

# Air Infiltration Review

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## Indoor Air Queries Ventilation

### A Perspective on Indoor Air '90 by Martin Liddament

*Indoor Air '90, a conference held in Toronto, Canada, on 29 July - 3 August 1990*

This, the fifth in a triannual series of conferences, was the largest yet, attracting a total of 530 technical papers, 100 exhibitors and over 1200 participants. Opened by the Canadian Minister of State for Housing, The Honourable Alan Redway, the essential theme of his introduction was the need to understand the factors which influence indoor air quality and to base any decisions on solid information. It was noted that while up to 90% of an individual's time may be spent within buildings, many occupants did not appreciate that the quality of indoor air was often worse than that of outdoor air. The Minister also emphasised the need to ensure that energy conservation strategies were compatible with indoor air quality requirements. This was especially significant since buildings accounted for a substantial proportion of total energy use and hence had a vital impact on global environmental issues as well as building environmental concerns.

The conference was structured in the form of a series of plenary sessions, parallel symposia and poster presen-

tations, forums and summary sessions. It was clear from the very start of the conference that ventilation and building airtightness was to become a much debated subject with conflicting discussion and confusion on the significance of these parameters on indoor air quality. The term "tightness" especially was used to describe many problems including the re-entry of exhaust air, poor ventilation, bad maintenance and the incorrect use of ventilation systems. In addition "inadequate ventilation" was used to express conditions of discomfort and poor temperature control. Interestingly where air change featured as a measurement, the relationship between air change and indoor air quality problems was generally very indefinite. In the first plenary session, Dr. Burge of East Birmingham Hospital in the United Kingdom presented a medical approach to the causes of building sickness. Based on surveys in 47 UK office buildings, Dr. Burge concluded that building sickness symptoms in naturally ventilated buildings were fewer than in air conditioned buildings although there were

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some bad naturally ventilated buildings and some good air conditioned buildings. In attempting to find a measurable cause for sick building symptoms, air change rates and parameters associated with poor ventilation such as high CO<sub>2</sub> concentration were eliminated. In his presentation, he pointed out that in the few buildings where air change measurements were made, ventilation rates in the naturally ventilated buildings were lower than those in the air conditioned buildings. In a subsequent Forum on Indoor Air Quality, Energy Conservation and the Environment, Mark Riley of Energy Mines and Resources, Canada asserted that Indoor Air Quality was compatible to energy conservation and that building energy conservation was a vital part of future environmental planning. He stated that it was the misapplication of design that creates problems. It was essential to treat the building as part of the ventilation system. In explaining the concept of the Canadian R2000 programme he showed that measured ventilation rates in these airtight dwellings were significantly greater than those of comparable dwellings of similar age.

Professor Ole Fanger of the Technical University of Denmark, also entered the ventilation argument by stating that the subject of ventilation leads to confusion. He pointed out that the same ventilation rates could be applied to two different buildings with contrasting results. He stated that it was essential to measure the source strengths of pollutants in buildings and he was surprised that instead, effort has focused on measuring the concentrations of pollutants. Source strength dictated whether a building was good or bad. Professor Fanger concluded that buildings as a source of pollution should be considered in any ventilation standard.

Other reported case studies attempted to find a link between ventilation rate and sickness symptoms. These too tended to be inconclusive or counter to expectation. Nagda of GEOMET Technology reported that during a summer monitoring period in which ventilation was varied by 25-30%, no marked differences in perceived IAQ or comfort were noted by the occupants.

Results presented by Giovanna Donnini of IRSST, Montreal, Canada, showed that occupant dissatisfaction of an office building actually increased with increasing outdoor air supply. This tended to correlate with the enhanced ingress of dust. In a dramatic measure to overcome a perceived "tight" building problem, a poster display by Gary Landrus of Building Ecology Group Research and Service, Canada, described the replacement of a sealed window system in a school by openable windows. Results showed that after a short period, dissatisfaction with the interior environment among occupants was at the same level as before installation took place.

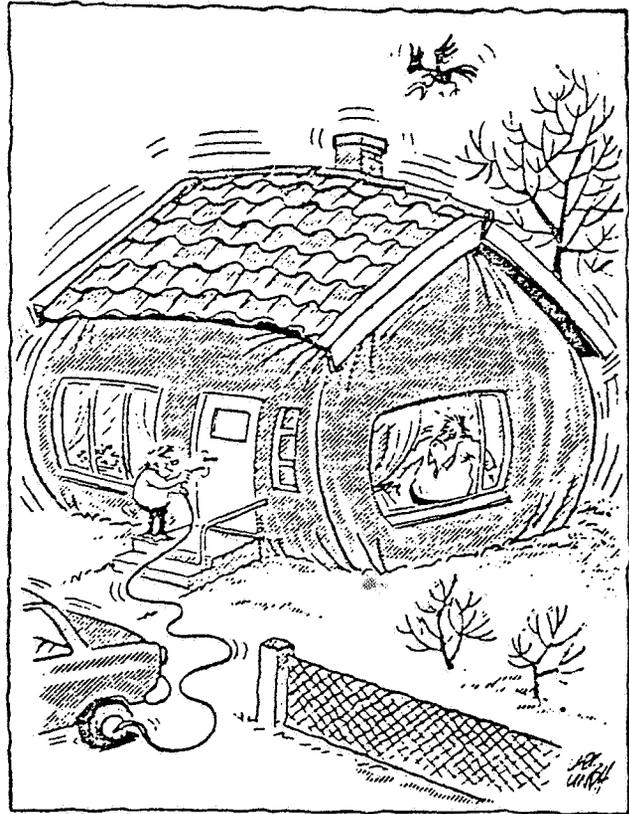


Figure 1: Build tight - ventilate right!

# Air Infiltration Review

Editor: Janet Blacknell

*Air Infiltration Review* has a quarterly circulation of 3,500 copies and is currently distributed to organisations in 40 countries. Short articles or correspondence of a general technical nature related to the subject of air infiltration and ventilation are welcome for possible inclusion in AIR. Articles intended for publication must be written in English and should not exceed 1,000 words in length. If you wish to contribute to AIR, please contact Janet Blacknell at the Air Infiltration and Ventilation Centre.

Conclusions and opinions expressed in contributions to *Air Infiltration Review* represent the author(s)' own views and not necessarily those of the Air Infiltration and Ventilation Centre.

In the review session on ventilation, Andy Persily of the US National Institute of Standards and Technology highlighted that the correlation between ventilation, pollutant levels and occupant perception was very weak. He also expressed concern over the apparent confusion about ventilation and thermal comfort; there seemed to be a lack of understanding of the role of ventilation in air quality. Tight buildings, he said, are not the problem and leaking buildings are not the answer. Andy then, in a flash back to Arne Elmroth's article in "Air Infiltration Review", Vol.1, No.4, August 1980 (Figure 1) reminded the audience that the ideal approach was to "Build Tight - Ventilate Right". Infiltration alone should not be relied upon to provide adequate ventilation.

The cause of sick building symptoms still remains elusive. Many pollutants were discussed, as indicated in Figure 2. Odour was considered an important indicator of unsatisfactory indoor air quality, and volatile organic compounds were the most widely cited of the odorous chemicals, this being closely followed by formaldehyde. Odourless compounds investigated included radon, carbon dioxide and carbon monoxide. However there seemed to be a general consensus that none of the popular pollutants contributed to the illness of occupants and in most instances their concentration, when measured, was below recognised standards. A possible pointer to a link between building sickness and a particular pollutant resided in the first and fourth plenary session presentations. In the first session, Dr. Burge noted that buildings with microbiological contamination tended to have increased numbers of symptomatic workers. However, he thought that in most cases, the airborne bacteria and fungi were not directly related to

symptoms but that soluble antigens in the air may be. In the fourth plenary session, Dr. David Miller, a biological research scientist from the Canadian Plant Research Centre, presented a detailed account of fungi and mycotoxins as contaminants in indoor air. In a "blind" study on a number of dwellings, those dwellings identified as having concentrations of toxigenic fungi were independently assessed as having air quality problems. He concluded that in the vast majority of buildings, airborne mycoflora are quantitatively lower and qualitatively identical to outdoor air. However, for those structures where this is not so, fungi are almost certainly the cause of totally preventable illness.

As a final thought, in view of the lack of specific information on the relationship between ventilation and indoor air quality problems, it is suggested that immediate consideration should be given to undertaking a comprehensive international review of existing information and data on this topic. This could provide valuable input to future ventilation specifications as well as clarifying the role of ventilation as an indoor air quality parameter.

*The proceedings of this conference, "Indoor Air '90, the Fifth International Conference on Indoor Air Quality and Climate", are available in five volumes, from:*

*International Conference on Indoor Air Quality and Climate, Inc., 2344 Haddington Crescent, Ottawa, Ontario K1H 8J4, Canada. Approx. price: \$200.*

*A list of contents are available on request from the AIVC.*

## Selected Pollutants Referenced in Indoor Air '90

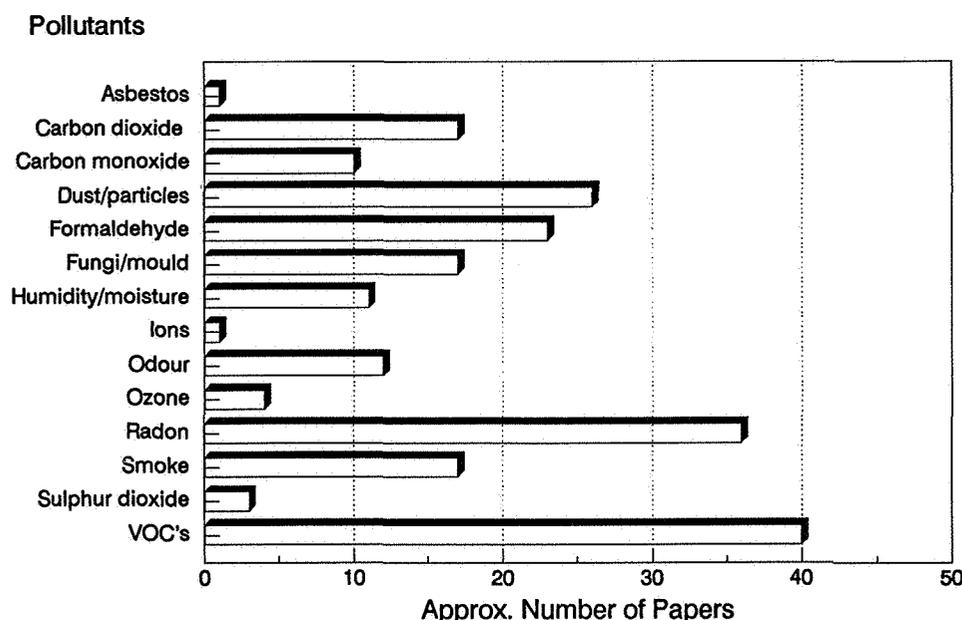


Figure 2

# Including Furnace Flue Leakage in a Simple Air Infiltration Model

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## Introduction

Although there are many simple infiltration models already available none of them have an appropriate method of dealing with what is often the single largest leak in a building; a furnace or fireplace flue. Flues are different from the distributed leakage used in simple infiltration models. Flues represent 10% to 30% of the total building leakage all of which is concentrated at one location above the ceiling height. The flue top is often unsheltered when the rest of the house is sheltered from wind. Because the flue is filled with room air most of the time this leads to an increased stack effect. The pressure-flow exponent,  $n$ , for a furnace flue is about 0.5 rather than the value 0.6 to 0.8 typical of the rest of the building leaks.

The Alberta Infiltration Model, AIM-2, gives improved estimates by incorporating the  $Q = C\Delta P^n$  characteristic into the model from first principles, treating the flue as a separate leakage site with its own wind shelter, and locating the flue outlet above the roof, rather than grouping the flue leakage with the ceiling leaks as in other models.

## Describing the Leakage Distribution

A power law pressure-flow relationship for a building envelope is assumed

$$Q = C_{total}\Delta P^n \quad (1)$$

Measurements on several test houses in the present study demonstrated that a single value for  $C_{total}$  and  $n$  accurately describe building leakage flows over a wide pressure range, from less than 1 Pa to over 50 Pa.

Sherman (1980) introduced the idea of using leakage distribution parameters  $X$  and  $R$  to describe the building envelope flow rates in terms of stack and wind factors,  $f_s$  and  $f_w$ . In AIM-2 we followed this approach, adding an additional flue fraction parameter,  $Y$ .

$$R = \frac{C_{(ceiling)} + C_{(floor)}}{C_{total}} \quad \text{"ceiling-floor sum"} \quad (2)$$

$$X = \frac{C_{(ceiling)} - C_{(floor)}}{C_{total}} \quad \text{"ceiling-floor difference"} \quad (3)$$

$$Y = \frac{C_{flue}}{C_{total}} \quad \text{"flue fraction"} \quad (4)$$

Stack and wind effects in AIM-2 are combined as if their independent pressure differences add linearly, and an additive correction term is introduced to account for the interaction of the wind and stack effects in producing the internal pressure, see Walker and Wilson (1990). Model validation, discussed later, used data sets from the Alberta Home Heating Research Facility chosen to be dominated by wind or stack effects, so that this correction term was negligible. In this way, AIM-2 could be tested against independent infiltration measurements without assuming any empirical constants other than the pressure coefficients available from existing wind tunnel data sets.

## Comparison of AIM-2 with Other Infiltration Models

The two models that most closely resemble the Alberta Infiltration Model (AIM-2) use variable leakage distribution. They are Sherman's orifice flow model from Sherman and Grimsrud (1980), often referred to as the LBL model, and a variable flow exponent (VFE) model, adapted by Reardon (1989) from Yuill's (1985) extension of Sherman's model to power law leakage. One of the other models chosen for comparison, Shaw (1985), is based on empirical fitting and superposition of the stack and wind pressure terms to measured data. The fourth model, Warren and Webb (1980) does not specify leakage distribution variation. The significant differences between AIM-2 and the LBL and VFE models are:

- The attic space above the ceiling is assumed in the LBL and VFE models to have a zero pressure coefficient. The attic pressure coefficient in AIM-2 is taken to be a weighted average of the pressure coefficients on the eave and end wall vents, and the roof surface vents. The eave vents are assumed to have the same pressure as the wall above which they are located.
- There is no furnace flue in the LBL or VFE model. In these models any furnace flue leakage is simply added to the ceiling leakage, and sees the attic pressure. In AIM-2 the furnace flue is incorporated as a separate leakage site, at the flue exit height above the roof. The flue is assumed to be filled with

indoor air at room temperature, and exposed at its top to a pressure set by wind flow around the rain cap.

- The floor leakage in Sherman's LBL model and in VFE was located above a crawlspace that was assumed to have a zero pressure coefficient. In AIM-2 the crawlspace pressure is taken as the average of the four outside wall pressures from wind effect. AIM-2 also deals with a house with a full basement or a slab-on-grade, where "floor" leakage is the crack around the floor plate resting on the foundation, plus cracks, holes and other leakage sites in the concrete foundation above grade. These floor level leakage sites are uniformly distributed around the perimeter of the house, and exposed to the same pressure as each of the walls on which they are located.
- The LBL model assumes orifice flow with  $n = 0.5$  in  $Q = C\Delta P^n$  of each leak. Both VFE and AIM-2 assume a single value of  $n$  in the range 0.5 to 1.0 for leakage through the floor, walls, and ceiling. The flue is assumed to have a value of  $n = 0.5$ .
- Both LBL and VFE assume that wind flow,  $Q_w$ , and stack flow  $Q_s$ , combine in quadrature as a sum of squares. AIM-2 assumes the flows add as a pressure sum plus an interaction term that accounts empirically for a wind-induced shift of the neutral pressure level.

### AIM-2 Stack Effect

The stack factor,  $f_s$ , is defined by

$$Q_s = C_{total} f_s P_s^n \quad (5)$$

where, in terms of the ceiling height,  $H$ , of the building, the stack effect reference pressure is

$$P_s = \rho_o g H \left( \frac{T_i - T_o}{T_i} \right) \quad (6)$$

with subscript "o" indicating outdoor and "i" indicating indoor conditions.

The stack factor  $f_s$  was determined by a numerical solution of the non-linear inflow-outflow balance equations, and is approximated in AIM-2 by

$$f_s = \left( \frac{1 + nR}{n + 1} \right) \left( \frac{1}{2} - \frac{1}{2} M^{2/4} \right)^{n+1} + F \quad (7)$$

The functional form of this approximation was selected to produce the correct behaviour of  $f_s$  for ceiling-floor difference ratio limits of  $X = 0$  and  $X = \pm 1$ , and at the limits  $R = 0$  where all leakage is concentrated in the

walls, and  $R = 1$  where all the leaks are in the floor and ceiling. The factors  $M$  and  $F$  are defined by

$$M = \frac{(X + (2n + 1)Y)^2}{2 - R} \quad \text{for} \quad \frac{(X + (2n + 1)Y)^2}{2 - R} \leq 1 \quad (8)$$

with a limiting value of

$$M = 1.0 \quad \text{for} \quad \frac{(X + (2n + 1)Y)^2}{2 - R} > 1 \quad (9)$$

The additive flue function  $F$  is,

$$F = nY (\beta_f - 1)^{\frac{(3n-1)}{3}} \left( 1 - \frac{3(X_c - X)^2 R^{1-n}}{2(\beta_f + 1)} \right) \quad (10)$$

where  $\beta_f$  is the ratio of flue height to ceiling height above floor level, and

$$X_c = R + \frac{2(1-R-Y)}{n+1} - 2Y(\beta_f - 1)^n \quad (11)$$

The variable  $X_c$  is the critical value of the ceiling-floor difference fraction  $X$  at which the neutral level passes through the ceiling in the exact numerical solution. For  $X > X_c$  the neutral level will be above the ceiling, and attic air will flow in through the ceiling. For  $X < X_c$  room air will exfiltrate through the ceiling. This critical value of  $X_c$  is useful in determining whether moist indoor air will exfiltrate through the ceiling and condense in attic insulation in winter. The role of the flue in reducing ceiling exfiltration is evident from the contribution of the flue leakage factor,  $Y$ , in (11).

The stack factor is shown in Figure 1 for typical values of  $n = 2/3$ ,  $\beta_f = 1.5$  for a house with no flue,  $Y = 0$ , and for a flue with 20% of the total leakage area,  $Y = 0.2$ . It is apparent that treating the flue as a separate leakage site with a stack height above the ceiling has a significant effect on the stack factor  $f_s$ .

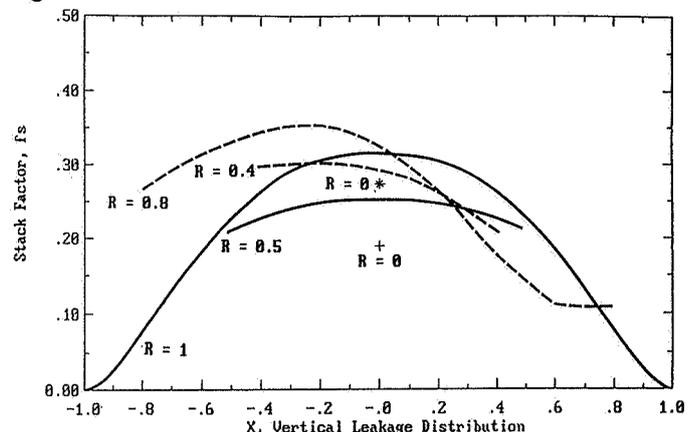


Figure 1: AIM-2 (Equation 7) for Stack Factor,  $f_s$ , with no Flue Leakage ( $Y=0$ ) solid line and +, and with 20% of Leakage in the Flue ( $Y=0.2$ ) and  $\beta = 1.5$  dashed line and \*.

## AIM-2 Wind Effect

The wind induced infiltration rate  $Q_w$  is defined in terms of wind factor  $f_w$  by

$$Q_w = C_{total} f_w P_w^n \quad (12)$$

where the reference wind pressure is expressed in terms of the unobstructed windspeed (with no local shelter) at eaves height, and an overall shelter coefficient,  $S_w$ ,

$$P_w = \rho_o \frac{(S_w U_e)^2}{2} \quad (13)$$

The exact numerical solution for  $f_w$ , and its approximating function in AIM-2 depend on the set of wind pressure coefficients used. Using the pressure coefficients from Akins (1979), the approximating function for wind factor for slab-on-grade or basement construction types is

$$f_w = 0.19(2-n) \left( 1 - \left( \frac{X+R}{2} \right)^{1.5-Y} \right) - \frac{Y}{4} (S-2YS^4) \quad (14)$$

$$\text{where } S = \frac{X+R+2Y}{2}$$

This form for  $f_w$  was chosen to produce the correct behaviour for the limiting values of all leakage concentrated in either walls, floor or ceiling, and for  $X=0$  where the floor and ceiling leakage are equal. Surprisingly, the flue height  $B_f$  does not appear in (14), because the exact solution indicated that the dependence on  $B_f$  is felt very weakly through the change in wind speed at the flue top.

For a house with a crawl space, the pressure inside the crawl space may be approximated by the average of the four walls which change the dependence of  $f_w$  on  $X$  and  $R$ . Approximating functions for this type of construction are discussed by Walker and Wilson (1990).

The wind factor calculated from equation (14) is shown in Figure 2 for  $n = 2/3$ ,  $Y = 0.2$ , and for no flue,  $y = 0$ . This shows that there is little difference between considering the flue leakage as a hole in the ceiling, venting into the attic, or as a separate leakage site with its own flue cap pressure coefficient above the roof. The major advantage of the separate flue leakage site is to allow it to have a different wind shelter than the rest of the building.

The exact numerical solution for  $f_w$ , used the wind pressure coefficient data set of Akins (1979), with wind normal to the upwind wall, and all walls of the same length. Three other wind tunnel data sets, ASHRAE

(1989), AIVC (1986) and Wiren (1984), for wall and roof pressure coefficients were also used to find numerical solutions for  $f_w$ . These other sets of pressure coefficients produced wind factors that are functionally similar, but with a difference in magnitude. The two extreme results are for the Wiren and ASHRAE data sets, that produce values of  $f_w$  10-20% larger and 10-20% smaller than Akins data set.

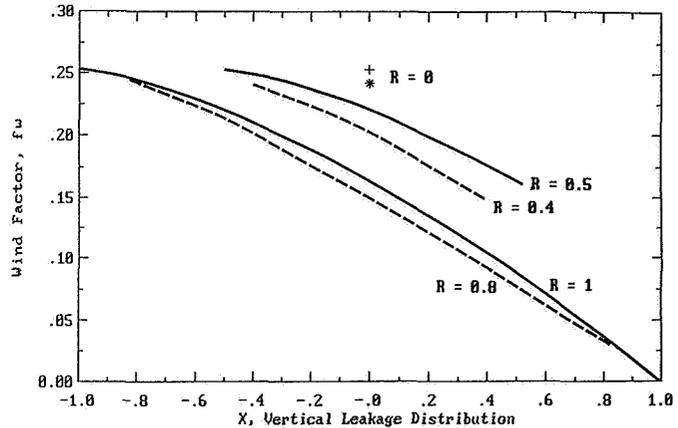


Figure 2: AIM-2 (Equation 14) for Wind Factor,  $f_w$ , with no Flue Leakage ( $Y=0$ ) solid line and +, and with 20% of Leakage in the Flue ( $Y=0.2$ ) and  $B_f = 1.5$  dashed line and \*

## Shelter Coefficients

Local shielding by nearby buildings trees and obstructions is very difficult to estimate by simply inspecting the building site, and uncertainty in estimating the local shelter coefficient  $S_w$  in (13) is often the major source of error in estimating wind driven air infiltration rates.

In AIM-2, the shelter coefficient  $S_{wo}$  for the building walls is combined with a different coefficient  $S_{wflue}$  for the top of the flue

$$S_w = S_{wo} (1 - Y) + S_{wflue} (1.5Y) \quad (15)$$

$S_{wflue} = 1.0$  for an unsheltered flue, which protrudes above surrounding obstacles, and  $S_{wflue} = S_{wo}$  for a flue top which has the same wind shelter as the building walls. The factor 1.5 is an empirical adjustment found by comparing AIM-2 to an exact numerical solution for each leakage site with its own shelter and pressure coefficient.

## Comparison of AIM-2 with Measured Infiltration

AIM-2 has been validated by comparing its predictions to air infiltration measurements in two houses at the Alberta Home Heating Research Facility. Continuous infiltration measurements were carried out in the test

houses using a constant concentration SF<sub>6</sub> tracer gas injection system in each house described by Wilson and Dale (1985). Envelope leakage characteristics were measured in the two houses using fan pressurization over the range from 1 Pa to 75 Pa.

House Number	Flue Configuration	Flow Coefficient m <sup>3</sup> /s Pa <sup>n</sup>	Exponent n	Leakage area A <sub>0</sub> at 4 Pa cm <sup>2</sup>
4	closed	0.007	0.7	65
4	open with 7.5 cm dia orifice	0.010	0.66	93
5	open 15 cm dia. flue	0.020	0.58	158

Table 1: House leakage characteristics

The single storey houses are of wood frame construction with polyethylene vapour barriers and full concrete basements. Details of the house construction are given in Wilson and Dale (1985).

For house 4 with the flue blocked it was estimated that 50% of the leakage is in the walls and 25% in the floor and 25% in the ceiling. For house 4 with a 7.5 cm diameter flue it was estimated that 40% of the leakage was in the walls, 30% in the floor and 15% in the ceiling. For house 5 with a 15 cm diameter flue 60% of the leakage was estimated to be in the flue, 30% in the walls, 5% in the floor and 5% in the ceiling.

The predictions of AIM-2 and these four models, are compared to measured data in Figures 3 and 4 for unsheltered conditions (north and south winds) in house 5 with a 15 cm diameter flue, and for house 4 with the flue blocked. These results show that AIM-2 has the best overall performance for houses with furnace flues, mainly because the furnace flue is treated as a separate leakage site with its own wind pressure coefficient.

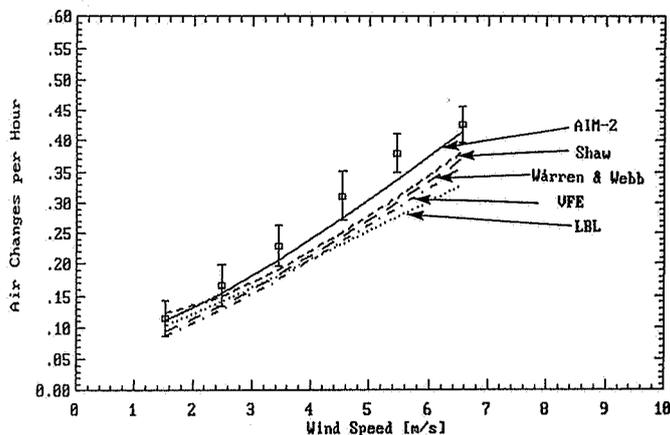


Figure 3: Comparison of ventilation models with measured data for unshielded windspeed dependence (North and South winds) in house #5 with an open 15 cm diameter flue with  $\Delta T \geq 10^\circ\text{C}$  and  $U \geq 1.5$  m/s (279 hours)

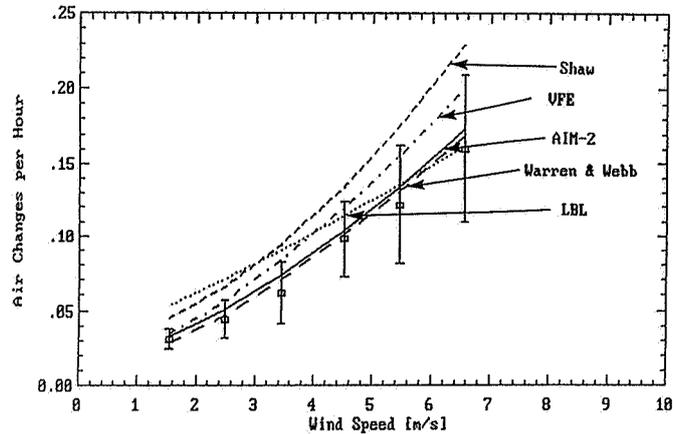


Figure 4: Comparison of ventilation models with measured data for unshielded windspeed dependence (North and South winds) in house #4 with blocked flue, and  $\Delta T \leq 10^\circ\text{C}$  and  $U \geq 1.5$  m/s (285 hours)

Figure 5 compares the windspeed dependence of the models for house 5 with an open 15 cm diameter flue where the house is heavily sheltered by a row of adjacent test houses (east and west winds). All the models except AIM-2 underpredict the wind effect infiltration rate  $Q_w$  significantly because they are unable to account for unshielded flue leakage on a shielded building. The same data that were binned for Figure 5 are shown individually in Figure 5b. This shows the amount of scatter present in the measured hourly data and the need for data binning for clearer model comparisons. Figure 6 illustrates the superior performance of AIM-2 in predicting stack effect flow  $Q_s$  for house 4 with a 7.5 cm diameter restriction orifice in the flue.

In conclusion, AIM-2 with separate flue leakage has shown a significant advantage over existing simple air infiltration models. Current efforts to improve AIM-2 are focused on developing accurate methods for estimating wind shelter from surrounding structures.

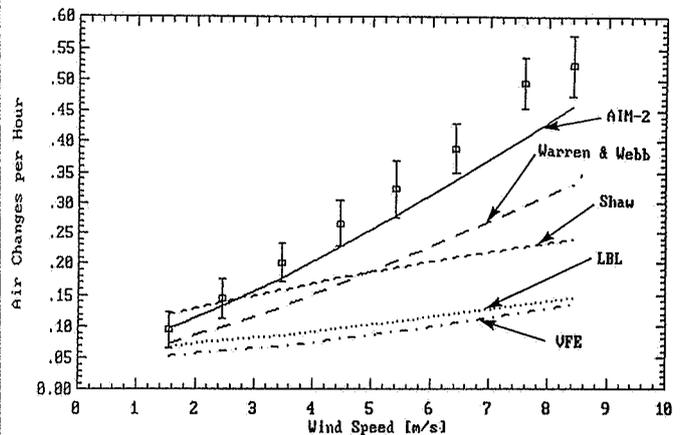


Figure 5: Comparison of ventilation models with measured data for shielded windspeed dependence (East and West winds) in house #5 with an open 15 cm diameter flue with  $\Delta T \leq 10^\circ\text{C}$  and  $U \geq 1.5$  m/s (461 hours)

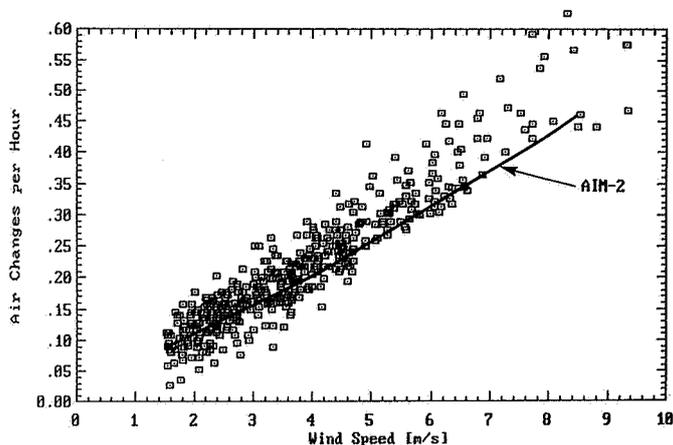


Figure 5b: Comparison of AIM-2 with unbinned measured data for shielded windspeed dependence (East and West winds) in house #5 with an open 15 cm diameter flue with  $\Delta T \leq 10^\circ\text{C}$  and  $U \geq 1.5$  m/s. 461 hours of unbinned data.

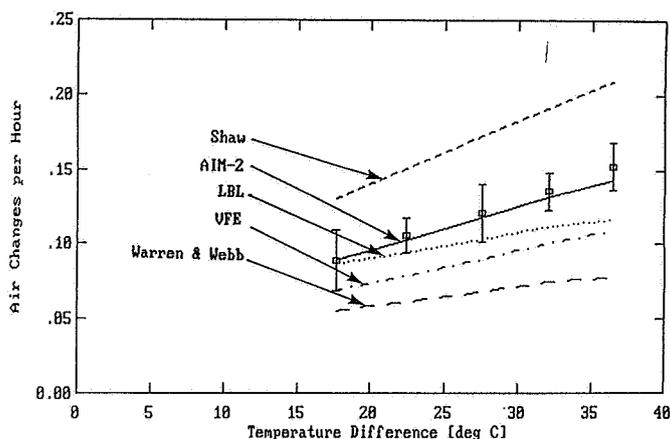


Figure 6: Comparison of ventilation models to measured data for DT dependence in house #4 with an open 7.5 cm diameter orifice in the flue, with  $\Delta T \leq 10^\circ\text{C}$  and  $U \geq 1.5$  m/s.  $R = 0.3$ ,  $X = 0$ ,  $Y = 0.4$ , and  $B_f = 1.5$  (102 hours).

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# An Experiment for Airflow Determination by Quadratic Programming

by Jan-Bertil Mattson

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During the past decade a multitude of tracer gas techniques to measure ventilation rates in buildings have been developed. These techniques have proven very useful for a building which may be treated as a single zone. However, as the need to understand more complex buildings increases, it has become necessary to develop more advanced mathematical methods.

In buildings with mechanical ventilation there are spaces with substantial pressure differences, which bring exfiltration, infiltration and transferred air between the rooms. For such buildings we have converted a multiple cell theory to a quadratic programming problem, and developed a computer programme, MCSPID for airflow identification. Today MCSPID is in practical use to simultaneously determine flow rates for air supply, exhaust air, transferred air, infiltration and exfiltration with a single tracer gas. The purpose of this paper is to describe an application in a seven room building, where all airflows are determined in an active tracer gas experiment.

## Multiple Cell Model

The basic theory is probably well known. The mixing in each room or cell is supposed to be perfect and instantaneous, and the airflows are constant. In that case the flow model can be represented in matrix form as:

$$V \cdot \dot{c}(t) = Q \cdot c(t) + p(t)$$

where  $c(t)$  = tracer gas concentration vector

$V$  = diagonal volume matrix

$p(t)$  = tracer gas supply vector

$Q$  = flow matrix

The flow matrix  $Q$  contains the interflows between the cells, except for the diagonal elements, which are the entire flow rate from each cell. If the flow system consists of  $n$  cells, both the flow and volume matrix have dimension  $n \times n$  and the concentration and gas supply vector have  $n \times 1$ . There are  $n$  volumes and  $n^2$  flows to identify, in all  $n(n+1)$  model parameters.

The measurements proceed during  $m$  sampling intervals of length  $T_s$ . If  $Q_{ii}$  is the total air flow from cell  $i$ , and  $Q_{i1}, Q_{i2}, \dots, Q_{in}$  are the air flows to cell  $i$  from cells  $1, 2, \dots, n$ , then the momentary mass balance during sample  $t_k$ ,  $k = 1, 2, \dots, m$  for cell  $i$  is:

$$V_{ii} \cdot \dot{c}_i(t_k) - Q_{i1} \cdot c_1(t_k) - \dots + Q_{ii} \cdot c_i(t_k) - \dots - Q_{in} \cdot c_n(t_k) = p_i(t_k)$$

A column-vector  $x_i$  can be formed for the volume and airflows (model parameters), a matrix  $A_i$  for concentrations, and a column-vector  $p_i$  for tracer gas supply. Each row in  $A_i$  and  $p_i$  contains concentration and gas supply during one sample.

$$x_i = [V_{ii}, -Q_{i1}, +Q_{ii}, \dots, -Q_{in}]^T$$

$$A_i = [\dot{c}_i(t_k), -c_1(t_k), \dots, c_i(t_k), \dots, -c_n(t_k)]$$

$$p_i = [p_i(t_k)]$$

The mass balances for cell  $i$  are then simplified to:

$$A_i \cdot x_i = p_i$$

To avoid derivatives and reduce measurement noise, Bjorn Hedin has derived a method for integrating the mass balance equation (Hedin 1989). If we however remain at the momentary expression, the time derivatives are calculated with an iterative method, using old model parameters:

$$\dot{c}_i(t) = D \cdot [c_i(t + T_s) - c_i(t)]$$

$$D = V^{-1} \cdot Q \cdot [\exp(V^{-1} \cdot Q \cdot T_s) - 1]^{-1}$$

As we are interested in identifying airflows between all cells, the model is extended for a complete multiple cell system. The column-vectors for model parameters and gas supply are extended to include all cells, and a new matrix  $A$  of concentrations is established, where  $A_i (i=1 \dots n)$  are located into the diagonal.

$$A = \begin{pmatrix} A_1 & 0 & & 0 \\ 0 & A_2 & & 0 \\ \cdot & \cdot & \dots & \cdot \\ \cdot & \cdot & & \cdot \\ 0 & 0 & & A_n \end{pmatrix}$$

$$x = (x_1, x_2, \dots, x_i, \dots, x_n)^T$$

$$p = (p_1, p_2, \dots, p_i, \dots, p_n)^T$$

Then the mass balance for a multiple cell becomes:

$$A \cdot x = p$$

As the dimension of  $A_i$  is  $(n+1) \times m$ , the dimension of  $A$  becomes  $n(n+1) \times nm$ . The vector  $x$  contains all  $n(n+1)$  model parameters, and receives the dimension

$n(n+1)*1$ . The vector  $p$  contains all tracer gas flow rates (for  $n$  cells during  $m$  sampling intervals), and has the dimension  $mn*1$ .

## A Linear Programming Problem

If the number of sampling intervals is exactly one more than the number of cells a solution will be possible, which however will not give any proper model parameters. Considerably more sampling intervals are necessary, which result in an overdetermined equation system. The solution will be the volumes and airflows which minimize the norm  $\|r\|$  of the residuals  $r$ .

$$\|r\| = Ax - p$$

If the solution is to correspond to a flow system it remains to consider physical constraints, such as volumes and interflows, which have to be positive or zero. The same is also true for infiltration and exfiltration, which are the negative sum of each row and column of  $Q$ . The last constraint is expressed with the vector  $G$ , whose elements are  $+1, -1$  or  $0$ .

$$x \geq 0 \quad Gx \geq 0$$

Lars Jensen first formulated a multiple cell model with linear constraints during Roomvent 1987 (Jensen 1988). He chose a linear norm, where  $\|r\| = \sum r_i$  and applied a linear programming method to determine the airflows.

## A Quadratic Programming Problem

Another procedure used for identification is the method of least squares, where the model parameters are selected by minimizing the sum of the squared error. The norm is accordingly:

$$\begin{aligned} \|r\|^2 &= \sum r_i^2 = r^T r = (Ax - p)^T (Ax - p) \\ &= p^T p - p^T A x - x^T A^T p + x^T A^T A x \end{aligned}$$

The constant term  $p^T p$  has no influence on  $x$  at minimizing. The expression is further simplified by:

$$C = -2A^T p \quad B = A^T A$$

Instead of  $\|r\|^2$  we have received the following quadratic equation to minimize:

$$f(x) = C^T x + x^T B x$$

which still has the constraints  $Gx \geq 0$  and  $x \geq 0$ .

The identification of an air flow system has thus become a problem of minimizing a quadratic equation under certain constraints. This is a quadratic programming problem, which is solved by Lemke's complementary pivot algorithm. This method for identification of multiple cell systems was introduced by Bjorn Hedin at the AIVC conference in Finland 1989 (Hedin 1989), and applied by Lars Jensen in his programme MCSPID.

One important benefit of converting the model to a quadratic equation is that the dimensions of the matrixes are not dependent on the number of samples. The vector  $C^T$  has the dimension  $n(n+1)*1$ , and the matrix  $B$  has  $n(n+1)$ , and they are therefore only dependent on the number of cells. The influence of measurement errors decreases as the number of samples increases. Therefore it is possible to identify a large air flow system on personal computers, almost independent of the number of measurements, by converting the multiple cell model to a quadratic programming problem.

## Building and Experiment

Tracer gas measurements for airflow determination by quadratic programming have been performed in our experimental building Minilab (Figure 1), which consists of seven rooms and has mechanical air supply and exhaust air. The air conditioning unit is completed with a damper for recirculated air. Each room forms a cell and is furnished with a fan to assure a perfect and instantaneous mixing. The main air supply and exhaust air ducts make another two cells, but with small volumes. The size of each room is  $24.3 \text{ m}^3$  or  $29.4 \text{ m}^3$ . Some rooms are connected with ducts and fans for transferred air, which increases the pressure difference and brings infiltration and exfiltration.

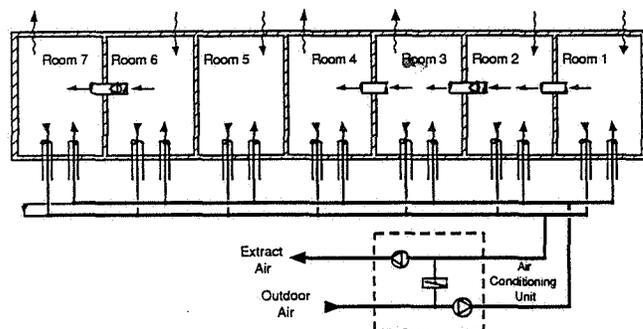


Figure 1: Minilab

## Airflows Measured With Tracer Gas

The measurements are performed with constant tracer gas supply during about one hour in each cell. The throttle valve for gas supply is transferred to a personal computer, where an input file to the programme MCSPID is established. Quadratic programming takes general physical constraints into consideration. There are however further constraints for each special experiment, such as fixed parameters, which are to be stated in the input file. The volumes are fixed, and we may also specify where transferred air is impossible, as well as infiltration or exfiltration.

The output file contains a matrix with calculated air flow rates and volumes. The result from the Minilab building is to be found in Figure 2. Because of numerical problems there are small differences in total flow rates. Recirculated air is 39%.

	From Room 1	From Room 2	From Room 3	From Room 4	From Room 5	From Room 6	From Room 7	From Air Supply	From Exhaust Air	Infiltration	Total
To Room 1								12.3		1.1	13.4
To Room 2	8.9							4.7		4.7	18.3
To Room 3		15.6						4.2			19.8
To Room 4			9.1					4.0			13.1
To Room 5								7.4		4.4	11.8
To Room 6								16.1		4.2	20.3
To Room 7						12.6		8.3			20.9
To Air Supply									62.6	97.3	159.9
To Exhaust Air	3.7	2.8	4.4	13.0	11.2	7.7	18.0	103.1			163.9
Exfiltration			6.3					2.9		101.3	110.5
Total	12.6	18.4	19.8	13.0	11.2	20.3	20.9	160.1	163.9	111.7	

Figure 2: Airflow rates (l/s) measured with tracer gas

## Airflows Measured with Orifice Plates

All ducts for air supply, exhaust air and transferred air are furnished with an orifice plate, where the airflow rate is measured with an u-tube (Figure 3). The measuring range is 9-25 l/s and the uncertainty in measurement is 7.5 %, according to the manufacturer. Values in brackets are outside measuring range, and infiltration and exfiltration are estimated. It appears from the figures, that the measurements with tracer gas and orifice plates differ max 15% (when inside measuring range).

	From Room 1	From Room 2	From Room 3	From Room 4	From Room 5	From Room 6	From Room 7	From Air Supply	Infiltration	Total
To Room 1								11.1 (0.9)		12.0
To Room 2	9.0							(3.9)	(4.7)	17.6
To Room 3		15.0						(3.5)		18.5
To Room 4			8.9					(3.0)	(0.5)	12.4
To Room 5								?	?	10.7
To Room 6								14.9	(5.7)	20.6
To Room 7						14.4		(3.0)		17.4
To Exhaust Air	(3.0)	(2.6)	(3.9)	12.4	10.7	(6.0)	15.3			
Exfiltration			(5.7)				(2.2)			
Total	12.0	17.6	18.5	12.4	10.7	20.4	17.5			

Figure 3: Air flow rates (l/s) measured with orifice plates

## Comparison Between Modelled and Measured Concentrations

Beyond the airflow rate estimations, modelled concentrations are calculated with estimated air flow rates and actual tracer gas supply. The deviation between the model and the measurements, is tabulated in Figure 4. The most substantial deviation is for room 7, where the standard deviation is 11 ppm.

	Mean dev.	Standard dev.	Max. dev.	Min. dev.
Room 1	-0.46	7.43	23.56	-14.95
Room 2	-1.03	5.04	9.55	-11.49
Room 3	-0.93	7.12	11.79	-24.08
Room 4	-0.85	6.62	14.21	-18.41
Room 5	-0.44	2.77	13.93	-5.33
Room 6	-0.29	5.64	17.13	-10.40
Room 7	1.70	11.25	23.33	-26.14
Air supply	-0.22	2.42	3.84	-9.68
Exhaust air	-0.11	2.81	6.62	-8.79

Figure 4: Deviations between measured and estimated concentration

The concentrations according to the model and as measured, are shown in Figure 5, where it appears that all pairs of curves are almost identical. Tracer gas is first brought into the air supply duct, which causes the first small peak after about 80 min. Then gas is supplied to each of the seven rooms and at last into the exhaust air duct.

According to the theory the concentrations in all cells have to be measured at the same time, but in reality they were measured 30 seconds after one another. Interpolation or Kalmanfilter would have improved the model, which however has not been done.

These measurements took place in a building without complicated spaces, and where perfect mixing was rather easy to achieve. Under such circumstances, it was possible to find a method of quadratic programming, which has been proved by this experiment.

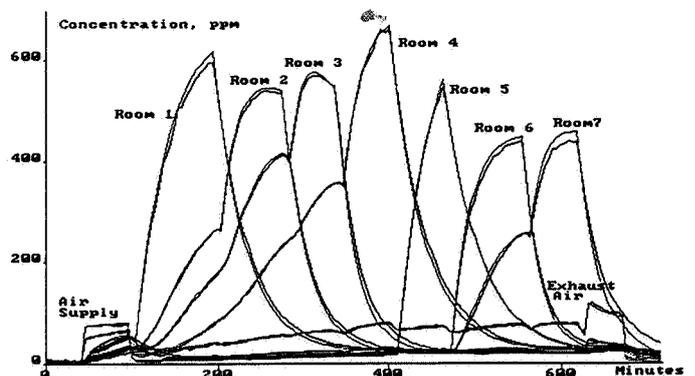


Figure 5: Tracer gas concentration in nine cells

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# Ventilation and Energy Loss Rates After Opening a Window

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## Introduction

Multizone air infiltration and ventilation modelling is in progress, both by the development of computer programs with a modular structure and a user friendly interface (Ref.1) and by R & D on the algorithms which can be used as a basis for those modules (Refs.1,2).

In building simulations it is important to be able to evaluate user behaviour on the ventilation and energy loss rates. This implies that one should be able to predict firstly, how often and how long occupants open and close windows, and secondly how ventilation and energy loss rates depend on building parameters, meteorological parameters and opening time.

As a solution to the first problem, a stochastic model of user behaviour has been developed (Ref.3), which generates time series of window opening angles with the same statistics as the measured openings for the heating period.

On the other hand, previous discussions of the ventilation rates through a given single large opening (Refs. 4-9), showed that to calculate the stack effect it is sufficient to know the inside-outside air temperature difference ( $T_{in}-T_{out}$ ), while for the calculation of the influence of the wind one relies on empirical relations (Refs.4,7). In all these discussions, the ventilation rate ( $V$ ) is found from the velocity in the window opening and the window size, while the heat loss rate by single sided ventilation equals

$$Q = c \rho V (T_{in} - T_{out}) \quad (1)$$

In this note we discuss the problem (concealed by the latter statement) of calculating the inside air temperature which varies with time and is, when not measured directly, in general not known. The inside air temperature ( $T_{in}$ ), which is in between the outside air temperature ( $T_{out}$ ) and the building temperature, can be calculated with a simple model recently developed and validated for a few real cases (Refs.10,11). The new ventilation model takes heat transfer between the air and walls into account, uses a dynamic wall thermal model and includes as limiting cases the situations where (Figure 1):

(i) the cold air enters as a gravity wave, replaces the warm air completely (adiabatic walls, no heat transfer),

and the stack ventilation comes to a halt after one air change (Refs.5,8).

(ii) the entering cold air is warmed up to the building temperature (infinite heat transfer), the stack ventilation rate is constant and calculated from the temperature difference between the outside air and the building (Refs.4,6,7).

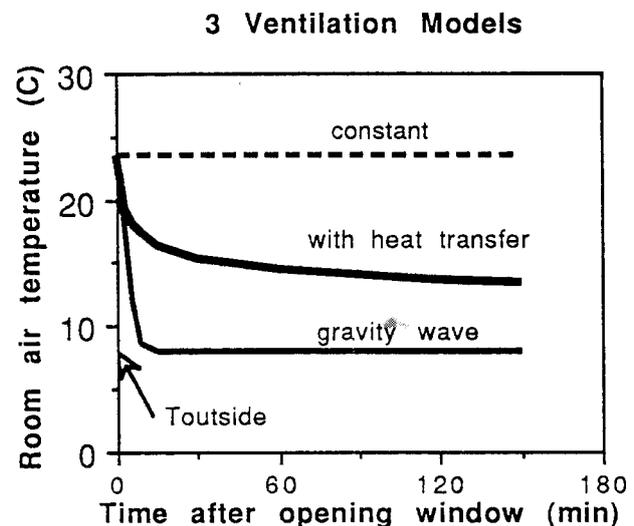


Figure 1: Predictions of three ventilation models compared for a particular case. The constant temperature and the gravity wave model are limiting cases of the model including heat transfer.

## Mechanisms

The model follows naturally from the following observations on an empty room in thermal equilibrium with the surrounding building structure: after opening a single window to the cold outside one observes first a sudden drop in air temperature, followed by a slow decrease in temperature. One notes that the wall surface temperature is higher than the air temperature and that the slow decrease in air temperature is due to a decrease in wall surface temperature. Moreover, it is found that the relative magnitude of the initial temperature drop is larger in smaller rooms and for larger window areas.

Finally the new model (Figure 2), couples the heat flow rate through the aperture (Equation 1) with heat transfer between the air and the walls ( $T_{in}$  drops below  $T_w$ ) and with a wall surface temperature  $T_w(t)$ , decreasing with time. The model introduces two new parameters, (1) the heat exchanging wall surface area and (2) the average thermal permeability of the walls.

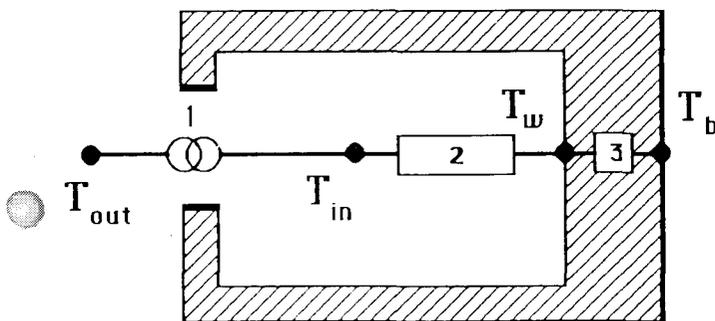


Figure 2: Ventilation and thermal models coupled in a four node network. 1.Heat source  $Q$  (Equation 1). 2.Boundary layer resist  $1/C_2S_i h_c$ (Equation 3). 3.Dynamic thermal wall resistance (Equation 4).  $T_b = T_w(0)$ , initial wall temperature.

**Ventilation.** The heat loss rate due to ventilation is calculated with Equation 1, and for the ventilation rate one could take the expressions proposed in References 4,7 or 9. The model has been tested for large inside outside temperature differences and small wind velocities. In this way the uncertainties concerning wind induced ventilation were avoided. For the single sided ventilation through an opening of height  $H$  and width  $W$ , the stack effect equation is used with a discharge coefficient  $C_1 = 0.6$ :

$$V = 2/3(WH/2)C_1\sqrt{gH(T_{in}-T_{out})/\tau} \quad (2)$$

**Heat transfer.** The temperature difference between the air and the wall surface,  $T_{in}-T_w$ , is described by a heat transfer coefficient  $h$ , and equals  $q/h$ , where  $q$  is the heat flux density. A fixed value of  $h = (6 \pm 1) W/m^2K$  appeared to be consistent with the measurements (Refs.10,11). The value of  $q$  is calculated from the total heat flux  $Q$  (Equation 1), and the heat exchanging wall surface area  $C_2S_i$ :

$$T_w - T_{in} = Q/C_2S_i.h \quad (3)$$

The coefficient  $C_2$  is the fraction of the total wall surface area  $S_i$ , which is active in the heat transfer process (Refs.10,11). The meaning of  $C_2$  is illustrated in Figure 2. The larger the distance between the window and the ceiling, the smaller is  $C_2$  (Figure 3b).

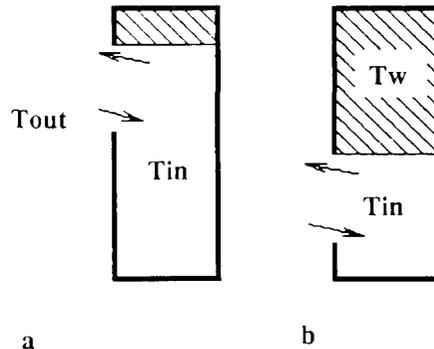


Figure 3: Above the upper level of the window warm air is trapped. This remains at the same temperature as the wall,  $T_w$ , and the ceiling and walls above the window level do not cool by convection. The coefficient  $C_2$  is about 0.8 in a and 0.4 in b.

**Wall thermal model.** Until opening the window at time  $t=0$ , the walls are considered to be in thermal equilibrium with the building at temperature  $T_b$ . For the heat flux density  $q$ , the wall surface temperature  $T_w$  becomes an explicit function of time. For a wall characterized by a thermal permeability  $\beta = \lambda \rho c$ , a simplified model for  $T_w(t)$  is used (Refs.10,11).

$$T_w(t) - T_w(0) = 2q\sqrt{t/\pi\beta} = R_{dyn} q \quad (4)$$

where the dynamical thermal resistance of the wall  $R_{dyn}$  is zero at  $t=0$ , increasing as the square root of time.

While in the derivation of Equation 4,  $q$  is required to be constant,  $q$  varies in general with time. The complete solution of the diffusion heat equation is an integral over the time history of  $q(t)$ , and the approximation is acceptable when the actual value of  $q(t)$  is close to its time averaged value.

Equation 4 has been used for example, to measure  $\beta$  in situations where the room wall materials are not known, or when the walls are not all made out of the same material. To this effect it is sufficient to heat the closed room with an electrical fan heater and to plot the measured air temperature as a function of time (Ref.11).

**Coupled ventilation and thermal modelling.** After combining Equations 1-4 one solves the final non-linear expression for  $T_{in}$ . A few iterations are sufficient for each value of  $t$ .

There are two new parameters in the model.

(i) the product  $C_2S_i.h$  in Equation 3, which concerns the rapid drop in temperature after  $t=0$ . While the uncertainty in the total wall surface area  $S_i$  can be made small and the error in  $h$  is about 20%, the parameter  $C_2$  can vary by a factor of two or more and when not known dominates in an error analysis.

(ii) the wall parameter  $\beta$  (Equation 4), which concerns the slow decrease in  $T_{in}$ . This parameter can become important when the window stays open for many hours, or when isolating materials are used on the inside and  $\beta$  is small.

To show the order of magnitude of the effects predicted by the model, we have plotted in Figure 4, the inside air temperature drop ( $T_w$  constant), the ventilation rate and the heat loss rate, as a function of outside air temperature and for three values of the product  $C_2S_i$ .

To illustrate the relation between the three figures' parts, we note that e.g. a decrease in  $T_{in}-T_{out}$  of 20%, decreases the ventilation rate by 10% but the energy loss rate by 30%.

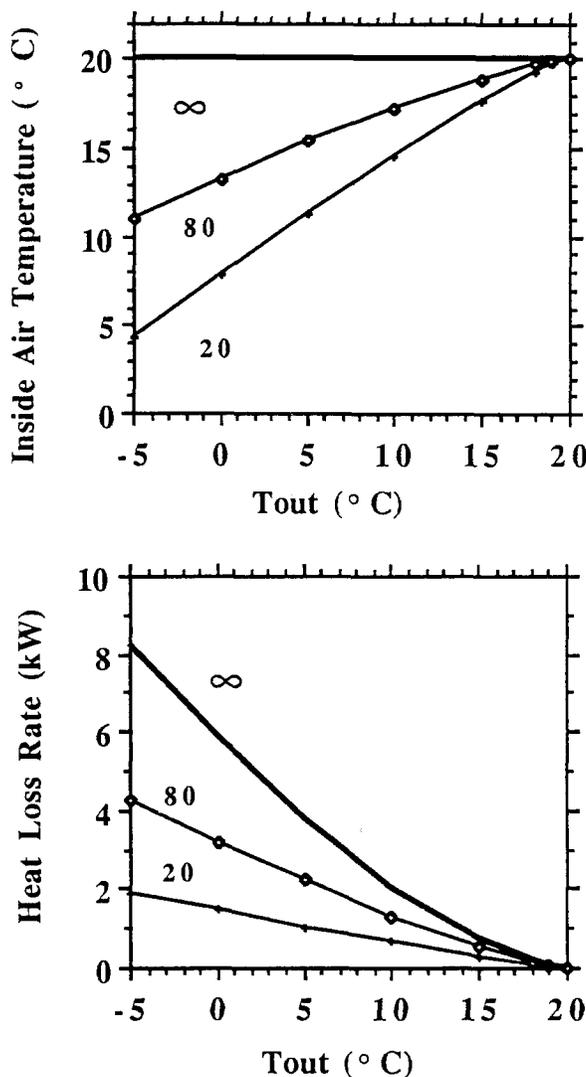


Figure 4: Model predictions for a window 1.5m high, 0.8m wide. The inside air temperature ( $T_w$  constant 20 °C), the ventilation rate and the heat loss rate as a function of outside air temperature, for three values of the heat exchanging wall surface  $C_2S_i$ : 20 m<sup>2</sup>, 80 m<sup>2</sup> and infinitely large.

## Experimental Tests

The model has been validated for a number of real cases. Most measurements have been performed in offices of the LESO laboratory. A typical office (2.8 x 3.4 x 4.2m<sup>3</sup>) is characterised by a high thermal mass made up of concrete walls  $\beta = 10^6$  [W<sup>2</sup> s/m<sup>4</sup>K<sup>2</sup>] of 10cm thickness covering a high thermal isolation. A value of  $C_2 = 0.65$  can be calculated from the position of the window (area 0.9m<sup>2</sup>), while the data (extending over a ten hour period) fit for  $C_2 = 0.5$  (Ref.10).

In Figure 5 we present results on a lightweight structure (Ref.11). The walls of the cabin (equipped with standard office furniture) are well insulated by 5cm polystyrene on the inside covered by metal plating. The experimental value for the (dynamic) wall thermal permeability of  $\beta = 6 \times 10^4$  [W<sup>2</sup> s/m<sup>4</sup>K<sup>2</sup>] was much higher than expected for polystyrene, showing the usefulness of an independent measurement. The inside temperature was measured in the middle of the cabin at half height. For a correct appreciation of the agreement between the measured and calculated value of  $T_{in}$  in Figure 5a, we note that a vertical temperature gradient installs after the opening of the door whose value decreased from about 2 to 1K/m. However there are no free parameters in the model. We also note that the model predicts an initial temperature drop which is too abrupt. Indeed, the heat capacity of the air and the presence of furniture have not been included in the model. This can be corrected by adding a capacity between  $T_w$  and  $T_{in}$  in Figure 2, although this would add a parameter to the model.

## Discussion and Conclusion

The model which is proposed here allows to calculate the ventilation and energy loss rate after opening a single window, and this as a function of room geometry, the thermal wall properties and as a function of opening time. It can be used together with a model describing user behaviour, for more realistic energy calculations and air infiltration modelling.

A simple sensitivity study has shown that the model is stable and not very sensitive to the precise values of the parameters.

More detailed testing is necessary and the model can be improved in various ways, although probably at the expense of additional parameters. For future studies of the effect of wind on single sided ventilation, this model can play an important role by measuring the additional cooling produced by the wind.

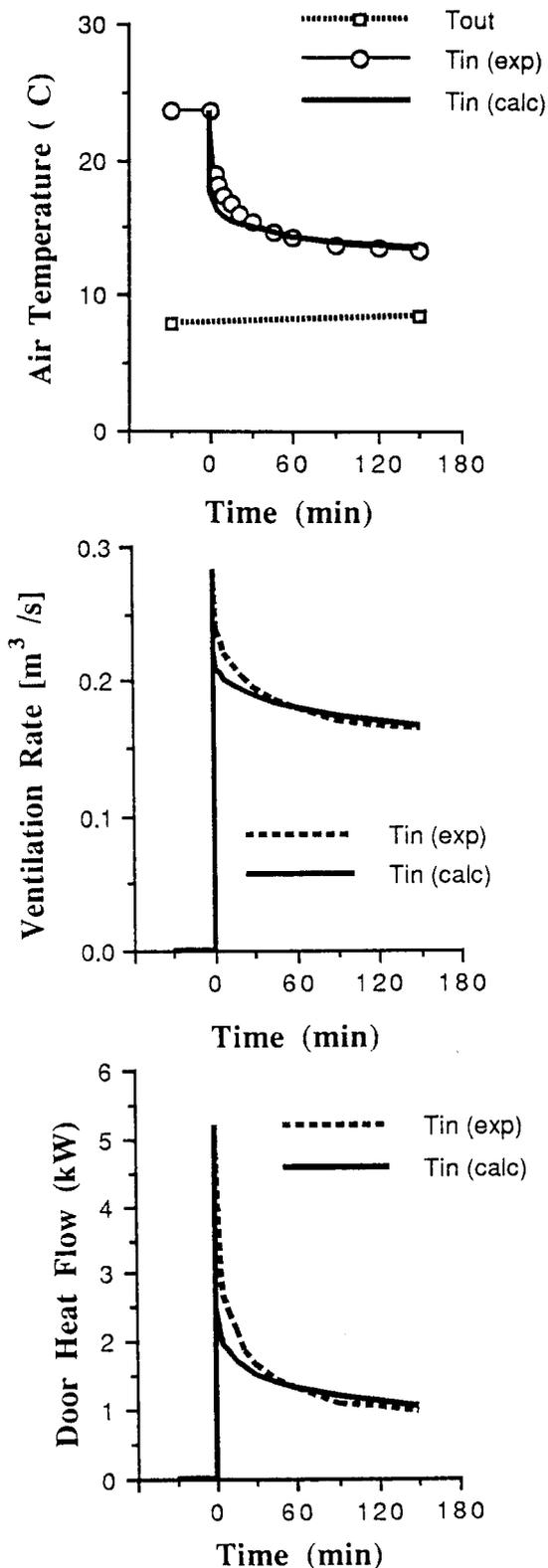


Figure 5: Effect of the opening of the door ( $0.7 \times 2\text{m}^2$ ) of a well insulated cabin ( $2.1 \times 2.6 \times 6.5 \text{m}^3$ ). a) Measured  $T_{in}$  as a function of time compared with the model. The coefficient  $C_2 = 1$  and there are no free parameters. Ventilation b) and heat loss rates c) after opening the door are calculated from the values of  $T_{in}$  in Figure 5a with the help of Equations 1 and 2.

## Acknowledgements

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# Roomvent '90

by James Piggins, AIVC



*Report of a conference held in Oslo, Norway on 13-15 June 1990*

" - to bring together experts both scientists and engineers, working in the field of room air movement" this was the purpose of the second Roomvent'90 conference opened on the 13th June by Prof. Eystein Rodahl, of Trondheim University, Norway, Chairman of the Programme Committee. Veritas, near Oslo provided an attractive setting for the 170 participants, significant increase in numbers from the first conference held in Stockholm, Sweden in 1987.

The programme began with three keynote lectures. The first given by Prof. Bjorn Magnussen, NTH, Norway on "Numerical Mathematical Modelling of Airflow in Ventilated Rooms". This paper emphasised the necessity, usefulness and efficiency of mathematical models combined with experimental and other theoretical investigations. Prof. Magnussen outlined the need to address comfort criteria via modelling, the use of physical sub-models to deal with the separate aspects of the problem and the importance of correct boundary conditions. In particular the correct modelling of physically important processes such as turbulence, which are not included in the solution of the total flow field. The use of the simple  $k-\epsilon$  turbulence model was stated as an example of these procedures. Stating the need for practical results and that any models developed should have a specific useful purpose, Prof. Magnussen felt that the technology is available for simulation of air-flow in rooms with the necessary accuracy both for ventilation and fire and smoke dispersion modelling.

Prof. Magnussen was followed by the second keynote speaker Prof. Ole Fanger, DTH, Denmark who talked about "Comfort criteria related to ventilation spaces". Prof. Fanger described the different factors affecting human comfort and how these are defined. He described how thermal air quality is defined by the predicted mean vote (PMV) a function of the clothing and activity of the occupants and the predicted percentage dissatisfied (PPD) a function of PMV. The sensitivity of people to draught risks (the most common cause of complaint) varies enormously, but the average draught risk can be assessed by temperature and velocity measurements. The third factor affecting comfort is perceived indoor air quality, this is detected by the nose and eyes which are sensitive to a large number of irritants in the air. Prof. Fanger argued that indoor air pollution can be defined using olfs. An olf is defined as the pollution produced by a standard person, although any pollution source may be defined in olfs. He defines perceived air quality in a space in terms of decipols. A space with a pollution source of one olf ventilated by 10

l/s of clean air has a perceived air quality level of one decipol, i.e. 1 decipol = 0.1 olf/(l/s). It is necessary to take into account pollution from sources other than people this includes outdoor (i.e. source) air quality. Should this be unreasonably low the outdoor air may need filtering before use. Lacking any simple instrument to measure air quality, Prof. Fanger suggested the use of people themselves as a first step in defining good/bad perceived air quality.

The third keynote speaker of the conference was Prof. Ove Strindehag, Flakt Evaporator, Sweden, who spoke about "Improved performance of ventilation system components". Prof. Strindehag argued that concern about standards of air quality will mean higher outdoor air flows, filtration of outdoor air and concentrated efforts to reduce pollution sources within buildings. In the best cases where the pollution emission from building materials is at a minimum, the required airflow rates can be determined from the rate of pollutant emission of the occupants themselves, i.e. carbon dioxide and odor levels. However, air flow rates are on the increase in newer systems in order to reduce pollutant levels and improve indoor air quality. This will lead to a demand for higher specification ventilation systems, particularly in terms of energy efficiency and noise levels. New standards may well be set for new components, but Prof. Strindberg argued that correct specification and installation of existing components could solve many of the problems. One solution he recommends is larger capacity fans which will reduce sound problems and power consumption due to the higher efficiency achieved from the motors. Regular maintenance is also vital for good air quality, low noise operation and energy efficiency. It is particularly important to reduce dangerous air pollutants produced by the ventilation systems themselves, such as water droplets from cooling towers, which encourage bacterial and fungal growth in air conditioning systems. It is also important that the ventilation system is a solution to the indoor air quality problem rather than the cause of it. Careful component and total system design will contribute to this.

The remainder of the conference was split into parallel sessions in which a total of 67 papers were presented. These developed the themes presented in the keynote presentations and covered experimental measurements, numerical techniques, field studies and designs.

*Abstracts of relevant papers are published in the current issue of AIVC Recent Advances to AIRBASE. A full copy of the proceedings is available from the publishers: Norsk VVS, P Box 5042 Majorstua 0301 Oslo 3, Norway Telephone: 47 2 60 13 90 Fax: 47 2 69 36 50*

# A Review of Building Airtightness and Ventilation Standards

by Ken Colthorpe

This publication updates AIVC Technical Note 14 of the same title.

Since the first publication of this Technical Note in 1984, which collected together and summarised the standards from some twelve countries dealing with airtightness and ventilation requirements of buildings, there has been a greater emphasis on the need for energy efficiency combined with maintaining air quality. With the relatively high cost of energy still acting as a spur, and the greater use of energy both in the home and in commercial and industrial premises, more stringent measures are being promulgated to enforce the requirements of standards in many countries.

It is recognised that the improvements in the airtightness of buildings can lead to a reduction in the air quality, unless minimum ventilation rates are maintained.

This is also reflected in the specifying of concentration levels of pollutants in the air both inside and outside the building. Levels of carbon dioxide are used in some standards as a basis for the rate of ventilation pres-

cribed and there are now in operation some mechanical ventilation systems that operate and are triggered when the level of carbon dioxide has reached a specified maximum value. Many countries are now employing heat recovery systems which extract heat from the exhaust air from a building and transfer it to the cooler fresh air required for ventilation. Standards on the use of these systems covering the testing of the equipment, and the installation have been in existence for a number of years and are likely to be more extensively specified.

This revised Technical Note brings together the current updated standards from 13 AIVC participating countries and adds the key standards on the subject from one non AIVC country together with those from other International Organisations.

The more pertinent aspects of the standards and regulations are reviewed and a subject analysis is shown in tabular form (see Table 1) covering the airtightness and ventilation requirements of the various countries.

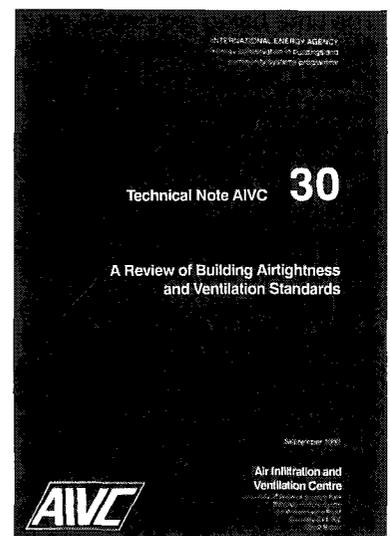
TABLE 1: REQUIREMENTS AND RECOMMENDATIONS FOR AIRTIGHTNESS AND VENTILATION RATES IN SOME COUNTRIES

	Scandinavia				Europe							America		Far East
	Den.	Fin.	Nor.	Swe.	Belg.	Fra.	Ita.	Neth.	Swi.	UK.	FRG.	Can.	USA.	N.Z.
<b>Airtightness:</b>														
Components	W	W	R	N	W	-	W	W	W	W	W	W+D	W+D	W
Whole Buildings	N	N	R	R	N	-	R	N <sup>1</sup>	R	N	N	N <sup>1</sup>	N <sup>1</sup>	N
<b>Minimum Ventilation Rates:</b>														
Dwellings	R	R	R	R	R <sup>3</sup>	R	R	R	N <sup>2</sup>	R	R	R	R	N <sup>2</sup>
Other (Industrial/Commercial)	R	R	R	R	-	-	R	R	R <sup>5</sup>	R	R	R	R	N <sup>4</sup>
<b>Key:</b>	R = Recommendation exists N = No recommendation exists W = Recommendation for windows only W+D = Recommendation for doors and windows only											1 Draft standard in preparation 2 Recommendations exist for internal kitchens, bathrooms, toilets. 3 A voluntary standard that may soon be replaced. 4 Government legislation exists for bathrooms, toilets and laundries. 5 Only for some types of rooms.		

Country Abbreviations:

Den:Denmark; Fin:Finland; Nor.Norway; Swe.Sweden; Belg.Belgium; Fra.France; Ita.Italy; Neth.Netherlands; Switz.Switzerland; UK.United Kingdom; FRG.West Germany; Can.Canada; USA.United States of America; NZ.New Zealand.

Table 1: Requirements and recommendations for airtightness and ventilation rates in some countries



# AIVC's Fifth Worldwide Survey of Current Research into Air Infiltration, Ventilation and Indoor Air Quality

by Mark Limb, AIVC Scientist

The Air Infiltration and Ventilation Centre's worldwide survey of current research into air infiltration and related topics provides organisations in participating countries with regularly updated information about on-going research in this field. In particular the major objectives of this survey are to encourage the international cross fertilization of research ideas and to promote co-operation between research organisations in different countries.

The Air Infiltration and Ventilation Centre's fifth worldwide survey of current research has attracted 233 replies, from a total of 23 different countries. All of the AIVC participating countries are represented in the survey. Response from the non participating countries was greater than in the 1986 survey, with France and Japan providing the majority of the replies. The distribution of replies can be seen in Figure 1. It can be seen that nearly half the replies received were from three countries; the United States of America (35 replies), Canada (30 replies), and the United Kingdom (39 replies).

The time being expended on individual projects is in the region of 1 to 3 person years. However there are some notable exceptions, with 15 long term projects of between 10 000 to 20 000 hours, and 6 projects over 20 000 person hours. These projects tend to involve research into more than one subject, and include projects from New Zealand, Germany and Belgium.

The overall picture is that there are an estimated one million hours of research effort being documented by this survey.

In terms of the total number of replies received and the subjects covered, this survey represents the most comprehensive review of current research yet published by the AIVC. The project summaries from the 23 countries essentially cover all aspects of air infiltration into indoor air quality research. It is interesting to note that research into indoor air quality has increased, as too has research into simulating airflow movement.

Research on ventilation and heating systems and strategies which was deficient in the 1983 survey, has continued to increase into the 1990's as alternative en-

Origin and Distribution of Survey Replies

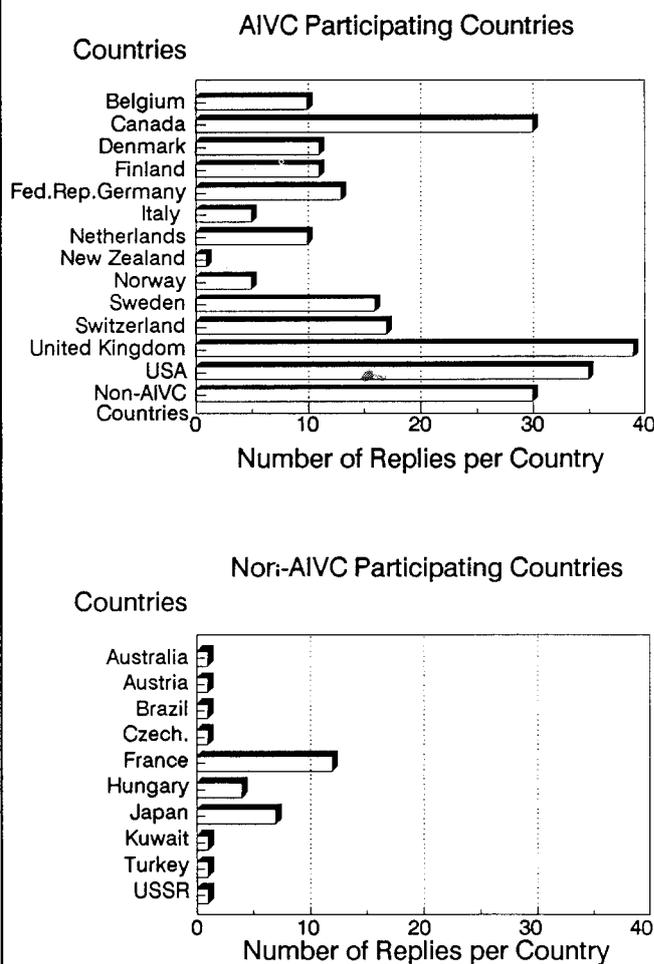


Figure 1

ergy sources and a renewed drive towards energy conservation and efficiency has provided a greater impetus to improve the way we use energy.

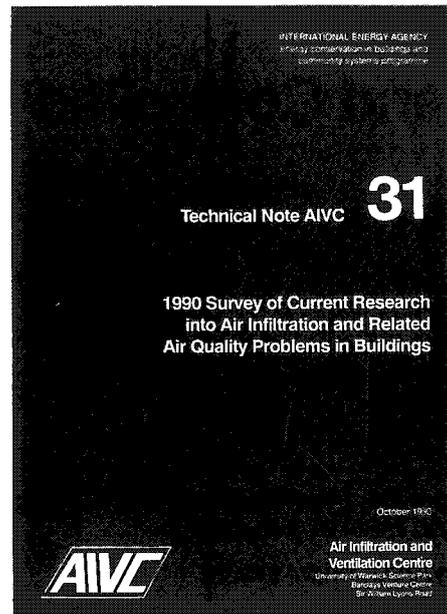
Simulation work on multizone indoor airflows has continued to increase since the fourth AIVC survey in 1986,

but so too have airflow models in general, air quality models and models attempting to predict occupant behaviour.

The use of passive tracer gas techniques initially identified in the last survey, has also increased. These passive techniques allow air change rates to be unobtrusively monitored in occupied buildings.

Indoor air quality research has also increased. Of specific interest have been the causes and sources of indoor air pollution, the effect of indoor climate on occupant health and comfort, and the effect of airtightness measures and minimum ventilation rates on indoor air quality. The overall aim of such projects is the development of energy efficient buildings, which also have low levels of indoor air pollution.

The survey of research will soon be available either as an AIVC Technical Note, or as a parallel database running alongside AIVC's bibliographic database AIR-BASE, which is currently available as a PC version directly from the AIVC.



### Review

## Measuring Ventilation using Tracer Gases

by Peter W Grieve M.Sc.

*Brüel & Kjær, October 1989*

Obtainable from:

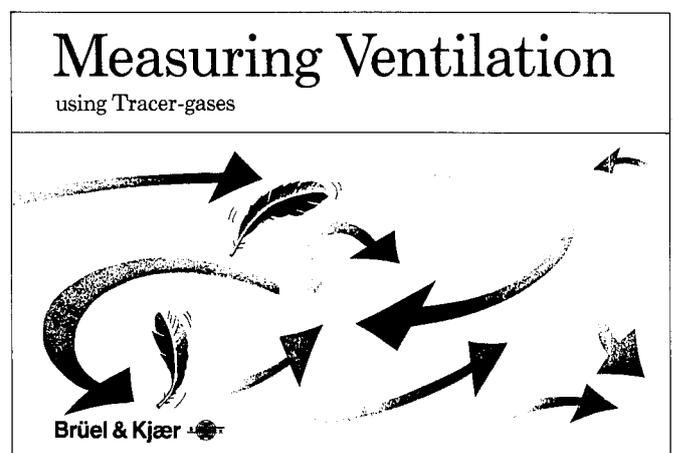
Brüel & Kjær World Headquarters,  
Nærum,  
DK-2850,  
Denmark,

Telephone: +45 42 80 05 00

Fax: +45 42 80 14 05

A useful booklet on ventilation is now available free from Brüel & Kjær, the Danish instrument maker. The booklet answers some basic questions about measuring ventilation using tracer gas. It explains some of the techniques and terminology used and gives some application examples.

The booklet is fully illustrated in a clear, straightforward style. While it makes ventilation measurements understandable to the novice, it is adequately supported with calculation methods and equations to provide access to ventilation measurement technology.



Review

# Demand Controlled Ventilating System: State of the Art Review

Edited by Willigert Raatschen

IEA Energy Conservation in Buildings and Community Systems Programme, Annex 18,  
February 1990

Sweden, Stockholm, Swedish Council for Building Research, D9:1990 ISBN 91-540-5169-X

Available from AIVC, Price: £12.00

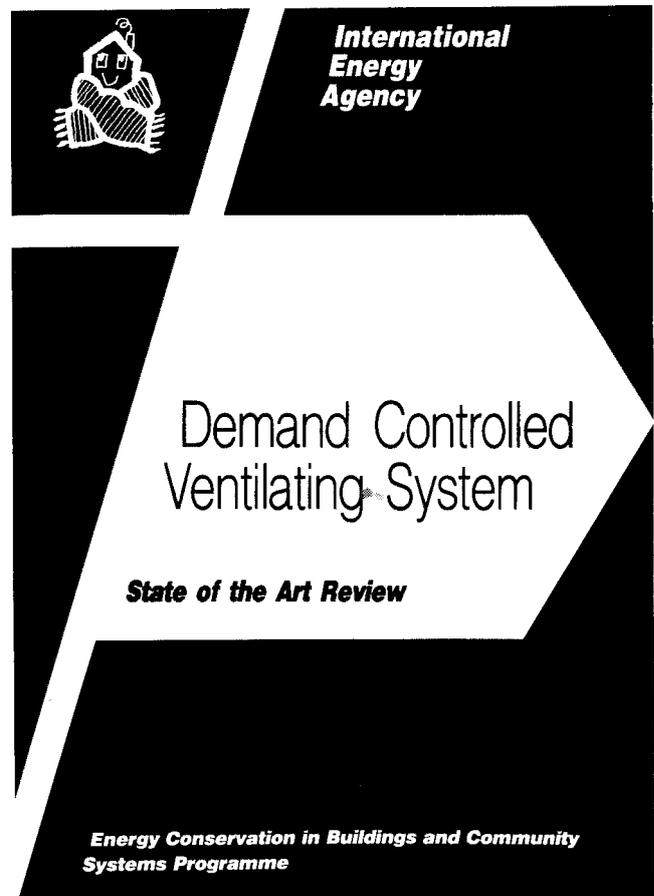
or from Svensk Byggtjänst, S-171 88 Solna, Sweden

The 124 pages of this publication give a state of the art review of demand controlled ventilating systems. It is the result of a collaboration between ten International Energy Agency countries. Information on air quality control by means of the ventilating system has been gathered and weighed. The report gives examples of the use of air quality demand controlled ventilating systems by different users in different types of domestic, office, and school buildings.

As a background the international standards for a few pollutants in the indoor environment are listed, together with some examples of measured pollutant levels in various building types.

An overview of the sensor market explains the function principle of humidity, carbon dioxide and mixed-gas sensors. Summaries of more than thirty projects over the last ten years were reviewed with respect to air quality demand controlled ventilating systems.

Conclusions are given about suitable pollutants to govern the system, the capability of sensors and other influencing factors saving energy without sacrificing the indoor air quality. It is also a guide to the future test work of the ten countries in Annex 18: Demand Controlled Ventilating Systems.



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**Air Infiltration Review.** Quarterly newsletter containing topical and informative articles on air infiltration research and application.

**Recent Additions to AIRBASE.** Quarterly bulletin of abstracts added to AIRBASE, AIVC's bibliographic database.

## GUIDES AND HANDBOOKS

Applications Guide 1 (1986) Liddament, M.W. 'Air Infiltration Calculation Techniques - An Applications Guide'

Applications Guide 2 (1988) Charlesworth, P.S. 'Air Exchange Rate and Airtightness Measurement Techniques - An Application Guide'

Handbook (1983) Elmroth, A. Levin, P. 'Air infiltration control in housing. A guide to international practice.'

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TN 10 (1983) Liddament, M., Thompson, C. 'Techniques and instrumentation for the measurement of air infiltration in buildings - a brief review and annotated bibliography'

TN 11 (1983) Liddament, M., Allen, C. 'The validation and comparison of mathematical models of air infiltration'

TN 13 (1984) Allen, C. 'Wind pressure data requirements for air infiltration calculations'

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TN 20 (1987) 'Airborne moisture transfer: New Zealand workshop proceedings and bibliographic review'

TN 21 (1987) Liddament, M.W. 'A review and bibliography of ventilation effectiveness - definitions, measurement, design and calculation'

TN 23 (1988) Dubrul, C. 'Inhabitants' behaviour with regard to ventilation.'

TN 24 (1988) 'AIVC Measurement Techniques Workshop: Proceedings and Bibliography'

TN 25 (1989) Blacknell, J. 'A subject analysis of the AIVC's bibliographic database - AIRBASE'

TN 26 (1989) Haberda, F and Trepte, L. IEA Annex IX 'Minimum ventilation rates and measures for controlling indoor air quality.'

TN 27 (1990) Bassett, M. 'Infiltration and leakage paths in single family houses. A multizone infiltration case study.'

TN 28 (1990) Sutcliffe, H. 'A guide to air change efficiency.'

TN 29 (1990) Feustel, H E, et al 'Fundamentals of the multizone air flow model - COMIS.'

TN30 (1990) Colthorpe, K 'A review of building airtightness and ventilation standards.'

TN31 (1990) Limb, M 'AIVC's fifth worldwide survey of current research into air infiltration, ventilation and indoor air quality.'

DCV (1990) Raatschen, W (ed.) Demand controlled ventilating system: state of the art review.' Annex 18 report.

## AIVC CONFERENCE PROCEEDINGS

1st 'Instrumentation and measuring techniques', Windsor, UK, 1980.

2nd 'Building design for minimum air infiltration', Stockholm, Sweden, 1981.

3rd 'Energy efficient domestic ventilation systems for achieving acceptable indoor air quality', London, UK, 1982.

4th 'Air infiltration reduction in existing buildings', Elm, Switzerland, 1982.

5th 'The implementation and effectiveness of air infiltration standards in buildings', Reno, USA, 1984.

6th 'Ventilation strategies and measurement techniques', Het Meerdal Centre, Netherlands, 1985.

7th 'Occupant interaction with ventilation systems' Stratford-upon-Avon, UK, 1986.

8th 'Ventilation technology - research and application', Uberlingen, West Germany, 1987.

9th 'Effective Ventilation' Ghent, Belgium, 1988

10th 'Progress and trends in air infiltration and ventilation research' Espoo, Finland, 1989

mf Proceedings of AIVC conferences numbers 1-10 are also available in microfiche form.

## LITERATURE LISTS

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- 1) Pressurisation - infiltration correlation: 1. Models.
- 2) Pressurisation - infiltration correlation: 2. Measurements.
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- 5) Domestic air-to-air heat exchangers.
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