seems to be linear. The chemical measurements show a similar trend in two cases (filters U1 and U3 in Figure 6). When the source strength of a used filter depends linearly on the airflow through the filter, the source strength (in olf) of a used filter with an airflow of 500 l/s (normal value for this type of filter) is rather high. Until now the source strengths were determined with airflows below 500 l/s. Further studies on this dependency are therefore required.

Bluyssen (12) describes, using the so-called age-model, that after the filter emissions have passed the HVAC-system and reach the space (office, a part of these emissions have died out, in other words have reacted with oxygen or with each other. Another possible explanation of this reduction is the adsorption or absorption of the emitted compounds by the HVAC-components or building parts that are passed on the way.

The total volatile organic compounds (TVOC) were measured at the same locations as the panel members sniffed to evaluate the perceived air quality (A, B and R in Figure 1), to determine if a relation between TVOC and perceived air quality exists in this study. Figure 7 shows that a direct relation does not exist. In this figure all measured values of TVOC (ppb) are related to the perceived air quality (decipol) values. In Figure 8 the source strength of the used filters in olf is related to the source strength of the used filters in µg/s per used filter. For two used filters, U1 and U3, a linear relation between the source strength in olf and µg/s exists. For the other two it seems that the source strength in µg/s is rather constant, while the source strength in olf increases.

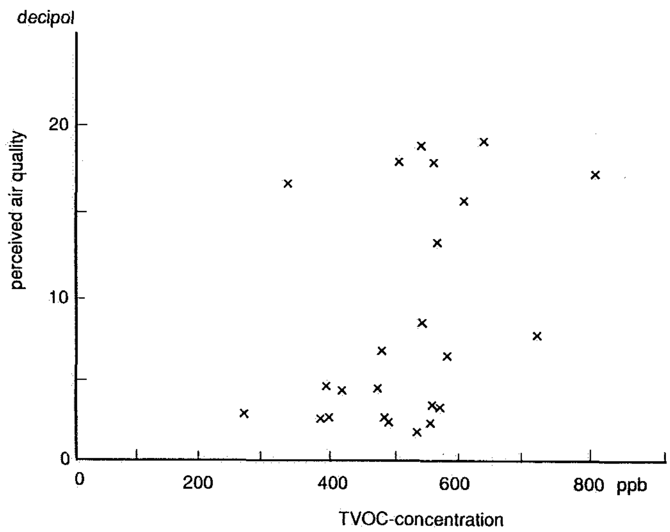


Figure 7: TVOC-concentration (ppb) versus perceived air quality (decipol) for each filter tested

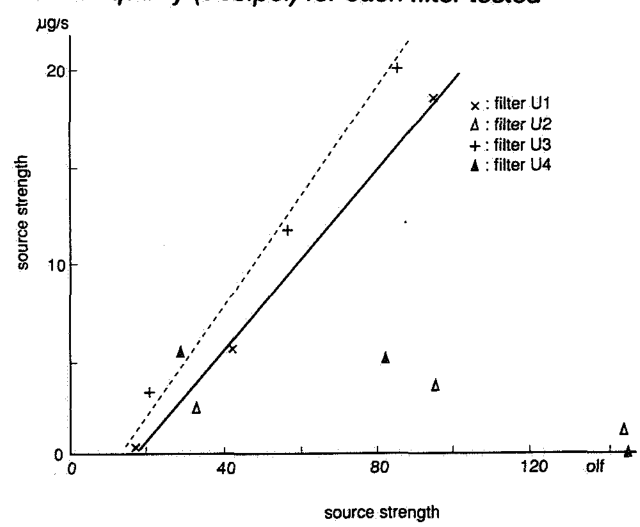


Figure 8: Source strength in olf versus source strength in µg/s for each filter tested

This indicates that the particulate matter in the four filters with different locations may have different characteristics. The human being may perceive some pollutants as annoying, while the concentration of the pollutant is not detectable by the instrument, or the perceived pollutants are not belonging to the TVOC that are measured. For each filter tested the increase of TVOC is however low.

The measured pressure drop over the filters tested, and the measured mass of the filters tested are given in Table 1. Although the filters tested were all of the same type and manufacturer, filter U3 had a mass which did not fit with the others. Its mass was lower than the mass from a new filter, indicating that filter U3 was not the same as the other filters. More data are required to determine a possible relation between source strength and mass or pressure drop of a filter.

The emission rate of a used filter seemed to be dependent on the airflow (or the velocity) of the passing air. Since the airflow of the passing air had no influence on the emission rate of a new filter the airflow (velocity) probably influenced only the emission from the dust in the filter rather than the filter in itself.

The used filters tested can thus contribute to the indoor air pollution. The collected dust seemed to be the reason for this, since new filters contributed negligibly. Valbjørn et.al. (13) sampled dust from filters and found that the perceived air quality and the release of TVOC were negligible. In another study, Martikanen et.al. (14) studied intake ventilation filters for the occurrence of growth of microbes. They concluded that microbes are able to grow on filters even with a temperature of 5 °C and a relative humidity of 75%.

Airborne dust provides nutrients for microbial growth. Many fungi produce odorous metabolic products and irritants. This might be the explanation for the contribution of the used filters (U1, U2, U3 and U4) to indoor pollution. Although the tested filters were studied with a relative humidity between 21 and 37%, they were all taken out of systems where they were exposed to a relative humidity approximately equal to the outdoor relative humidity (80-90%). This implies that fungi may be present in the filters and may have contributed to the pollution. Further studies on among others the influence of water in filters are therefore required. If the relative humidity is the cause of the indoor pollution contribution of a used filter, a solution to prevent this can certainly be found.

Conclusions

A trained panel was used to evaluate the source strength of new and used air filters in a ventilation system.

The pollution contribution of a new filter to the indoor air was found to be small. Used filters may contribute significantly to pollute indoor air.

Further studies are recommended on pollution from filters as function airflow. The impact of relative humidity and fungi growth in filters on pollution source strength should be investigated.

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An Invariant of the Age of Air: Proof and Applications

Michele Mantegna, Department of Energy, Turin Polytechnic, Italy

Abstract

After a short introduction on the "age of air" concept in ventilation, a theorem of conservation for the age of air at the outlet is presented. Restrictions to validity and two applications (to measurements and numerical simulation) are described. A rigorous analytic proof is shown; based on field theory.

1. Introduction

The age of air is a useful parameter to evaluate artificial ventilation effectiveness (see (1), (2) for details about definition and computation). For our purpose, suffice it to say that the age of air in a point is the time taken by air flowing out of the inlet (or inlets) to reach that point. Obviously the age of air at the inlet is zero by definition.

When air flows into a room, it should not contain any pollutant. As it remains inside the room, its age rises. Age increase is a symptom of diminished aptitude to respiration. The phenomenon is not reversible. No natural process can purify air inside a room, though there are devices which actually improve air quality such as ionizers, ozonizers, etc. Only dust content may decrease spontaneously thanks to precipitation if particle size is large enough (for instance metal particles). However the best remedy to huge particle production is the installation of cowls (3), (4).

2. Theorem of Conservation

The above mentioned theorem has been already quoted in a recent paper (5). Here is a complete list of hypotheses.

Consider a room of given size. It may have any shape.

1) It has volume V;

2) it is ventilated artificially;

3) flow is stationary;

4) volume flow is $\dot{V}$;

5) there is a single inlet and a single outlet;

6) air flows throughout the whole room but for a few points at the most.

As a consequence of these hypotheses we list the following two theses:

1) air age at the outlet is independent from position of both the inlet and the outlet, in any position although not coincident;

2) air age at the outlet is $V/\dot{V}$.

Proof of the theorem is described before a thorough discussion of hypotheses in order to emphasise their necessity. Notice that air age at the outlet plays the role of a ventilation invariant thanks to the theorem. The idea of invariant is derived from mathematics and physics where has become more and more important during the 20th century (6).

Reason why the thesis has been divided in two parts will become clear later.

We suggest at least two applications of the invariant. First of all it lends itself to test appliances for air age measurement. Another application is to check the approximation of algorithms to compute air age. We will discuss the latter in depth.

3. Proof

Early studies on air age were mainly experimental. Though the definition of air age is linked to the velocity field, methods are known which do not require detailed knowledge of velocity. For instance in the CVC (Controlled Ventilation Chamber) of the University of Basilicata (Potenza, Italy) one can measure only velocity module. They use contaminants in order to evaluate air age. Contaminants are also referred to by Mats Sandberg in his heuristic proofs (1).

Numerical simulation of artificial ventilation provides velocity field in the chamber as a natural output. It is easier to compute air age (both local and global) from a velocity field instead of simulating contaminants diffusion. That is possible but requires another group of equations (transport equations of contaminants).

It is preferrable to ascertain the theorem of invariance straight from flow dynamics principles (which are correct) instead of resorting to heuristic hypotheses. Analytic proof of the theorem has a further advantage: it shows the independence from Navier-Stokes and turbulence model equations.

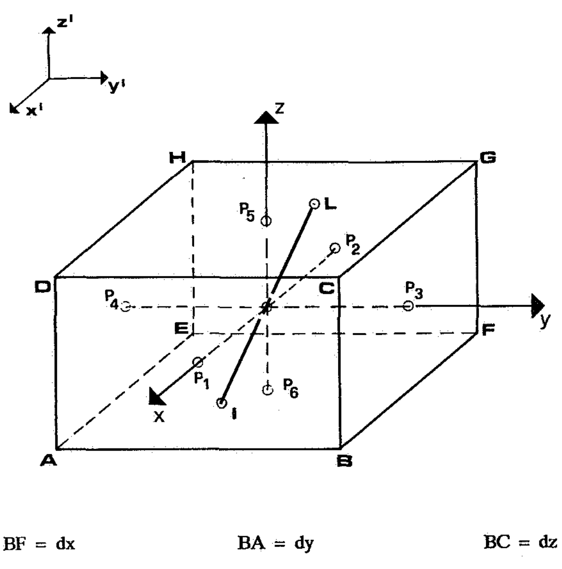
Out of simplicity we have been looking for the smallest set of flow dynamics equations which allows the proof. Since most transport equations are not linear and not easy to manipulate, we shall start from the continuity equation. It has the advantage of linearity.

However the above mentioned theorem cannot derive from the continuity equation only. Continuity equation does not even contain air age. Moreover it is obvious that continuity equation is satisfied by one second old as well as by one hour old air.

Even the air age equation is not enough to prove the theorem because it does not provide any information about the flow. Consider for instance two velocity fields. The former is balanced in mass, that is to say it satisfies the continuity equation. The latter is unbalanced and velocity is zero near the inlet. One can apply the algorithm described in (2) to both fields. In the former case air age gets a finite value at the outlet; in the latter case air age at the outlet is not definite.

This confirms that the equation of transport of air age alone is not enough to prove the invariance theorem. Both equations are required. We shall resort to general methods in field theory (8).

Consider an infinitesimal parallelepiped centred around the point P of coordinates (x,y,z) inside the ventilated room (Fig 1). P_1, P_2, P_3, P_4, P_5, P_6 are face centres. We shall write a transport equation for air age. Nowadays transport equations are commonly used to emphasize analogies between different equations. Flow is supposed to carry properties (density, momentum, enthalpy, turbulent kinetic energy, etc.). We shall extend this principle to air age.



BF = dx BA = dy BC = dz

P_1 = (x + 1/2 dx, y z)
P_2 = (x - 1/2 dx, y z)
P_3 = (x , y + 1/2 dy, z)
P_4 = (x , y - 1/2 dy, z)
P_5 = (x , y , z + 1/2 dz)
P_6 = (x , y , z - 1/2 dz)

Figure 1

The general form of transport equation has four terms: (accumulation rate of the property inside the boundary) + (convective transport of the property through the boundary)

= (diffusive transport of the property through the boundary)
+ (production rate of the property inside the boundary) [1]

Thanks to hyp. 3) first term of left side is zero. Because of the nature of air age, first term of right side is zero too. Diffusive transport relies upon microscopic velocities which exist even when fluid is still. Since air age is due to velocity fields, its diffusive transport is negligible.

We argue that the transport equation for air age contains two terms:

(convective transport through the boundary) = (production rate inside the boundary) [2]

As an example consider the mass conservation equation for a stationary flow, where $\dot{g}$ is the internal mass production rate, ρ is density, $\vec{c}$ is velocity vector and c_x,c_y,c_z are its scalar components along Cartesian axes:

$$\frac{\delta(\rho c_x)}{\delta x} + \frac{\delta(\rho c_y)}{\delta y} + \frac{\delta(\rho c_z)}{\delta z} = \dot{g} \qquad [3]$$

Generally $\dot{g}=0$. Thus mass conservation equation becomes continuity equation:

$$\frac{\delta(\rho c_x)}{\delta x} + \frac{\delta(\rho c_y)}{\delta y} + \frac{\delta(\rho c_z)}{\delta z} = 0 \qquad [4]$$

Let us write the first term of eq. (2) for air age convective transport. ABCD face of the infinitesimal parallepiped is crossed by mass flow

$$[\rho c_x + 1/2 \frac{\delta(\rho c_x)}{\delta x} \delta x] \; dydz \qquad [5]$$

with air age

$$\tau + 1/2 \frac{\delta \tau}{\delta x} dx \qquad [6]$$

Total air age flow exiting ABCD face is:

$$[\rho c_x + 1/2 \frac{\delta(\rho c_x)}{\delta x} dx] \, . \, (\tau + 1/2 \frac{\delta \tau}{\delta x} dx) \; dydz \qquad [7]$$

Mass flow entering EFGH face is:

$$[\rho c_x - 1/2 \frac{\delta(\rho c_x)}{\delta x} dx] \; dydz \qquad [8]$$

with air age:

$$(\tau - 1/2 \frac{\delta \tau}{\delta x} \delta x) \; dydz \qquad [9]$$

Total air age flow entering EFGH face is:

$$[\rho c_x - \tfrac{1}{2}\frac{\delta(\rho c_x)}{\delta x}dx] \,.\, (\tau - \tfrac{1}{2}\frac{\delta \tau}{\delta x}dx)\, dydz \qquad [10]$$

Net air age flow exiting parallelepiped in X-axis direction is:

$$\rho c_x \frac{\delta \tau}{\delta x} dxdydz + \frac{\delta(\rho c_x)}{\delta x} \tau dxdydz \qquad [11]$$

Remembering derivation rule of a product:

$$\frac{\delta(\rho c_x \tau)}{\delta x} dxdydz \qquad [12]$$

The same way one can prove that net air age flow parallel to Y-axis direction is:

$$\frac{\delta(\rho c_y \tau)}{\delta y} dxdydz \qquad [13]$$

and in z-axis direction:

$$\frac{\delta(\rho c_z \tau)}{\delta z} dxdydz \qquad [14]$$

Thus left side of eq. (2) is:

$$\left[\frac{\delta(\rho c_x \tau)}{\delta x} + \frac{\delta(\rho c_y \tau)}{\delta y} + \frac{\delta(\rho c_z \tau)}{\delta z}\right] dxdydz \qquad [15]$$

We shall now compute air age production rate inside the infinitesimal parallepiped. The idea is that air age production is the consequence of time taken to cross the parallepiped. Delay causes air aging.

Consider a velocity vector $\vec{c}$ having the same centre P as the parallepiped:

$$\vec{c} = c_x \vec{I} + c_y \vec{J} + c_z \vec{K} \qquad [16]$$

According to continuity equation (4), air flow described by $\vec{c}$ coincides with flow entering the parallepiped through faces EFGH, AEHD and ABFE:

$$\rho c_x dydz + \rho c_y dxdz + \rho c_z dxdy = [\rho c_x - \tfrac{1}{2}\frac{\delta(\rho c_x)}{\delta} dx]$$
$$dydz + [\rho c_y - \tfrac{1}{2}\frac{\delta(\rho c_y)}{\delta} dy]\, dxdz$$
$$+ [\rho c_z - \tfrac{1}{2}\frac{\delta(\rho c_z)}{\delta} zdz]\, dxdy$$

and exiting the parallepiped through faces ABCD, BFGC and CGHD.

Trajectory crosses parallepiped surface in I and L which lie on faces perpendicular to Z-axis, as an example. With a geometric argument one finds that component of IL along Z-axis is:

$$IL \,.\, \frac{c_x}{||\vec{c}||} = dz \qquad [17]$$

Notice that this length bears a sign.

Component of $\vec{c}$ along Z-axis is:

c_z

This component bears a sign too.

Aging of air through the parallepiped is:

$$\frac{dz}{c_z} \qquad [18]$$

Here are two remarks:

1) division (18) is possible only if velocity component are different from zero. That is why hyp. 6) is required;

2) use of signed variables gives an always-positive result (19) for air age production according to physical interpretation. Non-negative source distinguishes transport equation for air age from other transport equations (for instance momentum, turbulent kinetic energy) having positive or negative sources from point to point.

Total air age production rate inside parallepiped is the product of effective air flow parallel to Z-axis by production term given by (18):

$$\rho c_z dxdy \,.\, \frac{dz}{c_z} = \rho dxdydz \qquad [19]$$

Notice that it is the mass of air inside parallepiped. One can verify that result is independent from choice of axis (X, Y, or Z).

Replacing terms in eq (2) we get:

$$\left[\frac{\delta(\rho c_x \tau)}{\delta x} + \frac{\delta(\rho c_y \tau)}{\delta y} + \frac{\delta(\rho c_z \tau)}{\delta z}\right] dxdydz = \rho dxdydz$$

[20]

Since balance is correct for any infinitesimal parallepiped inside the chamber, we shall integrate eq. (20) over the whole volume V:

$$\iiint_V \left[\frac{\delta(\rho c_x \tau)}{\delta x} + \frac{\delta(\rho c_y \tau)}{\delta y} + \frac{\delta(\rho c_z \tau)}{\delta z}\right] dxdydz = \iiint_V \rho dxdydz$$

[21]

Right of the triple integral sign in the left side we recognize divergence of $\rho\tau\vec{c}$ which is the specific air age flow vector. Applying divergence theorem (also known as Gauss' theorem) (8) to transform triple integrals into double ones we get:

$$\iint_S \rho\tau \left(\vec{c} \,.\, \vec{n}\right) dS = \iiint_V \rho dxdydz \qquad [22]$$

where S is the chamber surface.

$\vec{c} \,.\, \vec{n}$ is the component of air velocity perpendicular to chamber surface, that is to say the component which introduces or extracts air from the chamber. But according to hyp. 2) ventilation is only artificial, then

$\vec{c}\cdot\vec{n}$ is zero everywhere but on the inlet and outlet. Say ρ the average density of air, C effective speed normal to inlet and outlet, A cross section area of inlet and outlet, I average air age at the inlet and outlet:

$$\rho_{out}A_{out}C_{out}\tau_{out}-\rho_{in}A_{in}C_{in}\tau_{in}=\iiint_V \rho dxdydz \qquad [23]$$

Since τ at the inlet is 0:

$$\rho_{out}A_{out}\tau_{out}=\iiint_V \rho dxdydz \qquad [24]$$

Triple integral in the right side of eq. (24) is equal to air mass M contained inside the chamber. Eq. (24) becomes:

$$\dot{M}\tau_{out}=M \qquad [25]$$

Ventilation generally deals with air volumes rather than air masses. Being air density ρ nearly constant for technical purposes, division of both numerator and denominator of eq. (25) by air density gives:

$$\tau_{out}=\frac{M}{\dot{M}} \qquad [26]$$

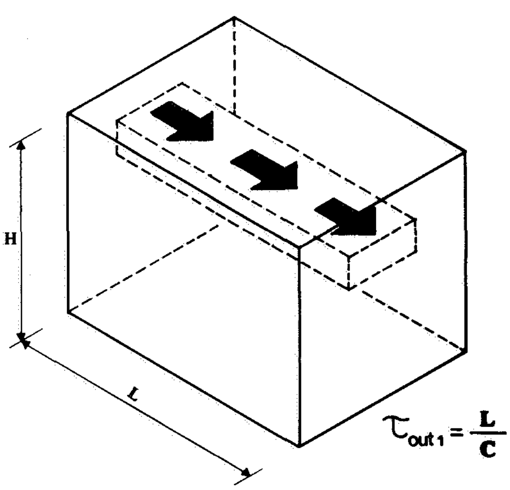
However theorem is not limited to constant density. Eq. (25) remains true even with large differences in density. That is why we have not required density to remain constant.

4. Limitations

Proof of the theorem relies upon hyp. 6). If hyp. 6) is given up, proof fails. An obvious question arises: does this depend upon a fault of the proof? If so, one could find a proof that does not need hyp. 6).

That is impossible. It is easy to see that theorem is false without hyp. 6).

Consider airtight chambers shown in Figure 2. They both satisfy all hypotheses with the exception of 6). We suppose that velocity is zero everywhere but for a finite region where velocity has module C. Thus we have an infinite number of stagnation points, contrary to hyp. 6).



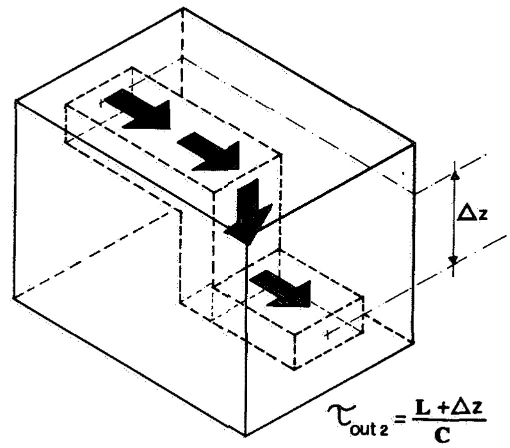


Figure 2

Such velocity fields contradict our idea of flows inside closed rooms and certainly do not satisfy Navier-Stokes equations (because at the interface with still air velocity gradients are not definite). Nonetheless these fields may satisfy continuity equation and do actually satisfy it. Air age is computed easily by dividing trajectory length by velocity which is the same in both chambers. Since trajectory in the latter chamber may be longer than in the former, theorem is false without hyp. 6).

Numerical verification has been performed that velocity fields obtained with conformal mapping (7) exhibit the invariance property though they do not satisfy Navier-Stokes equations.

Velocity field of an actual fluid, which satisfies all transport equations (continuity, Navier-Stokes, Turbulence, etc.) must agree with hyp. 6). Air inflow originates tangential stresses at the interface with still air in the chamber. These stresses propagate motion to the whole mass of air so that only fluid adjacent to walls can remain still. Actually molecular diffusion is active there too.

Velocity inside the chamber is zero only in a finite number of points, just as required by hyp. 6). Points of zero velocity are centres of recirculation vortexes. One can prove (8) that the impossibility of writing eq. (20) in a finite number of stagnation points does not harm the proof. That is because the set of stagnation point has a "zero" measure.

After all hyp. 6) is required only to ensure fireproof validity. Actual velocity fields do always satisfy it automatically.

5. Validation of Results of Numerical Simulation

Analytical solutions hardly ever apply to flow dynamics since no general solution is known for the motion of a real fluid, with the exception of a few cases. Moreover we still lack a complete and correct model of turbulence. That is why one has to resort to numerical methods, which provide velocity only in a finite number of points. Air age should be computed accordingly with algorithms such as (2).

Difficulty arises to evaluate accuracy in air age computation. As for reliability, air age invariant provides a simple criterion. Besides it guarantees that velocity field satisfies continuity equation. Unfortunately this does not guarantee correct solution of Navier-Stokes equations since they are not required by invariance theorem. Verification based on air age invariant is useful all the same thanks to its rapidity.

As for accuracy, invariance theorem offers a comparison to evaluate air age approximation inside the room, about which nothing is known otherwise.

We shall refer here to computer program quoted in (7) with a built-in algorithm for air age computation described in (2). Consider the two-dimensional chamber shown in figure 3 having square cross section of three metres by three. It is ventilated artificially with air at 5 centimetres per second through an inlet 30 centimetres high. Outlet is same size as the inlet. Numerical grid has been taken 10 by 10, 20 by 20, 30 by 30, 40 by 40. Each grid has been tested with three ventilation strategies (7) (fig 3).

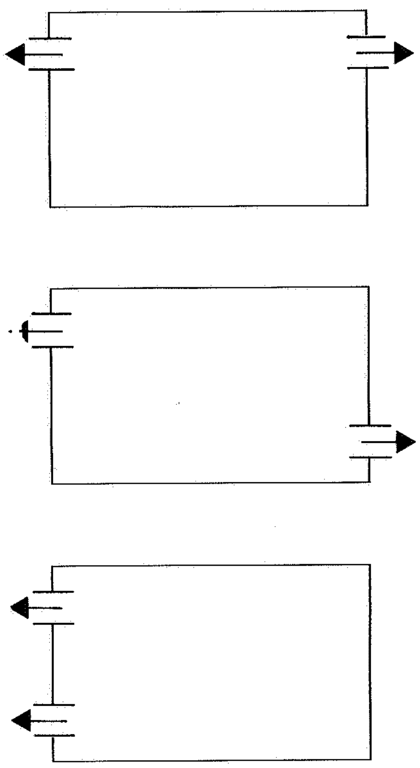


Figure 3

We have found that air age at the outlet is always the same. Differences are within rounding errors beyond the seventh significant figure. Thus thesis 1) of invariance theorem is verified, no matter how fine is the grid.

As for thesis 2) we have found that simulations satisfy it only approximately. However error is about 10 percent even with moderately fine grids of 20 by 20 cells. Error has been found to converge to zero as grid becomes finer, thus verifying thesis 2) too as a limit property.

Table 1 is enclosed which summarises results of simulations. First column contains grid fineness. Since thesis 1) is perfectly verified no matter how fine the grid, second column contains a single value of air age at the outlet. Third column contains ratio between approximate and correct air age at the outlet. Correct value at the outlet is 600 seconds in agreement with invariance theorem.

6. Final Remarks

Table 1 shows that a 40 by 40 cells grid is required by existing algorithm to provide 1 per cent precision on air age. On a powerful personal computer (386 class or higher) computation ends up in an almost negligible time even with such a fine grid. Much more time, even hours and hours, are taken by the computation of velocity field, especially with turbulence. The obvious outcome is the search for more effective algorithms to evaluate air age with quite coarse grids.

GRID	STEP (centimetres)	APPROXIMATE AGE (seconds)	APPROXIMATE CORRECT AGE
10 x 10	30,0	438	0,73
20 x 20	15,0	534	0,89
30 x 30	10,0	575	0,96
40 x 40	7,5	593	0,99

Table 1

Another research aim is the application of invariance theorem to the fine-tuning of measurement appliances, which has not been discussed in depth here.

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Annex 26: Energy Efficient Ventilation of Large Enclosures - Large is Beautiful!

Reprint of ECB Exco Technical Day, Maastricht, November 10, 1992
by Alfred Moser, Operating Agent
A Large Enclosure Does Not Necessarily Mean a Large Waste of Energy

A large enclosed space in a building invites people to enter. If properly designed, it protects from weather, noise, and the terror of road traffic. It may provide comfort, both psychological and physical, by the feel of space and social community as well as by a pleasant atmosphere created by good lighting, colors, indoor plants, and sometimes fragrances and music, and - correct ventilation (Fig 1). Concert halls, sports stadia, atria, shopping centers, and terminal buildings are designed to be BEAUTIFUL. If all systems work fine and indoor climate is good, they are also COMFORTABLE. This is also true for industrial and office buildings where good comfort enhances productivity.

Figure 1: Atrium of the KI building, Tokyo. Offices are open to the atrium. The space is designed to be relaxing and comfortable for the occupants. Fresh fragrances are added to the supply air, a stimulating or relaxing musical environment is programmes according to the daily rhythm, and a fluctuating air flow rate simulates varying breeze at the seashore (ASHRAE Journal, Aug. 92)

Good ventilation need not cost excessive energy. The obvious strategy is to concentrate the air change to the relatively small portion of the space occupied by people. But large temperature differences might blow this idea: the cold air accumulating along the inner surface of a glazing in winter may be rushing down like a waterfall hitting the occupants in the neck! This effect is much stronger if vertical dimensions are large (Fig 2). On the other hand, the laws of nature also come to the rescue, as they guarantee stable layers of air if temperature is stratified. To make ventilation work, the streams and behaviour of buoyant air must be accurately predicted and design guidelines established.

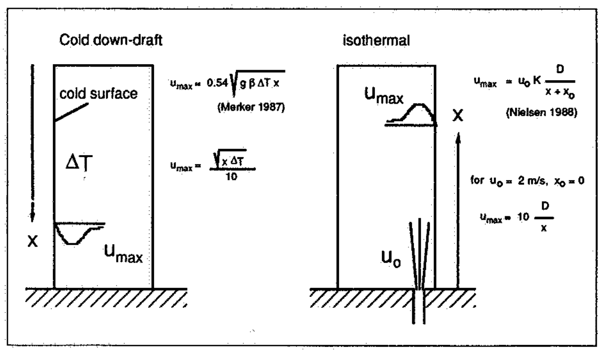

Figure 2: Cold air builds up speed with height; an isothermal jet slows down.

The Goal of the International Project

The objectives of the Annex are shown in Fig 3. The intention of writing new guidelines is ambitious. It is understood that these guidelines are based on and restricted to the ventilation systems, - with their advantages and problems, - of case-study buildings investigated by Annex 26 (see also project profile on the last page of this article).

Objectives

To develop methods* to minimize energy c
provision of:
- good indoor air quality and comfort,
- save removal of contaminants,
- distribution of fresh air

within large enclosures.

* **Analysis- Methods** — Modelling and Measurement

Design- Methods — Guidelines based on Annex-26 studies

Figure 3: Analysis of air motion and interpretion of ventilation problems of demonstration buildings are aimed at improved design guidelines.

The ANALYSIS of an air flow pattern includes field measurements, laboratory experiments, and computer models. Simple computer models, or ENGINEERING METHODS, may be applied by the designer himself. Complex numerical techniques and computational fluid dynamics to calculate the details of the entire flow field require specialized skills and adequate computer resources. Such FIELD MODELS have been tested at smaller rooms by the completed IEA Annex 20. Their performance and reliability have been improved considerably during the last few years, and they become an important tool to assist the design of large-space ventilation (Fig 4).

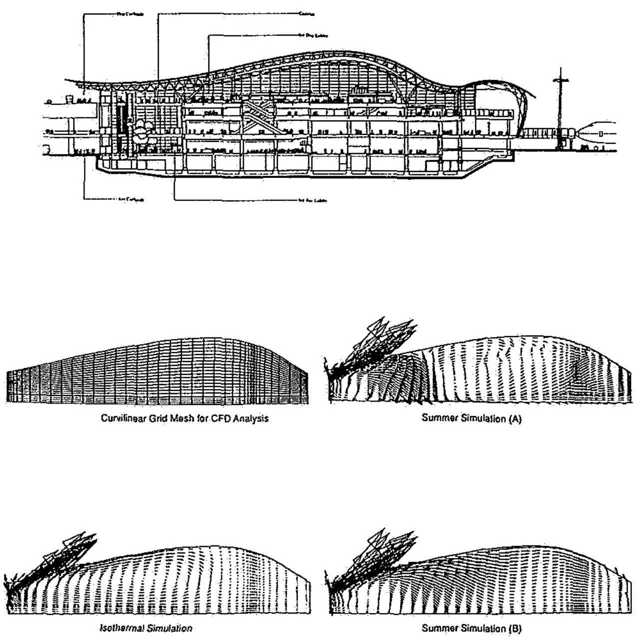

Figure 4: Application of computational fluid dynamics to the ventilation design of Kansai International Airport Passenger Terminal, Osaka, Japan (Proceedings ISRACVE, July 1992, Tokyo).

Design Guidelines are Based on Careful Analysis

At the "kick off" meeting of September 1992 the "space age" specialists decided on an approach that centres around the product of the Annex, - REPORTS with these working titles:

- Analysis tools to measure and model air motion, occupant comfort, and indoor air quality in large enclosures. This report is directed at researchers and consultants.
- Guidelines for the design of energy efficient ventilation of large enclosures based on Annex 26 case studies. This report is directed at designers and HVAC engineers.
- Large enclosure case studies of IEA Annex 26. This summary report describes the investigated building, their ventilation-related characteristics, along with specific desirable features or problems. It is directed at designers, HVAC engineers, and consultants.

To find simple and realistic strategies to design and operate energy efficient ventilation in rooms of extreme dimensions, actual buildings must be investigated. For this purpose, demonstration and case-study buildings within the participating countries are selected. Air flow, indoor air quality, and comfort problems are analysed by measurement and by simulation. This approach calls for an organisation of the work as shown in Fig 5.

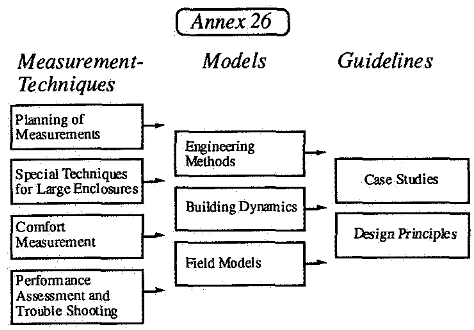

Figure 5: Work plan: Measurements and computer models support the analysis of case studies on existing buildings. A selected team of international experts will review the findings and formulated design principles for ventilation of large enclosures.

New Design Principles are Tested on Case-Study Buildings

A long list of possible case-study buildings has been compiled at the "kick off" meeting. The buildings contain large enclosed air spaces of the following five types:

I High level intermittent human occupancy (e.g. theatres)

II Dominated by vertical geometry (e.g. office with light well)

III Dominated by horizontal geometry (e.g. shopping mall)

IV Low level of human occupancy, but with non-human pollutant sources (e.g. industrial buildings)

V Miscellaneous (e.g. cathedrals, exhibition halls)

On some buildings, problems have been reported regarding energy consumption, thermal comfort, or unsatisfactory contaminant removal. In other buildings, apparently no problems exist. A simple energy audit is planned for all selected buildings to assess energy efficiency. In some cases, existing data may support this audit. Particular problems will then be investigated by well-planned field measurements.

Six of the proposed demonstration buildings are shown in Figs 6-11. Many more buildings are under discussion, and it will be up to the contributing countries whether any of these objects are actually offered for case studies. The final selection among the proposed buildings is made at an expert meeting in agreement with the contributing country. Selection criteria are listed in Fig 12.

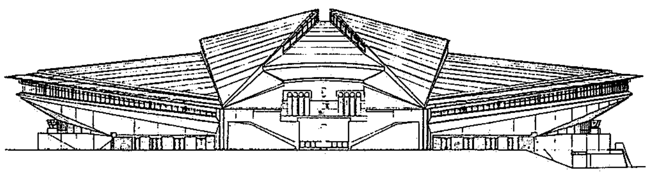

Figure 6: Tokyo Metropolitan Gymnasium, Japan, 1990. A 100 m diameter sports stadium (Type 1)

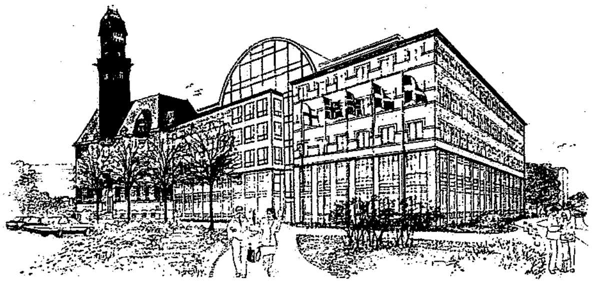

Figure 7: Scandinavian Center, Malmo, Sweden, 1992. Six-storey office building with 22.5 m atrium (Type II)

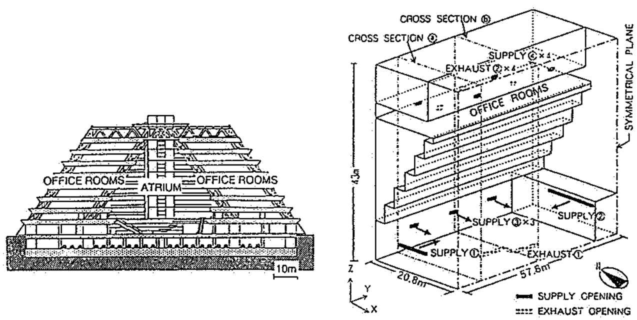

Figure 8: Matsushita Electric Industrial Communications Systems Center, Tokyo, Japan, 1992. Atrium of 43 m height in the centre of office building (Type II)

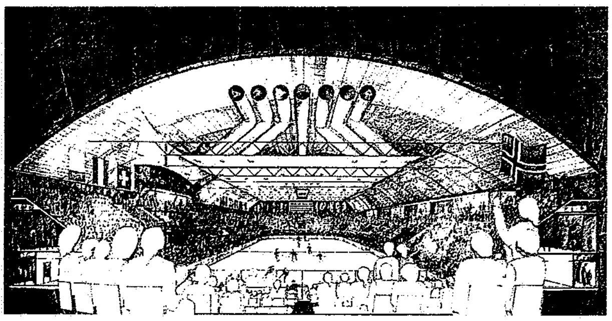

Figure 9: Gjovik Olympic Mountain Hall, Norway, 1993. The world's first underground Olympic stadium (Type I). The rock cavern of 60 m span and 90 m length will seat more than 5000 spectators.

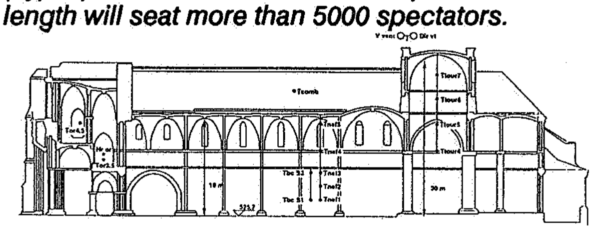

Figure 10: Cathedral of Lausanne, Switzerland. Problems with this 30 m high building include protection of historic wall paintings (fresco) and art pieces from modern pollutants and condensation (Type V)

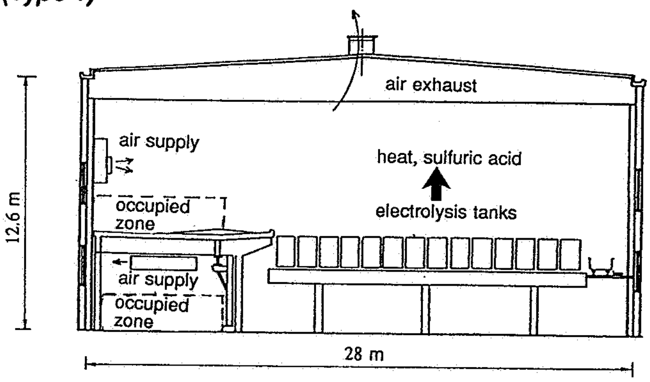

Figure 11: Outokumpu Zinc Hall, Kokkola, Finland, 1969. Industrial building (Type IV) with sources of heat and harmful contaminants at the electrolysis tanks.

Selection Criteria for Demonstration and Case-study Buildings

(1) Work on participant's buildings must be within national budget and available staff.

(2) Access to building by the participant for the purpose of investigations must be guaranteed.

(3) All building types listed in Project Definition should be represented within Annex 26.

(4) Building and its ventilation problems should attract collaboration of other groups in same or different participating country.

(5) Building should incorporate an unconventional feature.

Note: Providing a building is *not* a condition for Annex participation.

Figure 12: The buildings for Annex 26 case studies must meet a number of clearly defined conditions.

Conclusions

A key conclusion of the preparation phase is that the design guidelines are the main product of the project. Two conditions must be met before realistic design principles can be formulated:

Work must be based on real problems in ACTUAL buildings (case studies).

The field experience and know-how must be collected on a basis as wide as possible as each large enclosure is a special case. International cooperation is the right approach.

Project Profile

Title: "Energy Efficient Ventilation of Large Enclosures", Annex 26, a task-sharing Annex to the IEA Implementing Agreement on Energy Conservation in Buildings and Community Systems.

Objective: To develop methods to minimize energy consumption of large enclosures in the provision of

- good indoor air quality and comfort
- the safe removal of airborne contaminants
- the satisfactory distribution of fresh air

Products: Reports on energy efficient ventilation of large enclosures:

Measurement and modelling techniques for analysis

- guidelines for the designer
- case studies of IEA Annex 26

Case Studies: Demonstration and case study buildings serve to demonstrate measurement and modelling methods and to study problems.

Organisation: Two tasks:
1. Measurement techniques and case studies
2. Models

Time Schedule: 1-year preparation phase: April 1, 1992 - March 31, 1993

3-year working phase: April 1, 1993 - March 31, 1996

Operating Agent: The Swiss Federal Office of Energy (BEW). Contractor: Energy Systems Lab, Swiss Fed. Inst. of Technology, ETH, Zurich.

Meetings: 2nd Expert Meeting: Aachen, Germany, March 30 - Apr 2, 1993
3rd Expert Meeting: Poitiers, France, October 5-8, 1993

Information is available from your national Executive Committee representative or from the Operating Agent:
Tel: +41 1 256 36 41, Fax: +41 1 261 42 51, Telex: 817 379 ehhg ch, Electronic mail address: alfred.moser:iet.ethz.ch
Dr Alfred Moser, Energietechnik, ETH-Zentrum, ML, CH-8092 Zurich, Switzerland

International Energy Agency
Air Infiltration and Ventilation Centre

14th AIVC Conference Energy Impact of Ventilation and Air Infiltration

Copenhagen, Denmark
Tuesday 21st to Friday 24th September, 1993

Preliminary Notice

Ventilation contributes significantly to energy demand and also plays a key role in maintaining optimum indoor air quality. At its most basic level, ventilation is needed to satisfy the metabolic requirements of occupants but is further used to dilute and disperse pollutants generated in buildings. For energy efficiency it is essential to avoid excessive or unnecessary air exchange without disturbing the indoor environment. In addition, air infiltration or the uncontrolled entry of air further contributes to energy demand and can destroy the benefit of good ventilation design. Thus building airtightness is an essential ingredient for achieving energy efficient ventilation. Ventilation for cooling purposes is also an important consideration for which the criteria and energy performance need further investigation.

The purpose of the AIVC's 14th Annual Conference is to address the energy issues associated with ventilation and air infiltration.

Topics include:

- Assessing the energy impact of ventilation and air infiltration (including the results of national studies and evaluating energy impact according to building type).
- Methods to assess energy impact (i.e. measurement and computational techniques).
- Energy reduction through heat recovery.
- Demand controlled ventilation.
- System energy demand.
- The identification of ventilation needs.
- Alternative indoor air quality control measures (e.g. pollutant elimination, dehumidifiers, filtration, etc.).
- The influence of air infiltration on ventilation performance.
- Ventilation to meet cooling needs.
- Dealing with outside pollutants.
- Case studies and examples.

Further details from:

Mrs Rhona Vickers
Air Infiltration and Ventilation Centre
University of Warwick Science Park
Sovereign Court
Sir William Lyons Road
COVENTRY CV4 7EZ
Great Britain

Tel: +44(0)203 692050
Fax: +44(0)203 416306

Ventilation in Belgium: The Situation in 1992

by Peter Wouters, Belgian Building Research Institute

Belgium
Surface: 30.500 km^2
Inhabitants: 9.500.000
Density: 3 inh./km^2**
3 regions
Average daily maximum July: 22 °C
Average daily minimum January: 1°C

The climate in Belgium can be described as rather moderate: in wintertime, temperatures are in general a few degrees above zero, snow occurs but in general only a few days a year. In summer time, average maximum temperatures are around 23°C, therefore active cooling is not necessary for domestic applications.

Attention for ventilation in Belgium is rather new. Only since the beginning of the eighties and especially during the last five years, research concerning building airtightness, assessment in existing buildings of the air quality and the performances of the ventilation system was done. At the beginning, severe mould and condensation problems found in many dwellings on the one hand and attention for energy conservation on the other hand were the driving forces to start up research in this field.

Until now, nearly all individual dwellings and even a large part of apartment buildings use only of openable windows and doors and often a cooker hood in the kitchen. Fortunately, and partly as a result of the research activities, a national standard (NBN D50-001) concerning ventilation provisions for dwellings was adopted in 1992. This standard requires air supply provisions in living room, bedrooms, etc., and air extraction in kitchen, toilet and bathroom. Moreover, transfer openings around inner doors and/or walls are required. This ventilation can be realised just as well by natural ventilation provisions as by mechanical ventilation. In the case of natural ventilation provisions, they must give the required air flow rates at a pressure difference of 2 Pa.

In Belgium, the application of the rules given in a standard is not mandatory. However, it is considered as a state-of-the-art document and therefore can have a large impact on the building tradition. The Flemish government has voted a regional decree allowing to have regulations concerning requirements on thermal insulation and ventilation for all buildings in the Flemish region. Until now, such regulation is only adopted for the thermal insulation in dwellings. It is expected/hoped that the Belgian standard NBN D50-001 will become mandatory in the future.

At the national level, there exists since 1991 a regulation requiring mechanical ventilation provisions in all pubs, restaurants and hotels where smoking is allowed. In total there are some 60,000 establishments where such provisions should be installed. Until now, the majority are not yet in line with the requirements.

The airtightness of some 150-200 buildings has been measured during the last ten years. Both for dwellings and for schools, the n50-values are in the range of 1 to 40 h^{-1} with averages probably in the range of 6 to 10 h^{-1}. It is important to stress that the very high values are also found in recently constructed buildings.

Is there an improvement in airtightness? In Belgium, the large majority of the dwellings are individual dwellings (75% of the building stock). A systematic study of some thirty recently constructed detached dwellings is at the moment being conducted. The first results seem to confirm our impression that, although the airtightness of windows and doors has become in general very good, the n50-values are not as low as could be thought: values between 4 and 12 h^{-1}. Major reasons for this rather high values are:

- the fact that nearly all detached houses are individual projects, two or more identical detached dwellings are in Belgium very rare;
- in a large part of the cases, the client of the building contractor is the family who will live in the dwelling, not a housing estate company. Mainly for financial reasons, often a part of the finishing has to be done by the owner (insulation of inclined roofs, wall and ceiling finishing, etc.). This work is often very important for the airtightness;
- the technical competence concerning ventilation and airtightness is still very limited. In nearly all cases, the architect is in charge for all technical aspects, assistance of a specialized engineer does not exist.

It is very important to stress that, although the overall building airtightness of many new detached Belgian dwellings might be rather poor, it is certain that the airtightness of the envelope in individual rooms (living room, bedrooms, etc) is in most cases very good. Therefore, one cannot at all count on the background leakage for the room ventilation and appropriate ventilation provisions in these rooms is essential.

Finally, some information on the type of heating systems used can be given. District heating is hardly used. For dwellings, older dwellings are mostly heated by gas or oil appliances, often a central heating system. In newer dwellings, electrical heating systems are frequently used. From a ventilation point of view, the fact that most of the combustion appliances, even recently installed systems, are using room air for the combustion process is important. In the Netherlands, the situation differs significantly since nearly all recently installed gas appliances are using outside air for the combustion process and are completely airtight towards the occupied space.

Forthcoming Conferences

EEB International Symposium
Energy Efficient Buildings
Design, performance and operation under the auspices of the CIB Working Group W67: energy conservation in the building environment
DATE: March 9-11, 1993
VENUE: University of Stuttgart, Germany
CONTACT: U Fadel, Nobelstr. 12, 7000 Stuttgart 80, Germany
Tel: +49 711 970 3360
Fax: +49 711 970 3395

Indoor Air '93 The 6th International Conference on Indoor Air Quality and Climate
DATE: July 4-8, 1993
VENUE: Helsinki, Finland
CONTACT: Indoor Air '93, Prof Olli Seppanen, Helsinki University of Technology, SF02150 Espoo, Finland
Tel: 358 0 451 3600
Fax: 358 0 451 3611
THEMES: Human health, comfort and performance; indoor air quality and climate; control technology; policy, regulatory and legal issues

Heat Pump for Energy Efficiency and Environmental Progress
The 4th International Energy Agency Heat Pump Conference
26-29 April 1993
VENUE: Maastricht, The Netherlands
CONTACT: 4th IEA Heat Pump Conference, Conference Secretariat, Van Namen & Westerlaken, Congress Organization Services, PO Box 1558, 6501 BN Nijmegen, The Netherlands
Tel: +31 80 234471
Fax: +31 80 601159

International Conference
Thermal Comfort: Past, Present and Future
9-10 June 1993
VENUE: UK Building Research Establishment, Garston, Watford, UK
THEMES: The derivation of thermal comfort equations, context effects (i.e. field versus climate chamber studies), group effects (e.g. age, sex, culture and physiology), climate and adaptation, economics, energy conservation, health (e.g. SBS), psychological factors (e.g. attitudes and behaviour), applying the comfort equation and designing comfortable buildings, the adequacy of current guidelines, the heating and ventilation link, the current situation and way forward.
CONTACT: Mr Nigel Oseland, Conference Secretary, Building Research Establishment, Garston, Watford, WD2 7JR, UK
Tel: 0923 664164

International Building Performance Simulation Association. Third International Conference Building Simulation '93
16-18 August 1993
VENUE: Adelaide, Australia
CONTACT: Building Simulation '93 Secretariat, 264 Halifax Street, Adelaide, Australia 5000, PO Box 44, Rundle Mall, SA 5001
Tel: +618 232 3422
Fax: +618 232 3424
THEMES: Technology Transfer - user needs, integrated design tools
Applications - building performance simulation in practice, intelligent buildings and diagnostic routines
Simulation approaches - computing environments, relational data bases, expert systems, hypermedia
Fundamentals - conceptual models for building performance evaluation
Validation - quality control and assurance

Energy Impact of Ventilation and Air Infiltration 14th Annual AIVC Conference
21-24 September 1993
VENUE: Copenhagen, Denmark
CONTACT: Air Infiltration and Ventilation Centre, University of Warwick Science Park, Sovereign Court, Sir William Lyons Road, Coventry CV4 7EZ, Great Britain
Tel: +44 (0)203 692050
Fax: +44 (0)203 416306

CLIMA 2000 Engineering the Built Environment
November 13, 1993
VENUE: Queen Elizabeth II Conference Centre, London, Great Britain
CONTACT: CLIMA 2000, c/o CIBSE, 222 Balham High Road, London SW12 9BS, Great Britain
Fax: +44 81 675 5449
THEMES: Environmental and resource issues; design criteria relating to human safety, health, comfort and performance; component and system design and application; system and equipment performance, prediction and measurement; system commissioning, design and operation standards; education, training and qualification routes.

BEP '94 Consolidation and Progress
5-8 April 1994
VENUE: York, UK
THEMES: Aims to promote the dual themes of technology integration - to bring together the findings of the many R&D projects now reaching fruition; and design implementation - to explore designer requirements and mechanisms for design process integration.
CONTACT: Mrs Elaine Baker, BEPAC Administration, BRE, Garston, Watford, WD2 7JR, UK
Tel: 0923 664132, Fax: 0923 664780

Roomvent '94
June, 1994
VENUE: Krakow, Poland
CONTACT: Prof Stanislaw Mierzwinski, Institute of Heating, Ventilating and Air Conditioning, Silesian Technical University, U1, Pstrowskiego 5, PL 44100 Gliwice, Poland

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Representatives and Nominated Organisations

Belgium

*P. Wouters, Belgian Building Research Institute (WTCB/CSTC), rue de la Violette,21-23, 1000 Brussels, Belgium. Tel: 02-653-8801 Fax: 02-653-0729

P. Nusgens, Université de Liège, Laboratoire de Physique du Bâtiment, Avenue des Tilleuls 15-D1, B-4000 Liège,Belgium. Tel: +32 41 66 56 74 Fax: +32 41 66 57 00

Canada

*M. Riley, Buildings Group, Energy Efficiency Division, Efficiency and Alternative Energy Branch, Energy, Mines and Resources Canada, Ottawa, Ontario, K1A 0E4 Canada Tel: 613-996-8151 Telex: 0533117 Fax: 613-996-9416

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T. Hamlin, Research Division, Canada Mortgage and Housing Corporation, Montreal Road, National Office, Ottawa, Ontario, Canada K1A 0P7 Tel: 613-748-2984 Telex: 052/3674 Fax: 613 748 6192

Denmark

*O. Jensen, Danish Building Research Institute, P.O. Box 119, DK 2970 Hørsholm, Denmark. Tel: 45-42-865533 Fax: 45-42-867535

P.F. Collet, Technological Institute, Byggeteknik, Post Box 141, Gregersensvej, DK 2639 Tastrup, Denmark. Tel: 42-996611 Telex: 33416 Fax: 45-42-995436

Finland

*M Luoma, Technical Research Centre of Finland, Laboratory of Heating and Ventilation, Lämpömiehenkuja 3, PO Box 206, SF-02151 Espoo 15, Finland. Tel: 358-0-4561 Telex: 122972 Fax: 358-0-4552408

France

*P Hérant, ADEME, Route des Lucioles, 06565 Valbonne Cedex, France Tel: 33 93 95 79 00 Fax: 33 93 65 31 96

Ph. Duchêne-Marullaz, CSTB, 84 Ave. Jean Jaurès, BP 02 Champs sur Marne, 77421 Marne la Vallée, Cedex 2, France Tel: (33-1) 64 68 83 13 Fax: (33-1) 64 68 83 50

Germany

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Italy

*M. Masoero, Dipartimento di Energetica, Politecnico di Torino, C.so Duca delgi Abruzzi 24, 10129 Torino, Italy. Tel: (39-11) 564 4441 Telex: 220646 POLITO Fax: 39 11 564 4499

Netherlands

*W.F. de Gids, TNO Building and Construction Research, Dept of Indoor Environment, Building Physics and Systems, P.O. Box 29, 2600 AA Delft, Netherlands, Tel: +31 15 608608 (Direct: +31 15-608472) Fax: +31 15-608432

New Zealand

*M. Bassett, Building Research Association of New Zealand Inc (BRANZ), Private Bag, Porirua, New Zealand. Tel: 64-4-2357600 Fax: 64 4 2356070

Norway

*J.T. Brunsell, Norwegian Building Research Institute, Forskningsveien 3b, PO Box 123, Blindern, N-0314 Oslo 3, Norway. Tel: +47 2-96-55-00 Fax: +47-2-965542

H.M. Mathisen, SINTEF,Division of App Thermodynamics, N-7034 Trondheim, Norway. Tel: 07-593870 Telex: 056-55620

Sweden

*J. Kronvall, Technergo AB, Ideon, S-223 70 Lund, Sweden, Tel: +46 46 16 88 81, Fax: +46 46 16 88 80

J Lagerström, Swedish Council for Building Research, Sankt Goransgatan 66, S-112 33, Stockholm, Sweden Tel: 08-6177300 Fax: 08-537462

Switzerland

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UK

S. Irving (Operating Agent), Oscar Faber Consulting Engineers, Marlborough House, Upper Marlborough Road, St. Albans, Herts, AL1 3UT, Great Britain. Tel: +44(0)81-7845784 Telex: 889072 Fax: +44(0)81-7845700

*MDAES Perera, Environmental Systems Division, Building Research Establishment, Garston, Watford, WD2 7JR , UK Tel: +44(0)923 664486 Telex: 923220 Fax: +44(0)923 664010

M. Trim, Building Research Energy Conservation Support Unit (BRECSU), Building Research Establishment, Bucknalls Lane, Garston, Watford, Herts, WD2 7JR, Great Britain. Tel: +44(0)923 894040 Telex: 923220 Fax: (0)923-664010

P.J.J. Jackman, BSRIA, Old Bracknell Lane West, Bracknell, Berks, RG12 4AH, Great Britain. Tel: (0)344-426511 Telex: 848288

USA

*M. Sherman, Indoor Air Quality Division, Building 90, Room 3074, Lawrence Berkeley Laboratory, Berkeley, California 94720, USA. Tel: 510/486-4022 Telex: 910-366-2037 Fax: 510 486 5172 e-mail:MHSherman@lbl.gov

A. Persily, Building Environment Division, Center for Building Technology, Building 226, Room A313, National Institute for Standards and Technology, Gaithersburg MD 20899, USA. Tel: 301/975-6418 Fax: 301/975-4032

J. Talbott, Department of Energy, Buildings Division, Mail Stop Ce-131, 1000 Independence Avenue S.W., Washington D.C. 20585, USA. Tel: 202/586 9445 Telex: 710 822 0176 Fax: 202 586 4529/8134

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Published by
Air Infiltration and Ventilation Centre
University of Warwick Science Park
Sovereign Court
Sir William Lyons Road
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