10th AIVC Conference

Progress and trends in air infiltration and ventilation research

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Proceedings

Volume 2

CONTENTS (i)

Volume 2

Preface (v)

Posters:

P1. Accuracy and Development of Tracer-Gas Measurement Equipment.
B Kvisgaard and P F Collet. 1

P2. A Comparison Between the Step-up, Step-down and Pulse Injection Techniques for the Measurements of the Mean Age of Air.
R Niemela, A Saamanen and H Koskela. 17

P3. Development of a Microprocessor-Controlled Tracer Gas System and Measurement of Ventilation in a Scale Model.
S B Riffat. 31

P4. Methodologies for the Evaluation of Ventilation Rates by Tracer Gas Comparison.
I Meroni, C Pollastro, P Tirloni and W Esposti. 49

P5. A Comparison of Different Methods of Calculating Interzonal Airflows by Multiple Tracer Gas Decay Tests.
C Irwin and R E Edwards. 57

P6. A Modern Concept for Office Buildings: Energy Saving and Good Indoor Climate are no longer Contradictory.
W Braun. 71

P7. Air Change in Flats with Natural Ventilation: Measurements and Calculations.
A Olsson-Jonsson. 85

P8. Outdoor Air Inlet without Draught Problems.
G Werner. 101

P9. Energy Use for Transport of Ventilation Air.
P Wickman. 113

P10. Comparison of Air Infiltration Rate and Air Leakage Tests under Reductive Sealing for an Industrial Building.
P J Jones and G Powell. 131

P11. General Feature of a Two-Dimensional Isothermal Mean Flow inside a Ventilated Room with a Wall Mounted Obstacle.
P Berlandier, R Rapp, J C Sereys and J C Curin. 153

P12. Buoyancy-driven Air Flows in a Closed Half-scale Stairwell Model: Velocity and Temperature Measurements.
A S Zohrabian, M R Mokhtarzadeh-Dehghan and A J Reynolds. 167

P13. The Simulation Model of Industrial Conditioning Systems.
L Heikkenen. 189

P14. VAV-Duct Systems - Simulating.
P A Rantanen. 203

P15. Identification Methods for Multiple Cell Systems.
B Hedin. 209

P16. The COMIS Infiltration Model.
H E Feustel, M Grasso, J C Phaff, F Allard, M Herrlin, Y Utsumi, V B Dorer, L Mingsheng and H Yoshino. 233

P17. Air Infiltration in Canadian Homes - A Decade of Change.
M Riley and M Lawton. 253

P18. Ventilation, Air Flows in Buildings and Indoor Air Quality - RDSD and Developments in Germany.
L Trepte. 269

P19. An Overview of Infiltration and Ventilation Developments in France.
M Jardinier. 277

P20. Automated Tracer Equipment for Air Flow Studies in Buildings.
M R Bassett and H M Beckert. 293

P21. Infiltration and Ventilation Developments in Norway.
J T Brunsell and E Skaret. 311

P22. Air Infiltration and Ventilation. Progress and Trends in Sweden.
J Kronvall. 333

P23. Infiltration and Ventilation in Switzerland- Past and Future.
P J Hartmann. 345

P24. New Design of Central Units in Air Heating Systems for Heating and Ventilation in Domestic Buildings.
F Steimle and T Glimpel. 369

P25. Airtightness of Swedish Residences.
M D Lyberg and C A Boman. 383

P26. Demand Controlled Air Ductwork.
J Laine. 397

P27. Air Changes and Scatter in Mechanical Ventilation Rates in Swedish Residences.
M D Lyberg and C A Boman. 413

P28. Public Policy Considerations and the Development of a Code for the Control of Radon in Residences.
M Nuess and S Price. 419

International Energy Agency

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

Energy Conservation in Buildings and Community Systems

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

The Executive Committee

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

I Load Energy Determination of Buildings * II Ekistics and Advanced Community Energy Systems * III Energy Conservation in Residential Buildings *
IV Glasgow Commercial Building Monitoring * V Air Infiltration and Ventilation Centre
VI Energy Systems and Design of Communities * VII Local Government Energy Planning * VIII Inhabitant Behaviour with Regard to Ventilation *
IX Minimum Ventilation Rates * X Building HVAC Systems Simulation
XI Energy Auditing * XII Windows and Fenestration * XIII Energy Management in Hospitals * XIV Condensation XV Energy Efficiency in Schools
XVI BEMS - 1: Energy Management Procedures
XVII BEMS - 2: Evaluation and Emulation Techniques
XVIII Demand Controlled Ventilating Systems XIX Low Slope Roof Systems
XX Air Flow Patterns within Buildings
XXI Energy Efficient Communities XXII Thermal Modelling

Annex V Air Infiltration and Ventilation Centre

The IEA Executive Committee (Building and Community Systems) has highlighted areas where the level of knowledge is unsatisfactory and there was unanimous agreement that infiltration was the area about which least was known. An infiltration group was formed drawing experts from most progressive countries, their long term aim to encourage joint international research and increase the world pool of knowledge on infiltration and ventilation. Much valuable but sporadic and uncoordinated research was already taking place and after some initial groundwork the experts group recommended to their executive the formation of an Air Infiltration and Ventilation Centre. This recommendation was accepted and proposals for its establishment were invited internationally.

The aims of the Centre are the standardisation of techniques, the validation of models, the catalogue and transfer of information, and the encouragement of research. It is intended to be a review body for current world research, to ensure full dissemination of this research and based on a knowledge of work already done to give direction and firm basis for future research in the Participating Countries.

The Participants in this task are Belgium, Canada, Denmark, Federal Republic of Germany, Finland, Italy, Netherlands, New Zealand, Norway, Sweden, Switzerland, United Kingdom and the United States of America.

PROGRESS AND TRENDS IN AIR INFILTRATION AND VENTILATION RESEARCH

10th AIVC Conference, Dipoli, Finland
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Poster 1

Accuracy and Development of Tracer-Gas Measurement Equipment

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ABSTRACT

1979 a project was launched at Technological Institute, Copenhagen with the purpose of developing a method for continuous measurement of air change rates in occupied dwellings. Today - 10 years later - we can introduce the first generation of mass-produced measuring equipment performing measurements of air change rates employing the method of constant concentration of tracer gas. The principles used in the first model, which was introduced 1981, are largely identical to those used in the latest model. However, components and programmes have been changed several times. Furthermore, through the years new programmes that expand the capability of the measuring equipment have been developed.

The paper will discuss the development that has taken place over the 10 years, which problems have caused the biggest trouble and how they were solved. Also, the types of measurement performed with the equipment will be touched upon, and we shall take a closer look at a couple of special measurements. Finally, the accuracy of the equipment as well as the cost of reaching today's level of development will be discussed.

1. INTRODUCTION

The energy crisis in 1973 dramatically increased the interest in measuring the air change rate of dwellings. All buildings should be made as tight as possible in order to reduce the energy consumption for heating the infiltration air; and air change rate measurements were used to determine the effect of different methods of making buildings tight.

In Denmark the successful campaign to make buildings tight and to reduce airing had a serious "side-effect": Our lacking knowledge about the air change rates in buildings had made us go too far in our efforts to save energy, and we were faced with an indoor air quality problem in many buildings.

To build up the necessary knowledge about air change rates in buildings a project was launched at Technological Institute with the purpose of developing a method for continuous measurement of air change rates of occupied dwellings, and of determining the air change in various dwellings. Equipment to measure air change rates employing the the method of constant concentration of tracer gas was developed and in the autumn of 1981 the measuring equipment was ready for its first appearance in the field.

2. THREE GENERATIONS OF TRACER-GAS MEASUREMENT EQUIPMENT

2.1 The first generation

The first Danish equipment ever for continuous measurement of air change rates was being constructed at the Technological Institute in Copenhagen in 1979 on a grant from the Minstry of Trade. The project had the following impressive title: "Development of a method for continuous measurement of the natural air change, the relative humidity and temperature levels etc. in the tightened part of occupied dwellings, as well as single measurements of the size of the natural air change in dwellings with air-to-air heat exchangers".

The international cooperation that had been established through AIC (Air Infiltration Centre) greatly influenced the project right from the beginning. At AIC-congresses measurement methods could be discussed and fruitful contacts established. Especially our cooperation with Professor David T. Harrje of Princeton University was very productive. As a result of this cooperation one of his students, Timothy McNally, worked on our project for a year as a Fullbright student.

It only took 2 years to build the equipment; impressive, when considering that fact that it was many months before the ambitions had been reduced to a feasible level. Figure 1 shows the first generation measuring equipment ready for the first measurements.

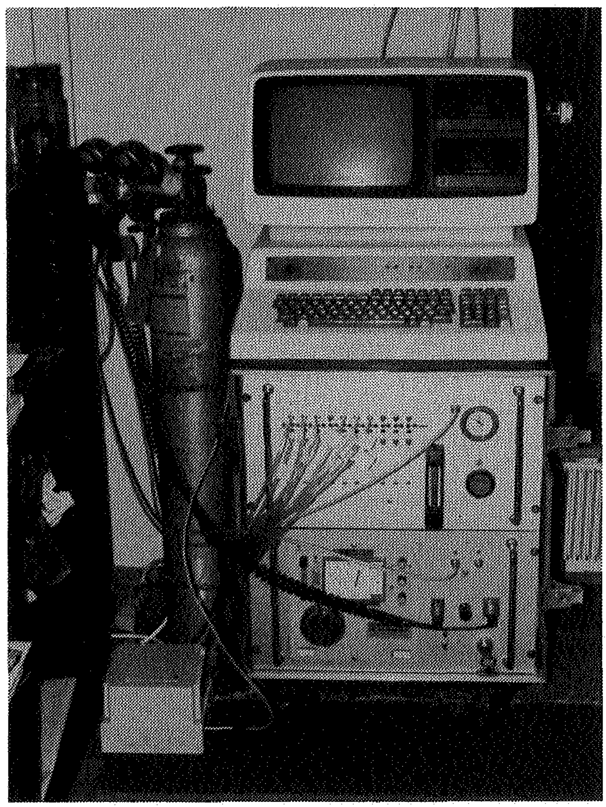

Fig. 1: First generation of tracer-gas measuring equipment

The equipment was designed to perform measurements in buildings with up to 10 rooms. The concentration in the rooms was kept at 50 ppm nitrous oxide, and the air change rate was calculated from the quantity of nitrous oxide necessary to dose to maintain the concentration. The tracer gas was dosed through 10 solenoid valves, and through another 10 solenoid valves air samples were collected from the 10 rooms.

With measurements in the field the equipment evolved from the development phase to the maintenance phase, and this phase turned out to be tremendous. It is a well known fact that a chain is not stronger than its weakest link, and this chain consisted of quite a number of weak links, ranging from the tailor made sampling and dosing equipment, and purchased gas analyzers and microcomputers, to the home-made computer programme.

To keep down expenses for the tracer gas and to avoid the uncertainty of the gas mixture concentrations we chose to dose with pure nitrous oxice. However, this requires a lot of the tightness of the dosing system all the way from the gas cylinder to the dosing points in the rooms. Most of the leaks in the dosing system were found at the tracer-gas cylinder or the pressure reduction valve on the tracer-gas cylinder. Even though, at each start-up, the equipment was tested for leaks with soap water we did not always find all leaks; therefore, we introduced a very efficient check-up of leaks and other "illnesses" in the dosing system which rescued many a measurement. The check-up method was to keep track of the consumption of tracer gas. Before and after each measurement the tracer-gas cylinder was weighed, and the gas consumption thus ascertained was compared with the total amount of doses emitted by the measuring equipment during the measurement.

The sampling system must also be tight. This was one of the biggest problems in the first generation measuring equipment. At a concentration of around 50 ppm nitrous oxide where the equipment was placed and in the rooms to be measured, a little false air had no significant effect; but if the equipment was placed outside the measuring area the error would be considerable.

At first, software errors were predominantly due to the fact that the dosing algorithm was adjusted so as to keep a constant concentration in the rooms, also when the air change rates varied considerably due to opening of windows etc. Later, the errors were more self-inflicted, introduced by frequent changes in the programme. Even small, apparently innocent, changes had drastic effects. Once, a small change in the programme caused us to lose an entire week's measurements. The measurement worked according to schedule, but at the end of the week no measuring data at all had been stored on the diskette.

Errors in the dosing system, the sampling system and the software may just be accepted - after all, it was a prototype. Errors in mass-produced equipment, however, is harder to accept; and the purchased parts for the tracer-gas equipment were not any more reliable than the parts produced by ourselves.

Quite involuntarily the equipment had to be operated by 2 persons. During the first measurement we found out that the box containing the entire system was too heavy and unhandy for one man to get in and out of vans and houses. This did not matter much, however, as 2 persons were necessary for installing the tubing as well as testing and starting up the equipment in one day.

Having been used for 18 months the first generation equipment was ready for a reconstruction. The immediate reason was a recommendation from the Health and Safety Executive that we do not use nitrous oxide as a tracer gas. Furthermore, we had encountered mechanical problems with the tightness of the sampling system, and fouling of the air lines in the system. Quite simply, the sampling system had been clogged up by dust, because we had "forgotten" to equip all sampling tubes with filters.

2.2 The second generation

Figure 2 shows the equipment after reconstruction; as a matter of fact it was a completely new system, as only few components from the first equipment were used again.

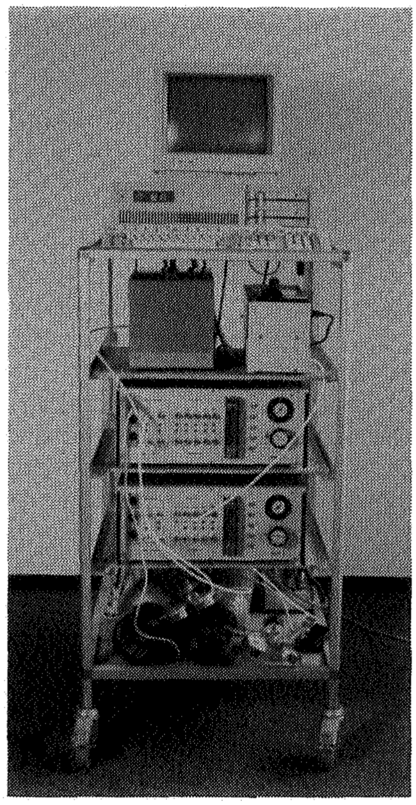

Fig. 2 Second generation of tracer-gas measuring equipment

The most substantial improvements in the second generation equipment were:

- Grouping of the equipment into units that could be carried by one person.
- Replacement of gas analyzer so as to be able to measure the tracer gas SF_6.
- New sampling system that was tight and equipped with filters on the inlet.
- Grouping of the dosing system in units of five dosing points each, thus prepared for measurements with two tracer gases.

With minor modifications this equipment has been used since 1983, and another four systems of this type have been built in the years until 1987. Three systems are at the Technological Institute, one is in Belgium, and one in Norway. The second generation equipment has generally worked satisfactorily, but the maintenance costs have been heavy. Especially, the costs for calibration of gas analyzers, pressure transducers, dosing nozzles, and temperature sensors. Again, the problem was too poor quality of the purchased components. In particular, it was not worthwhile trying to economize on the electronics (computers, A/D converters, etc.).

2.3 The third generation

1985 saw the beginning of the third generation equipment. This time the purpose was to produce a commercial product, reliable, handy, and easy to operate. The third generation is being produced by Brüel & Kjær, and uses the same principles as the second generation. Figure 3 shows the second and third generations.

The most substantial improvements from the second to the third generation are:

- The equipment is considerably smaller and less heavy.
- The communication between computer, gas analyzer, and dosing and sampling units has been fully digitalized, i.e. A/D converters in the computer can be avoided, and, consequently, small lap top computers can be used.
- The Brüel & Kjær gas monitor is stable over a long period of time; consequently, automatic recalibration of the monitor during measuring can be omitted.

- Supplementary air has been introduced on all dosing tubes, to dilute the tracer gas and to transport it faster to the dosing point.
- Automatic calibration of dosing nozzles has been introduced.
- An independent CPU in the dosing and sampling unit has made it possible to continuously monitor the dosing flow 10 times/second. This reduces the requirements for pressure reduction valves on the tracer-gas cylinders.
- More flexible and user-friendly software. For example, routines for calculating the age of the air have been introduced, and measurements with two tracer gases are now possible. Figure 4 shows an example of a screen picture.

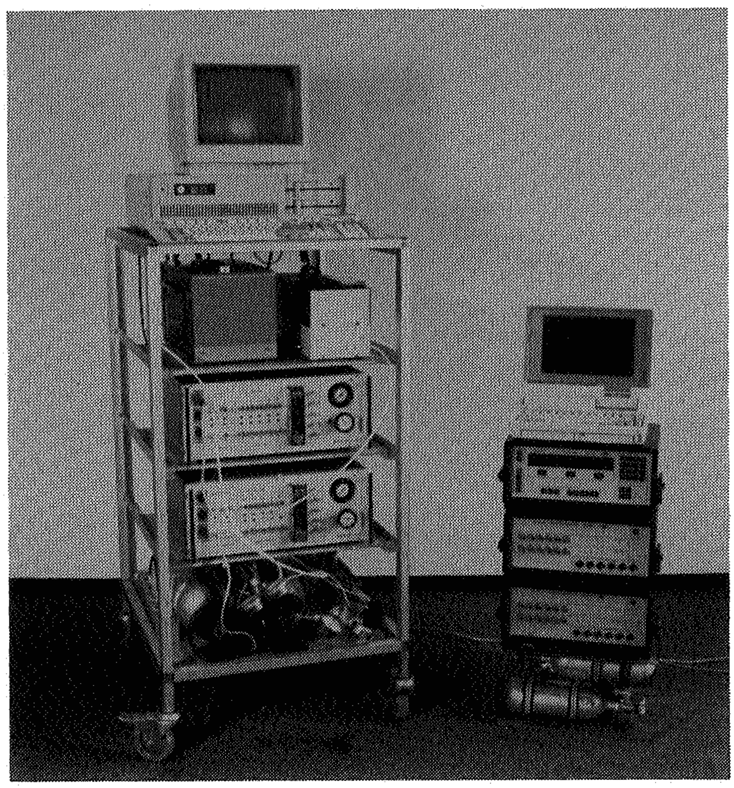

Fig. 3 The second and third generation of tracer-gas measuring equipment next to each other

In addition to these improvements the reliability of the equipment has been accentuated. The dosing system, for instance, is assembled in a clean room to avoid particles in the air lines that may clog up the dosing nozzles. Furthermore, semiconductor sensors for temperature measurements have been replaced by Pt100 sensors, and the number of cables for connection of the system has been reduced from seven to three.

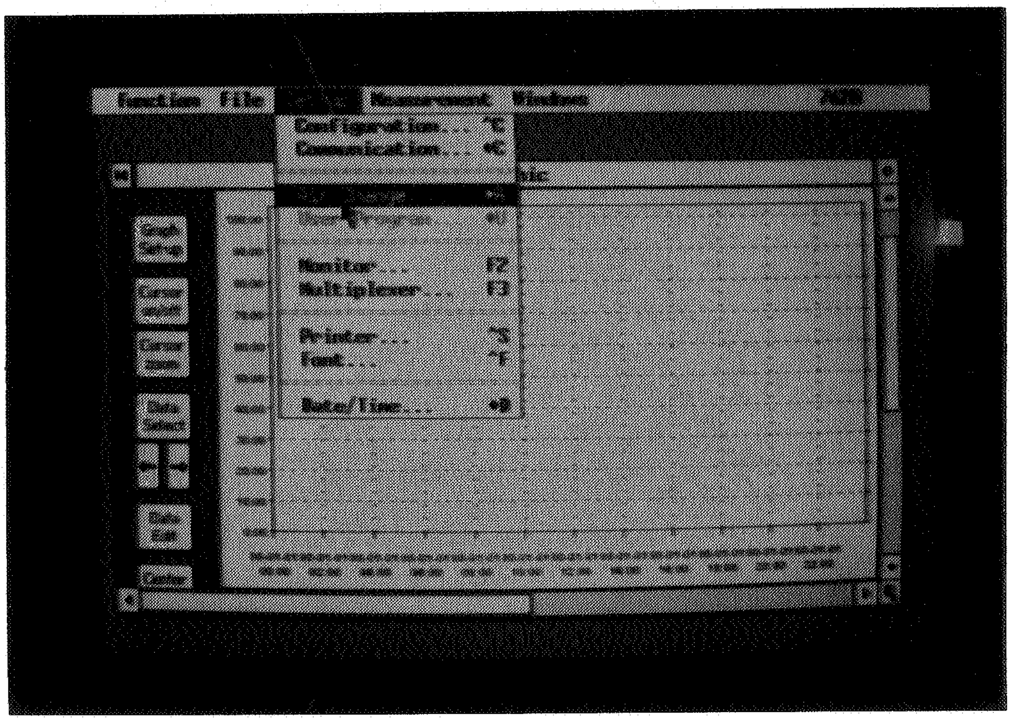

Fig. 4 Screen picture from the software that controls the tracer-gas measurements

As a new feature, to reduce the number of futile measurements, status flags have been introduced. For every measurement from the gas monitor there will be a number of flags giving information on its function. For example, a flag indicates whether the infrared source in the gas monitor is functioning correctly, whether the chopper frequency is correct, and whether the measurement is unreliable due to vibration noise. Likewise, the dosing and sampling unit indicates whether the suction tubing is clogged up, or whether the pumps are yielding sufficient pressure.

In addition to these measures, Brüel & Kjær's standard product test should also have improved the reliability of the equipment. Among other things, this standard includes testing at high and low temperatures, test for electrical shock, and a bump test, where the equipment is dropped 1000 times onto a desk from 25 mm height.

Of the new equipment, so far only the gas measurement principle has been tested for a longer period of time, as 2 of Brüel & Kjær's analyzers have for the past year been used together with the second generation tracer-gas measuring equipment for measuring with 2 tracer gases.The purpose of the measurements is to determine the air flow between the crawl space and the living rooms in one-family houses.

This third generation measuring equipment, shown on the below figure, comprises the sum of 10 years' experience in tracer-gas measurements in Denmark, and should result in making tracer-gas measurements a tool for solving problems, instead of being an art in itself.

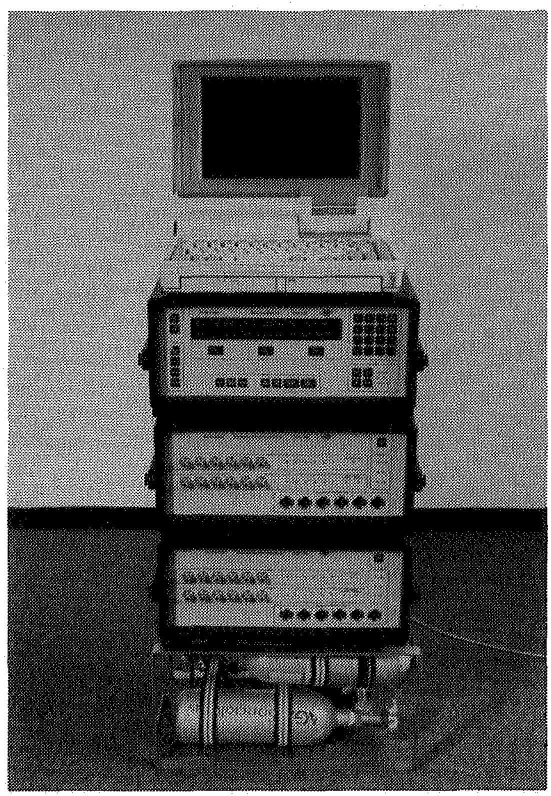

Fig. 5 Third generation of measuring equipment

3. MEASUREMENTS

Since 1981 approx. 75 air change measurements with tracer-gas measuring equipment have been performed at the Technological Institute, of which 41 have been performed in dwellings, 22 in schools and child care institutions, 5 in offices, and a single one in an industrial company. The equipment has been in operation for a total of approx. 430 days.

We would like to specially point out the largest and most difficult of those measurements - an air change measurement at a small brewery. Setting up the equipment alone took several days, as a total of 1500 m of tubing had to be installed, with 250 m as the longest stretch. The volume of the brewery was 100,000 m^3, and the method used was the constant concentration measurement with a target concentration of 300 ppb. Even though we had forced down the monitor, so that we measured with errors of approx. 20%, we did use 80 kg tracer gas for a week's measurements. The purpose of the measurement was to find out why the beer bottles in a store room in the centre of the building very quickly were covered with dust which made them look old. The first day's measurements showed that the store room was very well ventilated with air from other parts of the brewery and with infiltration air. Only about 10% of the air change measured in the store room came from the ventilation system for the store room. Furthermore, it was found that the outdoor air contained the same amount of dust as the air in the store room and had the same grain mixture, and thus the dust on the bottles was caused by too big air change in the store room.

A rather curious incident occured: We almost got the entire brewery closed down, because the local safety officer found that it was highly dangerous to be near our mixing fans.

Another interesting measurement was performed in a low-energy office building. All offices were equipped with a hinged window, a fan, and an electric heater. When the fan was turned on, the heater was turned off, so it was almost impossible to waste energy. The system was not completely reliable, though, as opening the window did not turn off the heat. Everybody in the office section expressed their satisfaction with the indoor climate, even though we measured an air change rate down at 0.1 times/hour. As a matter of fact, we even got as low as 0.05 times/hour during the night.

Another curious incident occurred here. Management found it peculiar that there was a peak in the air change rate around three o'clock in the afternoon. They feared that the increase in the air change rate was due to the employees leaving at that time instead of at four o'clock as according to agreement. However, a printout of the air change rates for all offices revealed the one office only caused the increase in air change. It turned out that the window in that office was often open at 3 o'clock because the employees in the office section had their coffee break there.

4. ACCURACY

Tracer-gas measuring methods are very accurate. When used in the laboratory they show good conformity with recognized reference methods for measuring air flow. When used in air change measurements with the constant concentration of tracer gas method the accuracy normally is better than 5%. Whether these accuracies also apply to field measurements, depends on the stability of the measuring equipment as well as on which additional sources of error to include in the field measurements. Based on an analysis of measuring errors in the case of flow measurements in ducts and in the case of air change measurements in rooms, we shall below estimate the error of tracer-gas measurements. Measuring flow in ducts will in this connection be considered the basis of the error analysis, as it has the best defined sources of error.

The accuracy of measuring methods cannot be discussed unless the quantity to be measured has been clearly defined. Is it m^3/h dry air at 20^oC and 1 atm, or is it m^3/h at actual pressure, temperature and humidity? We have chosen to measure m^3/h at 20^oC, 1 atm, and actual humidity.

Below is a list of the sources of error we could think of for our third generation measuring equipment:

Gas concentration measurement of the tracer gas SF6:

Zero point errors		5 ppb
Range error		10 o/oo
Zero drift (over 1 month)		5 ppb
Range drift (over 1 month)		15 o/oo
Zero drift due to temperature (18-26^oC)		2 ppb
Range drift due to temperature (18-26^oC)		12 o/oo
Zero drift due to pressure (970-1050 mbar)		2 ppb
Range drift due to pressure (970-1050 mbar)		4 o/oo
Interference from water vapour (0-15^oC Tdew)	0 ppb	
Interference from CO_2 (350-2000 ppm)	0 ppb	
Errors on calibration gas (14.8 ppm)		20 o/oo

The error for gas measurements is 8 ppb on Zero point and 30 o/oo on the measured value, when calibrating the monitor each month.

Dosing of tracer gas:

The tracer gas is dosed through a nozzle at the speed of sound. Therefore, the dosing is only depending on pressure and temperature of the gas upstream the nozzle. The error for the dosed gas quantity is dependant on the errors for these two measurements, the errors on measuring nozzle constants and errors on registrating dosing time. The nozzle constants are automatically measured in the dosing unit by a special calibration routine.

Error when establishing nozzle constant, approx.	18 o/oo
Error from variation in pressure measurement, approx.	5 o/oo
Error from variation in temperature measurements, approx.	5 o/oo
Error for dosing time, approx.	5 o/oo
Error due to tracer-gas impurities	10 o/oo
Leakage in dosing unit per channel	0.24 ml/h

The error for dosed quantity is 0.24 ml/h + 22 o/oo when calibrating the dosing unit each month.

Contamination of the sample during transport:

For our equipment the greatest risk of contamination of the sample is, when it is inside the dosing and sampling unit.

Error due to contamination of air sampler	5 ppb

In addition to errors in the measuring equipment, also the measuring method causes some errors. When measuring flow in ducts with constant emission of tracer gas there is only one source of error: Insufficient mixture of the tracer gas into the air. This kind of error can be estimated by means of a traverce measurement in the sampling point. The dispersion of the tracer-gas concentration indicates the upper limit for this measuring error. A typical mixture error when measuring flow in ducts is 20 o/oo.

Total error when measuring in duct with flow of 100 m^3/h and a dose of 0.5 ml/s:

Error, tracer-gas measurement (18 ppm)	30 o/oo
Error, dosing	22 o/oo
Error, contamination of sample (only negative errror)	0 o/oo
Error, insufficient mixture	20 o/oo
Total error for measurement	42 o/oo

Total error when measuring in duct with flow of 100000 m^3/h and a dose of 15 ml/s:

Error, tracer-gas measurement (0.54 ppm)	33 o/oo
Error, dosing	22 o/oo
Error, contamination of sample (only negative error)	9 o/oo

Error, insufficient mixture	20 o/oo
Total error for measurement	49 o/oo

When air change rates are measured with the method of constant concentration of tracer gas, it is possible to measure in one room or in many rooms with a more or less free air flow between the rooms. If the measuring area is divided into many rooms, the outdoor air change rate for each room as well as for the entire measuring area can be determined. It is relatively easy to show that the error on the measurement of the outdoor air change rate for one of the rooms in a measuring area can be very large if the room has a considerable air exchange with other rooms in the area. Large errors on the measurement of outdoor air change rates for the individual rooms, however, do not give a large error for the measurement of air change rate for the whole measuring area.

When the entire air change rate for the measuring area is measured, two errors caused by the method used should be taken into consideration:

1. The difference between the concentration in the measuring point and the average tracer-gas concentration of the air leaving the measuring area.
2. The error caused by fluctuations in the tracer-gas concentration in the measuring point.

Example: A measurement in a 300 m^3 building with an air change rate of 1.0 times/h. The difference between the average tracer-gas concentration in the room and the average tracer-gas concentration in different points at outer walls, ceilings and floors is max. +/-4%. The fluctuation of the concentration in each measuring point is also max. +/-4%.

For this measurement the error will be as follows:

Error, tracer-gas measurement (1 ppm)	31 o/oo
Error, dosing (Gaussian distribution)	22 o/oo
Error, dosing (only negative error)	2 o/oo
Error, contamination of sample (only negative error)	5 o/oo
Error, measurement of mean concentration	40 o/oo
Difference between concentration in air leaving measuring area and mean concentration in room	40 o/oo
Error, fluctuation of tracer-gas concentration (averaging time when calculating air change rate 1 hour)	57 o/oo
Total measuring error	96 o/oo

When filtering concentration measurements by means of kalman-filter, or when averaging over longer periods of time, the error for the actual measurement can be reduced considerably.

5. CONCLUSION

It is our hope that the development through the last 10 years has resulted in equipment which is easy to operate, easy to transport, does not require frequent calibration, and which does not consume a lot of tracer gas. The above mentioned characteristics will ensure that the expenses for performing tracer-gas measurements can be considerably reduced, and will therefore help propagate the measuring method to a wider audience. Even the fact that today you can buy a complete measuring system, will make the measuring method more known. When looking at the accuracy of the measuring method we can draw the conclusion that not much can be gained by refining the measuring equipment, as parameters such as accuracy of calibration gases and variation in time as well as in place in tracer-gas concentrations in the room now constitute the largest sources of error. Data for the variation in tracer-gas concentrations under measurement are scarce, so it is still difficult to give certain estimations about the accuracy of the measuring method in different situations. The current estimation of the accuracy at around 5% seems fairly correct, though.

The fact that the development of measuring equipment and methods incurred great costs will probably not come as a surprise, but the size of the costs is impressive nonetheless. Developing the first generation of measuring equipment took approx. 2.5 man years. When the second generation of measuring equipment was ready approx. 3.5 man years had been spent, and upon completion of the third generation a total of 12 man years had been spent.

PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

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Poster 2

A COMPARISON BETWEEN THE STEP-UP, STEP-DOWN AND PULSE INJECTION TECHNIQUES FOR THE MEASUREMENTS OF THE MEAN AGE OF AIR

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1. SYNOPSIS

A comparison of three injection manners, step-up, step-down and pulse, for determination of the mean age of air was made by using nitrous oxide and sulphur hexafluoride as tracer gases. The concentrations of nitrous oxide and sulphur hexafluoride were simultaneously measured with a dual-channel IR-analyzer. Tests were carried out in a test chamber with air change rates of 3 h^{-1} and 5 h^{-1}. The tracer gases were injected under three conditions: into the inlet air and directly into the room with and without extra mixing fans. The results suggest that the pulse procedure is as reliable as the two other methods used.

2. INTRODUCTION

Various tracer gas measurement strategies have been developed to explore airflow behaviour and the performance of ventilation systems (1,2,3,4). Tracer gas techniques have been applied in different spaces ranging from a small single room to multicelled structures. Depending on the information desired, a tracer gas is released into the inlet air or within the room. The most common measurement strategies are: decay method, step-up technique (constant injection) and constant concentration technique. These techniques have been used for determining ventilation rates, interzonal airflows and air leakage characteristics especially in residential, commercial and public buildings. Typically, in these buildings pollutant sources are passive and uniformly distributed. Contaminant emission rates are rather constant resulting in relatively low concentrations. Therefore, the air quality in these buildings may be expressed in terms of recommended airflow rates and specified ventilation configurations.

In industrial work rooms, where airflow rates usually are notably higher than in public buildings, air contaminants are emitted from numerous sources ranging from a single point source to wide sources with complex geometry. Cross-contamination from several sources occurs frequently. Industrial process emissions are often intermittent, consisting of phases of strong release rates, resulting in high concentration peaks. The occurrence of concentration peaks may be critical from the standpoint of health effects. Both average and peak exposure should be controlled with ventilation processes and other preventive measures. In these conditions we are not only interested in patterns of fresh airflows but also the dispersion routes of contaminants, and the spatial and temporal variation of contaminants in the zone of occupancy.

The tracer gas technique can be used to chart the behaviour of contaminated air. In order to obtain quantitative information on the spread of pollutants, the injection process of tracer gas should simulate as accurately as possible the actual contaminant release. The information taken from step-up, and pulse experiments may be most useful as far as industrial process emissions are concerned. If the system is linear, these two injection modes should theoretically yield the same information of the flow phenomena. The step-up and step-down injection modes are widely used in non-industrial and industrial buildings, whereas the pulse injection technique has been used to a lesser extent (5,6,7,8,9). Recently, Axley and Persily have emphasized the usefulness of the tracer pulse method (10).

The purpose of this paper is to compare three tracer techniques, step-up, step-down and pulse method, for determination of the mean exit age of air or contaminants simulated with two tracer gases. Nitrous oxide and sulphur hexafluoride served as tracers. The concentrations of the tracer gases were simultaneously measured with a dual-channel, rapid-response IR-analyzer. The tests were performed in a test chamber at air change rates of about 3 h^{-1} and 5 h^{-1} which usually occur in industry.

3. METHODS

In order to compare the injection methods for exploring airflows of general ventilation, 50 series of tests were carried out. The experiments were performed in an exposure test chamber (2.0 m wide, 2.5 m long and 2.0 m high). The inlet air induced by the exhaust fan was introduced into the chamber through the rectangular register near the ceiling on one of the shorter walls. Air was exhausted from the opposite wall through outlets positioned near the ceiling. The inlet air was drawn from the surrounding laboratory room at a flow rate of 0.008 m^3/s or 0.013 m^3/s corresponding to approximately 3 and 5 air changes per hour. The exhaust air was dumped outdoors. The flow rate of the inlet air was measured with a thermoanemometer (Alnor Compuflow

GGA-65P). The flow rate of the exhaust air could not be measured accurately due to flow disturbances in the outlet terminals.

Two extra fans were used in some tests to accelerate mixing of the chamber air. The tracer gas source consisting of simple plastic tubing (i.d. 5 mm) was positioned in the test chamber at a height of 1.2 m at a distance of 1.0 m from the wall with the inlet air terminal. During the tests, where tracer gases were injected into the inlet air, the injection tubing was placed in the center of the inlet air duct opening. The release rates of the tracer gases were controlled with pressure-reducing valves, conventional float rotameters and dry gas meters. Tracer gases, nitrous oxide (N_2O, density 1.83 kg/m^3) and sulphur hexafluoride (SF_6, density 6.41 kg/m^3), were supplied simultaneously into the inlet air or into the test chamber using the step-up, step-down and pulse injection procedures.

In the step-up method tracer gases were injected at a constant flow rate ranging from 40 ml/min to 175 ml/min depending on the ventilation rate of the test chamber and the tracer gas used. The injection time was about five times the time constant of the test chamber. The step-down experiment began when the injection of tracer gases was stopped. In the pulse method the tracers were fed at constant flow rates during 120 s or 240 s resulting pulse volumes of 0.4 l and 0.7 l.

Concentrations of tracer gases were simultaneously measured in the exhaust air by the dual channel infra-red gas analyser (Binos 4b, Leybold Heraeus). The response time of the analyzer was less than 10 s. The output of the Binos analyzer was connected to a microcomputer (Hewlett Packard 9000 Model 310) via an AD-converter (HP 3421A). The sample interval was set to 10 s. The baseline drifts of the infra-red gas analysers were checked before and after the experiment. The eventual correction was done before calculating the mean exit ages. The mean exit ages were calculated by computer programs described by Niemelä et al. (11).

Air velocity at the tracer gas injection site was monitored during the tracer gas tests. The air velocity was detected by an omnidirectional hot wire probe (TSI 1620, Thermo-Systems Inc.) at a sampling frequency of 1 Hz. Based on these instantaneous velocity readings the arithmetic mean and the standard deviation of the sampling period of 3 minutes were calculated and stored by the microcomputer.

4. RESULTS

Typical concentration responses for the three techniques used are shown in figures 1-2. It can be seen that notable concentration fluctuations occurred when the tracer gases were injected within the room. The mean exit ages measured with different procedures are given in the Appendix. Figure 3 gives a summary of these results in nondimensional form. The age values have been scaled by the nominal time constant based on the inlet airflow rate, τ_s=V/Qs, (V= the room volume, Qs= the flow rate of the air supplied).

When the tracer gases were injected into the inlet air, all procedures adopted gave the same value for the mean exit age within 6 % for SF_6 and within 11 % for N_2O. The results also show that the mean exit age of the inlet air was about 15 % less than the time constant based on the inlet airflow rate. The difference indicates air infiltration through the structure of the chamber. This was not surprising, because underpressure between the chamber and the surroundings existed.

When tracer gas was injected into the chamber with the mixing fans on, the mean value of the SF_6 step-up data was 7 % higher than that of the decay data, and the corresponding figure for N_2O was 13 %. Compared to the SF_6 pulse data, the step-up data yielded 5 % higher ages during the tests where tracer gases were injected into the chamber without extra mixing. The corresponding difference between the N_2O step-up and pulse procedures amounted to 20 %.

The step-up technique with N_2O yielded the worst repeatability, 8-12 % (expressed as a relative standard deviation). The repeatability of the pulse experiments ranged from 3 to 7 % and that of the step-down procedure was 5 % or less.

Figure 4 gives the typical air velocities and corresponding standard deviations based on the averaging time of 3 minutes monitored during the tests with and without mixing. A summary of the mean velocities, standard deviations and turbulence intensities is depicted in figure 5.

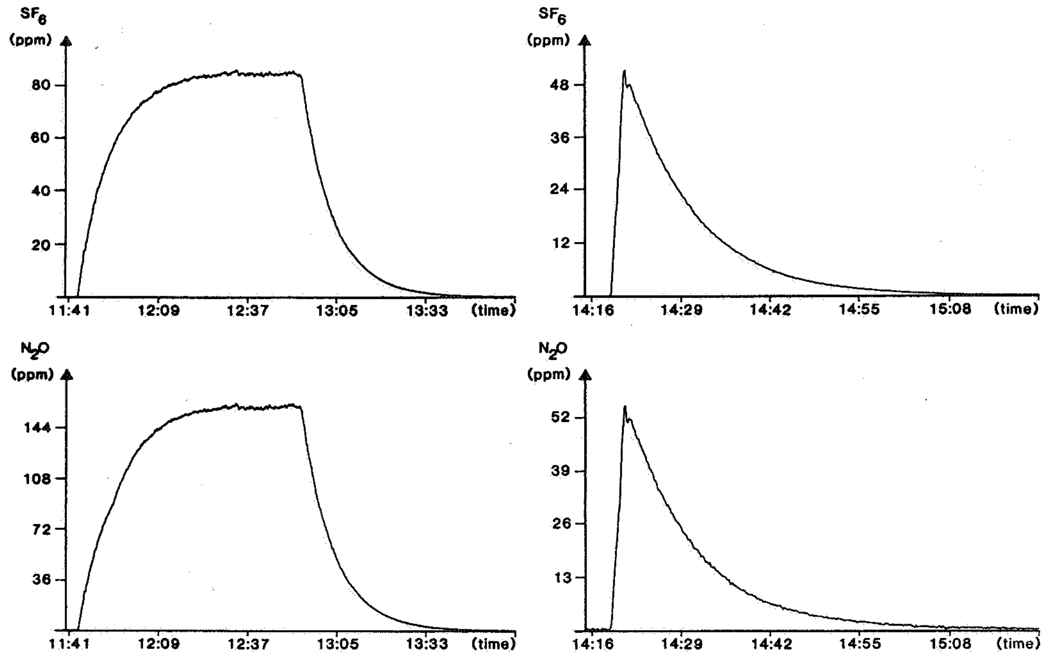


Fig 1: Typical concentration curves from tests with the injection of tracer gases into the inlet air.

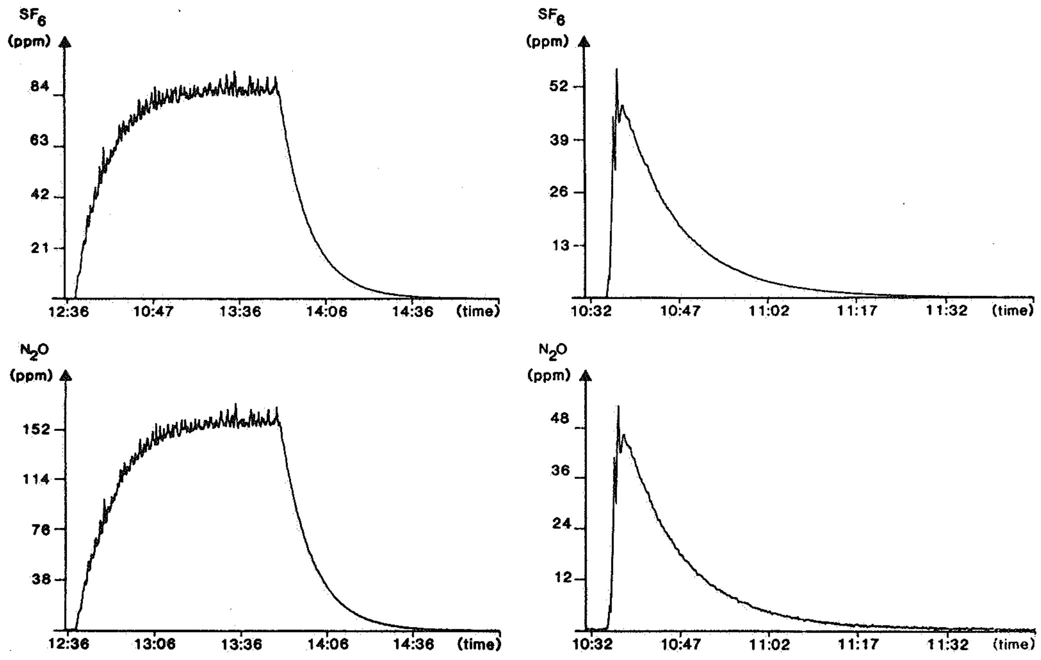


Fig 2: Typical concentration curves from tests with the injection of tracer gases in the chamber.

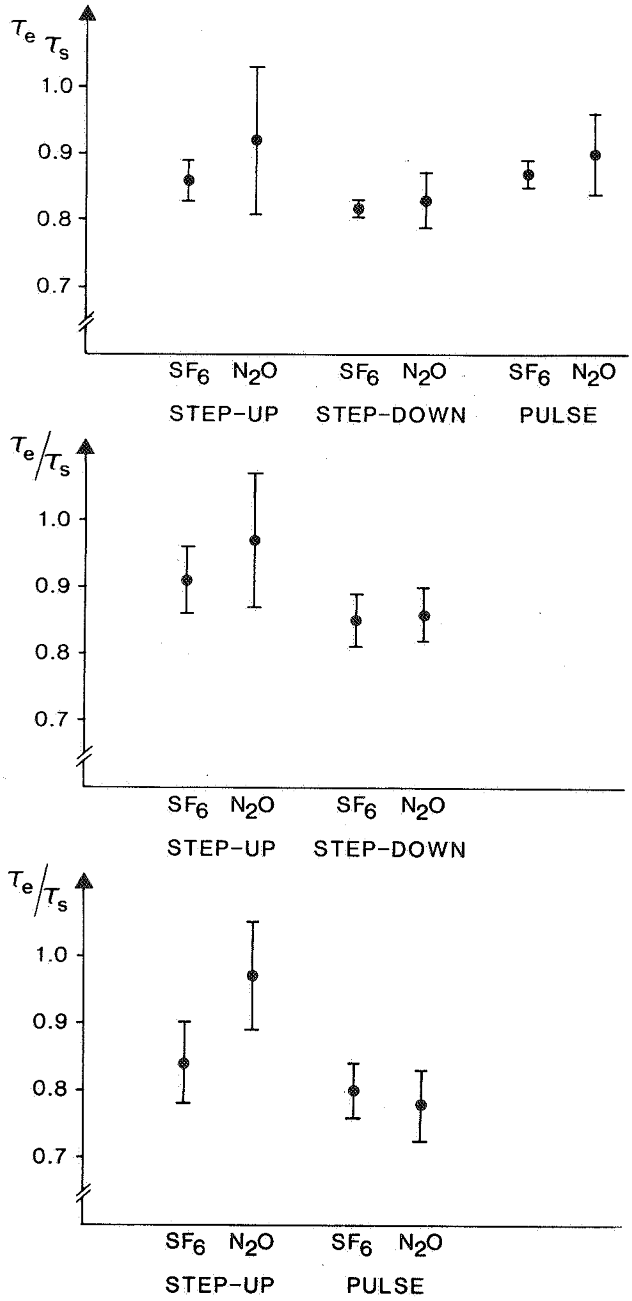
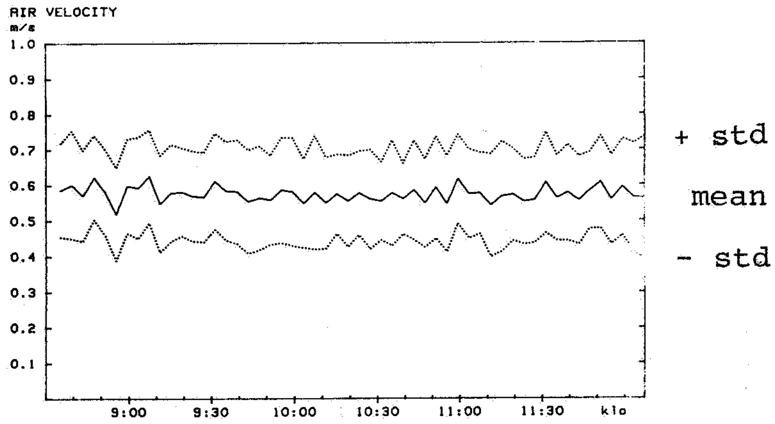


Fig 3: A summary of the test results.
Top: Injection into the inlet air.
Center: Injection in the chamber, mixing fans on.
Bottom: Injection in the chamber, mixing fans off

+ standard deviation

mean

- standard deviation



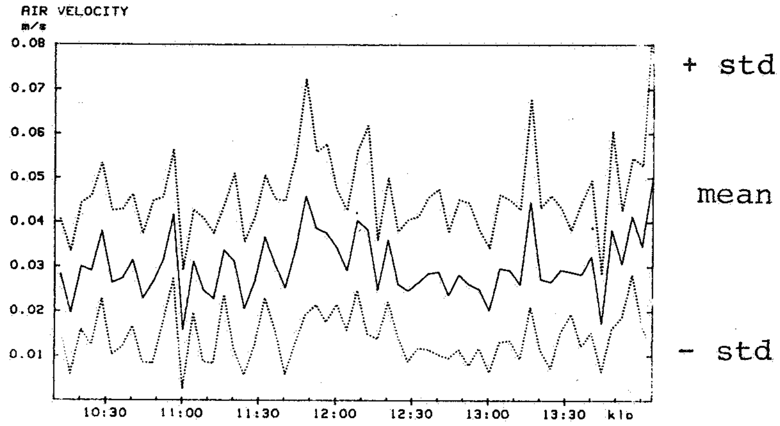


Fig 4: Air velocities recorded at the injection point with extra mixing (top) and without extra mixing (bottom).

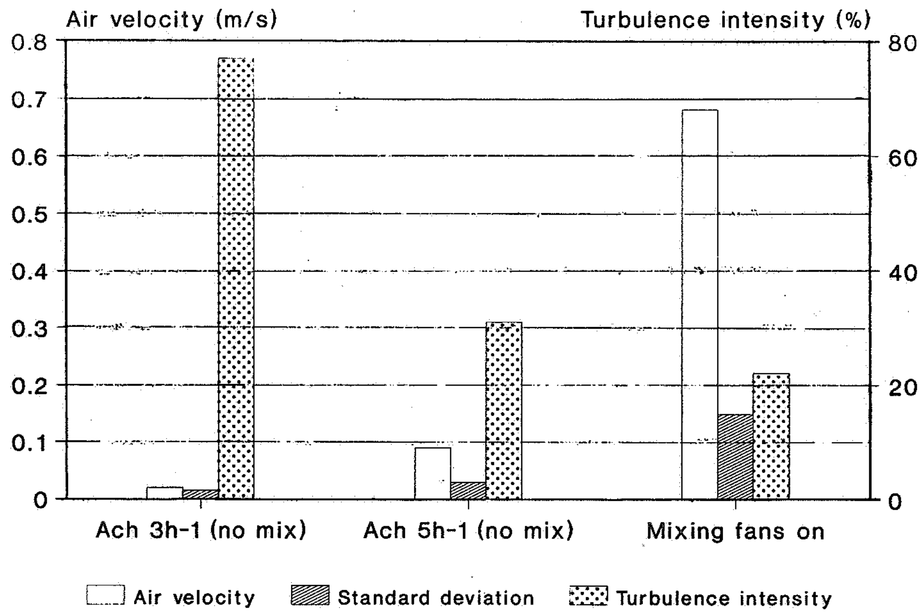


Fig 5: A summary of air velocities measured (ACH = air change rate).

5. DISCUSSION

The N_2O step-up procedure gave slightly higher age values than the corresponding SF_6 procedure. Also the relative standard deviation of the N_2O step-up data was greater than that obtained by other test procedures. This bias was likely due to the instability in the releasing flow rate of nitrous oxide. There was a current trend for the N_2O flow rate to decrease. Apart from these N_2O step-up tests no significant difference between the SF_6 and N_2O data was observed.

The results indicate that the pulse procedure can give as reliable results as the two other methods used. The relative standard deviations obtained in this study agree with the results of Sandberg (5). On the other hand, he reported a larger scatter with the decay method in the tests carried out in a more complicated multicell test house (12).

If a tracer gas is injected into a room to simulate a contaminant source, the pulse and step-up technique should give the same information. Notable concentration fluctuations may occur with step-up injection, especially when the injection point is located in the zone of low and unstable air movements (see figures 2 and 4). Due to fluctuations of the tracer gas concentration it may be difficult to establish the steady state. Therefore, relatively long measuring times are necessary. On the other hand, maintaining stability of the tracer flow rate can be a problem under long-term injections.

The advantage of the pulse method is that the measuring time is relatively short. In addition, as regards industrial short-term emissions, it may be feasible to inject the tracer gas with a time dependence similar to that of the actual contaminant. The disadvantage of the pulse technique is that it is necessary to inject a large amount of tracer gas very rapidly, and this may disturb the flow patterns under investigation. A rapid response gas analyzer is also needed for detecting quickly changing concentrations.

6. APPENDIX

The mean exit age, μe, and the nominal time constant based on the inlet airflow rate.

Test no.	Mean exit age (min) SF6	N_2O	SF6	N_2O	V/Qs (min)	Location of tracer gas source
	Step-up		Step-down			
1.	9.8	9.6	9.8	9.9	12.0	in inlet air
2.	10.8	12.8	10.1	10.3	12.3	"
3.	10.6	11.8	10.1	10.0	12.1	"
4.	11.0	NM	10.2	10.5	12.6	"
5.	10.3	10.7	10.0	10.2	12.3	"
	Pulse					
6.	10.6	10.6			11.8	"
7.	10.8	12.6			12.6	"
8.	10.7	11.1			12.3	"
9.	10.9	10.8			12.8	"
10.	11.0	11.0			12.9	"
	Step-up		Step-down			
11.	12.0	13.0	11.1	11.3	13.0	within chamber, mixing
12	13.3	15.0	11.8	12.5	13.3	"
13.	11.5	10.9	11.5	11.3	12.5	"
14.	11.3	13.0	11.3	11.3	12.8	"
15.	11.5	12.2	11.3	11.5	13.3	"
16.	19.4	21.4	17.6	17.1	20.7	"
17.	16.7	17.1	16.7	16.7	20.0	"
18.	19.4	21.4	16.6	16.6	20.0	"
19.	18.6	NM	18.2	17.6	21.4	"
20.	18.6	18.6	17.1	17.6	22.2	"

NM = not measured

APPENDIX CONT'D

Test no.	Mean exit age (min) SF6	N_2O	SF6	N_2O	V/Qs (min)	Location of tracer gas source
	Step-up					
21.	19.5	21.2			21.6	within chamber, no mixing
22.	16.4	24.0			20.5	"
23.	18.2	19.5			20.3	"
24.	17.8	16.9			20.1	"
25.	18.9	20.2			20.5	"
26.	20.8	22.9			22.2	"
27.	18.3	22.8			22.2	"
28.	19.0	23.1			22.3	"
29.	16.6	17.9			19.3	"
30.	19.9	23.3			24.0	"
31.	9.4	11.2			12.5	"
32.	10.4	12.0			12.5	"
33.	10.2	12.2			13.0	"
34.	10.2	11.1			12.8	"
35.	10.5	14.7			14.7	"
	Pulse					
36.	15.8	15.4			20.7	"
37.	16.7	15.8			20.7	"
38.	18.2	17.1			21.4	"
39.	16.2	15.8			20.7	"
40.	16.7	16.2			20.5	"
41.	17.1	17.1			23.1	"
42.	17.5	18.1			22.5	"
43.	14.2	13.8			18.8	"
44.	14.8	14.5			20.3	"
45.	13.7	14.2			17.9	"
46.	10.7	11.3			12.7	"
47.	10.8	10.5			12.8	"
48.	11.4	10.9			12.9	"
49.	10.3	10.4			13.1	"
50.	10.4	10.3			13.1	"

7. ACKNOWLEDGEMENTS

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PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

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Poster 3

DEVELOPMENT OF A MICROPROCESSOR-CONTROLLED TRACER GAS SYSTEM AND MEASUREMENT OF VENTILATION IN A SCALE MODEL

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SYNOPSIS

This paper describes the development of a microprocessor-controlled tracer gas system which is capable of collecting a large number of tracer gas samples at short or long intervals. The system can be used for accurate measurement of air flow through openings, e.g. cracks, windows and doorways.

The sampling speed of the system can be adjusted so that a larger number of tracer gas samples can be collected during the transient period of an experiment and smaller number of samples during the dominant period. This technique minimises the error in the term dC/dt (where C is the concentration of tracer gas and t is time) and hence allows an accurate estimation of air flow rate to be made.

Measurements of window ventilation and interzone air movement have been made in a scale-model. The model, which represents a two-storey house, was built from perspex and had the dimensions 0.85m x 0.85m x 0.85m. The model was provided with a number of windows. In order to carry out measurements of pressure distribution in a wind tunnel, the model was provided with a large number of external and internal pressure tappings. Pressure measurements were used to calculate air flow through openings and the results were compared with tracer gas measurements. Results are presented on ventilation rate, interzone air movement, pressure and discharge coefficients.

NOMENCLATURE

F	Volumetric air flow rate, m^3/s
A	Free area of an opening, m^2
U_{ref}	Reference wind tunnel speed, m/s
P_o	External pressure measurement at the pressure tappings, Pa
P_i	Internal pressure measurement at the pressure tappings, Pa
P_{ref}	Reference static pressure, measured at the free stream, Pa
ΔP	Pressure difference across the opening, Pa
Cd	Coefficient of discharge of an opening (dimensionless)
C_{po}	External pressure coefficient (dimensionless)
C_{pi}	Internal pressure coefficient (dimensionless)
C_1	Concentration of tracer gas at time t in zone 1, ppm
C_2	Concentration of tracer gas at time t in zone 2, ppm
$V1$	Interior volume of zone 1, m^3
$V2$	Interior volume of zone 2, m^3
ρ	Air density, kg/m^3

1. INTRODUCTION

Natural ventilation in buildings is an important area of investigation, as it has a significant effect on energy conservation, air quality and thermal comfort. The characteristics of natural ventilation in buildings are dependent on the wind environment around the structure and the thermal gradient across the building envelope. Prediction of natural ventilation in buildings is difficult because of the large number of variables involved and the interaction between these parameters. Measurements on scale-models in wind tunnels can provide useful data for the design of buildings prior to their construction. The use of scale models to study natural ventilation has been carried out previously by several authors[1,2,3,4], however little work on the accuracy of quantification of air flow through openings caused by wind-driven force appears in the literature. Some researchers[5] have reported a considerable discrepancy between the measured air flow through openings in buildings and that calculated using engineering Handbooks such as ASHRAE[6] and CIBSE[7]. To date most studies of natural ventilation have been carried out on open areas occupying more than 10% of the surface area of a wall. To author's knowledge the only published data available on small openings (occupying 1% or less of the wall area) are due to Dick[8], Bilsborrow and Fricke[9] and Etheridge and Nolan[10]. Natural ventilation through openings occupying less than 1% of the wall area is typically the situation encountered in buildings in cold climates when all windows are closed[9].

Natural wind-ventilation in a building can be determined from a knowledge of wind speed, pressure coefficient and area of the opening. Pressure coefficients can be obtained using full-scale buildings[11] or by scale-model experiments in a wind-tunnel[12,13,14]. In addition, wind tunnel testing provides a simple means of examining the effect of various factors such as, opening sizes and location, wind speed and direction, on the ventilation rate. Ventilation through a single opening (single-sided ventilation) has been investigated by De Gids and Phaff[15], Warren[16] and Crommelin and Vrins[17]. The scale-model described in this paper is used to verify cross ventilation prediction through openings. We have implemented the use of a new tracer gas system to determine the ventilation rate and interzone air movement in a simplified scale-model of a building.

2. DESCRIPTION OF THE SCALE-MODEL

Measurements of air flow were made in a scale-model, Figure 1. The simple model, which represents a two-storey house, had the external dimensions 0.85m x 0.85m x 0.85m. The interior of the model was divided into two floors; each had the dimensions 0.686m x 0.686m x 0.302m. The exterior and interior walls of the model were separated by a cavity wall construction. The model was built from clear acrylic sheet (perspex) in order that air flow patterns could be investigated using smoke.

The scale-model was provided with a large number of pressure tappings on both the internal and external walls. These tappings were made from brass tubes with an external diameter of 3.3mm and an internal diameter of 1mm. Plastic tubes were then extended from the pressure tappings, down the inside of the cavity wall and exited from the base of the model. The plastic tubes were subsequently connected to a multi-tube manometer.

On the walls of each floor of the model, one inlet and one outlet opening each of an area of 256mm^2 were made for ventilation. One additional large opening (area = 0.011m^2) was made between the two floors of the scale-model to represent the stairway. This opening was used to simulate the interzonal air flow between the two floors of the house.

The Loughborough University wind-tunnel was used for testing the scale-model. Free stream velocities of up to 10m/s could be generated in the test section using a blower and a variable speed motor. The test section was fabricated from perspex sheet and had two access doors located on the side walls of the tunnel. A grid was installed on the upstream side of the tunnel floor to simulate atmospheric boundary layer flow.

Pitot-static tubes and hot-wire anemometers were used in conjunction with precision manometers for measurement of static and velocity pressures in the wind tunnel.

3. MICROPROCESSOR-CONTROLLED TRACER GAS SYSTEM

Air flow measurements were carried out using a variable speed microprocessor tracer gas system, Figure 2. The system which is in its first stage of development, is capable of taking samples at intervals as frequent as every 5 seconds. In essence, the tracer gas system incorporates solenoid valves, tracer gas sample bags, a pulse pump, a microprocessor-based controller, a manifold and a by-pass valve. The short sampling period was achieved using a specially designed microprocessor controller. The tracer gas system was designed to take up to 40 samples from each zone at short or long intervals. In a typical experiment the system would collect up to 30 samples during the transient period of the experiment (at 5-10 second intervals depending on the size of the opening and the temperature difference) and up to 10 samples (at intervals greater than 10 seconds) during the dominant period. The large number of data points taken during the transient period minimised the error in the term dC/dt (see equations 6 and 7) and hence allowed an accurate estimation of infiltration and interzone air movement to be made.

As it was possible to adjust the sampling period of the tracer gas system within a wide range (seconds, hours, weeks or months), the system could also be used to measure long term averages of infiltration rate.

The sampling system could be used with different types of tracer gases. Nitrous oxide, N_2O, was used in this work has it has desirable characteristics in terms of detectability and cost. N_2O gas has been used successfully in previous air movement studies[18].

The tracer gas injection unit was designed so that it would allow either the decay or constant emission methods to be used. The decay method requires the use of only a small quantity of tracer gas to carry out air flow measurements, but a continuous flow of tracer gas at a fixed rate is necessary if the constant injection method is to be used.

After tracer gas release and mixing, samples were collected and injected automatically into a portable gas chromatograph or analyser. This allowed the concentration of tracer gas in each sample to be determined.

4. RESULTS AND DISCUSSION

4.1 Cross Ventilation in Each Floor

Following the installation of the scale-model in the test section of the wind tunnel, measurements of pressure distribution and window ventilation were carried out at different wind speeds. In the first set of experiments, cross ventilation in lower floor of the model was investigated. The tracer gas decay method was used in these measurements. A small amount of tracer gas was injected in the lower floor of the model and the dilution of tracer gas was monitored using the microprocessor-controlled tracer gas system.

Figure 3, 4 and 5 show tracer gas concentration against time for air flow parallel to the window (i.e. $\alpha = 0^o$) for three different wind speeds. The decay of tracer gas in the zone was found to be fast at higher wind speeds. The smoothness of the decay curves indicates that the mixing of tracer gas with air in the zone was uniform. The volumetric flow rates through the window were found to be $2.1 m^3/h$ and $5.5 m^3/h$ at wind speeds of 2.9m/s and 6.2 m/s.

The experiments were then repeated to investigate air flow through the window in the upper floor of the scale-model. The same wind speeds as used previously were used in these experiments. For low wind speeds, the air flow rate through the upper window was found to be slightly larger than the flow rate through the window in the lower floor. The reason for this could be the difference in turbulence level in the two floors. Figure 6 shows the variation of air flow rate through the lower and upper windows as a function of wind speed. The results correlated with:

$$F = 2.084 \times 10^{-4} \times U^{1.15}_{ref} \qquad (1)$$

Figure 6 also shows this correlation compared with a previous correlation, or 'rule of thumb', describing flow through windows[19]. The two correlations were found to be in close agreement.

4.2 Estimation of Ventilation Rate using Pressure Coefficients

The ventilation rate through an opening in a building may be estimated from:

$$F = C_d A (2\Delta P/\rho)^{0.5} \qquad (2)$$

Pressure measurements were used to estimate the internal and external pressure coefficients of the model. The external wind pressure coefficient, is defined as:

$$C_{po} = (P_o - P_{ref})/0.5\, \rho U^2_{ref} \qquad (3)$$

The internal wind pressure coefficient is defined as:

$$C_{pi} = (P_i - P_{ref})/0.5\, \rho U^2_{ref} \qquad (4)$$

Combination of equation 2, 3 and 4 gives:

$$F = C_d \; A \; U_{ref} \, (C_{po} - C_{pi}) \, |C_{po} - C_{pi}|^{-0.5} \qquad (5)$$

Pressure coefficients were determined for the internal and external pressure tappings. Figure 7 shows the variation of average pressure coefficient with the reference wind speed for the surfaces A, B, C and D. The formation of vortices around the sharp corners of the scale-model causes large negative pressure coefficients on the surfaces B and D.

Figure 8 shows a plot of local pressure coefficient with reference wind tunnel speed for three different locations on the surfaces B of the scale-model. The local pressure coefficient for B1, B2 and B3 (which represent the average pressure coefficient for each column of pressure tappings) is different for various wind speeds. The largest negative pressure coefficients were observed for B1 which is close to the sharp edge of the model (in direction of air flow) and the smallest negative pressure coefficient were observed for B3.

The discharge coefficients, C_d, for the lower and upper openings were calculated using equation 5. Figure 9 shows the variation of the discharge coefficient with the reference wind tunnel speed. The average value of the coefficient of discharge for the lower window was found to be approximately 0.56 and that for the upper window was found to be approximately 0.62. These results are similar to those obtained by Dick[8] for openings in houses. The analysis showed that the opening in the upper floor has a higher coefficient of discharge than that for the opening in the lower floor. The cause for this behaviour could be partly attributed to the difference in the pressure distributions over lower and upper surface walls of the model surfaces and partly to the difference in the fluctuation of the pressure acting across the openings.

4.3 Interzone Air Flow Measurements

Measurements of interzone air movement in the model were carried out using a single tracer gas technique. The experimental procedure involved the injection of a certain quantity of tracer gas in the lower floor (zone 1) while the opening between the lower and upper floor (zone 2) was closed (see Figure 10). Following tracer gas mixing under natural conditions, the communication opening between the two zones was opened and the decay of tracer gas was then monitored. If one applies the material balances for tracer gas in each zone, assuming that a steady state exists and the concentration of tracer gas in the outside air is negligible, then the rate of decrease of tracer gas concentration in zone 1 at time t is given by:

$$V_1 \, dC_1/dt = - \, C_1 \; (F_{10} + F_{12}) + C_2 \; F_{21} \qquad (6)$$

Similarly, the rate of decrease of tracer gas concentration in zone 2 at time t is given by:

$$V_2 \; dC_2/dt = - \; C_2 \, (F_{21} + F_{20}) + C_1 \; F_{12} \qquad (7)$$

The remaining flow rates can then be determined using the continuity equations as follows:

$$F_{01} = F_{12} + F_{10} - F_{21} \qquad (8)$$

$$F_{02} = F_{20} + F_{21} - F_{12} \qquad (9)$$

The tracer gas volumetric-balance equations can be solved using one of the analysis methods described by Riffat[20].

Air flow measurements were carried out using two microprocessor-controlled tracer gas systems. The first system was used to collect samples from zone 1 while the second was used to collect samples from zone 2. A known volume of tracer gas was released in zone 1 using a mass flow controller. After a mixing period of about 15 minutes, the communication opening between the two zones was opened and samples were taken at 25 seconds intervals. This sampling period was found to be adequate to provide a sufficiently large number of samples during both the transient and dominant periods.

Several experiments were carried out on the scale-model using different wind speeds. Figures 11, 12 and 13 show tracer gas concentration against time for three different wind speeds. The concentrations of tracer gas in the two zones were found to reach equilibrium in a shorter period when the wind speed was high. The smoothness of the tracer gas curves indicates that good mixing was achieved in the two zones. Tracer gas concentrations were used to estimate interzone air flow. Figures 14, 15 and 16 display schematics of interzonal air flows. The flow rates through the lower and upper windows were found to be greater for higher wind speeds. The flow rate through the window in the upper floor was found to be greater than the flow rate through the window in the lower floor in all experiments. This could be a result of the shape of velocity distribution of air at different heights from the wind tunnel floor. When the air velocity was 2.9 m/s, flow from the lower floor to the upper floor was 0.96 m^3/h and that from the upper floor to the lower floor was 0.72 m^3/h. At wind speeds of 4.5 m/s and 5.3 m/s, the air flow rate from the lower floor to the upper floor was negligible.

5. CONCLUSIONS

The experimental results show that cross ventilation in the scale-model is directly proportional to the reference wind tunnel speed. The ventilation rate through the window in upper floor of the model was slightly higher than that through the window in the lower floor. The ventilation rate through the windows depends upon the turbulence level.

The average coefficients of discharge for the lower and upper windows were approximately 0.56 and 0.62, respectively. The largest negative pressure coefficients were observed near the corners of the scale-model.

The air flow rate from the lower floor to the upper floor was negligible at wind speeds above 3.5 m/s.

The use of the variable speed sampling system has proved to be a reliable and simple approach for measuring air flow ventilation rate and interzone air movement.

6. ACKNOWLEDGEMENTS

The author wishes to thank the workshop staff at Loughborough University for constructing the scale-model and the tracer gas systems.

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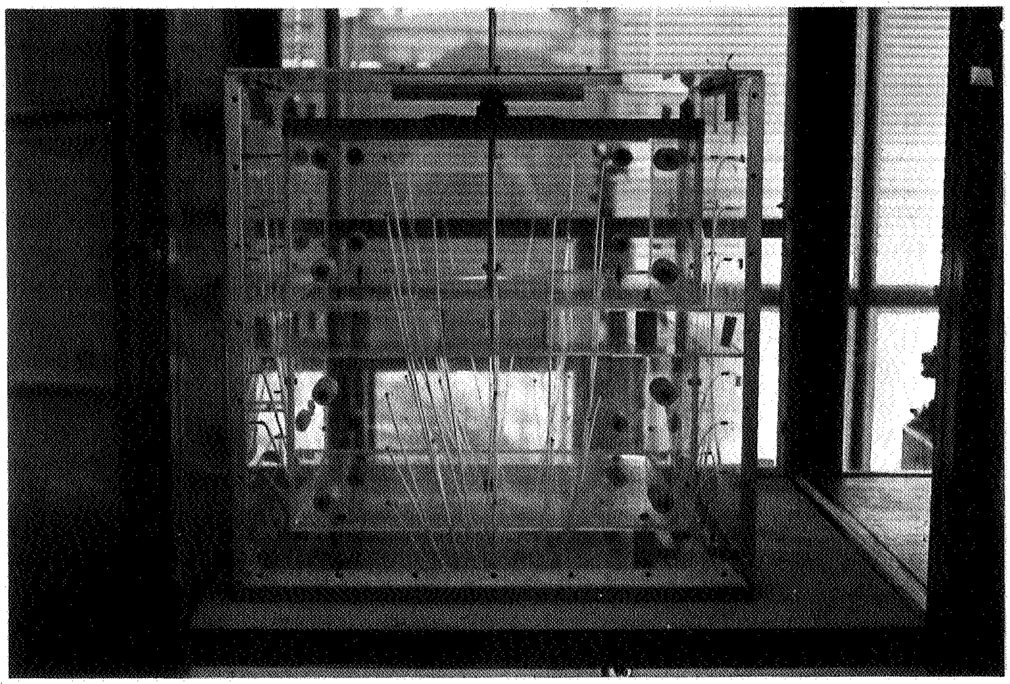

Figure 1 Photograph of the scale-model

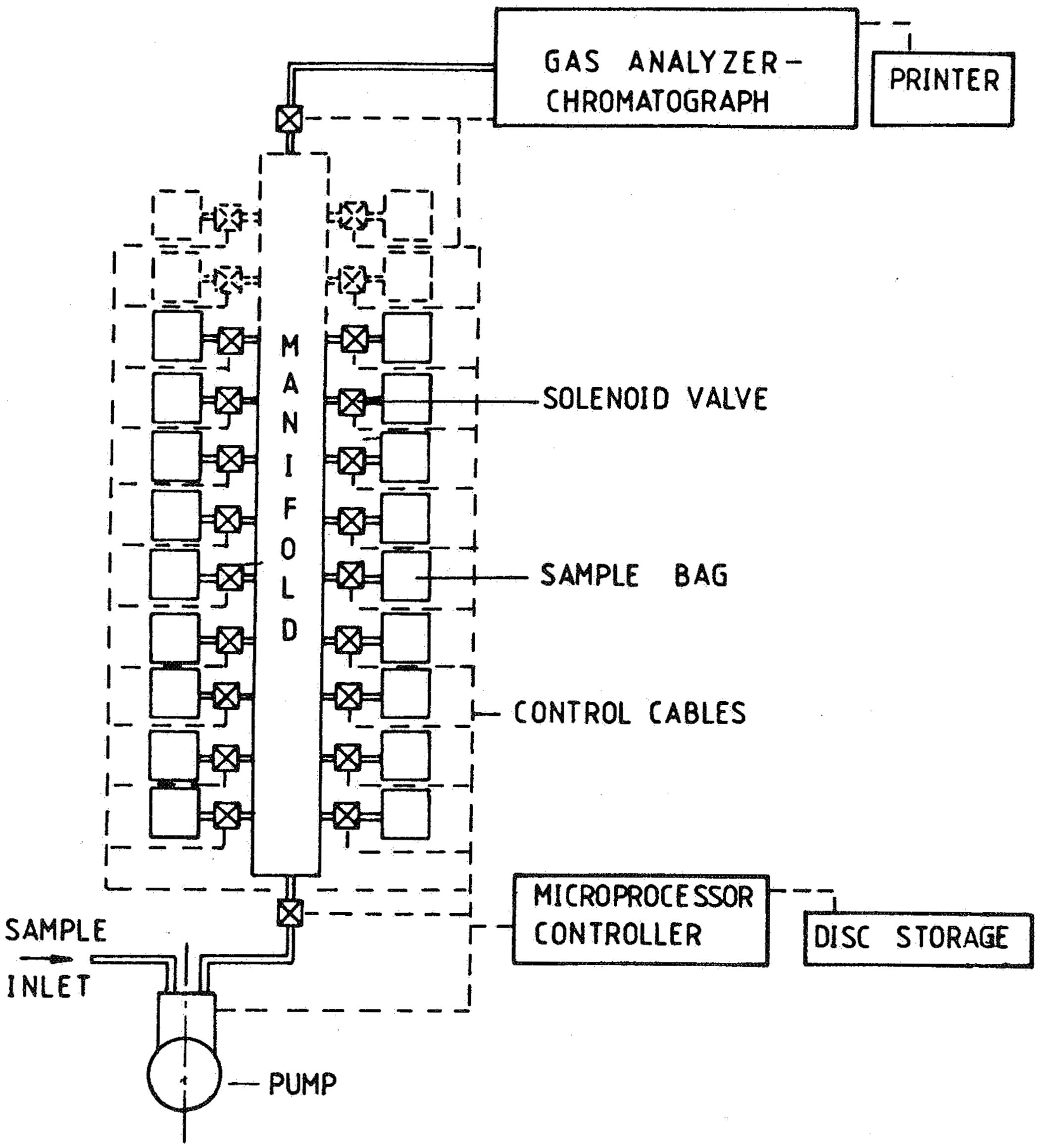

Figure 2 Microprocessor-controlled tracer gas system

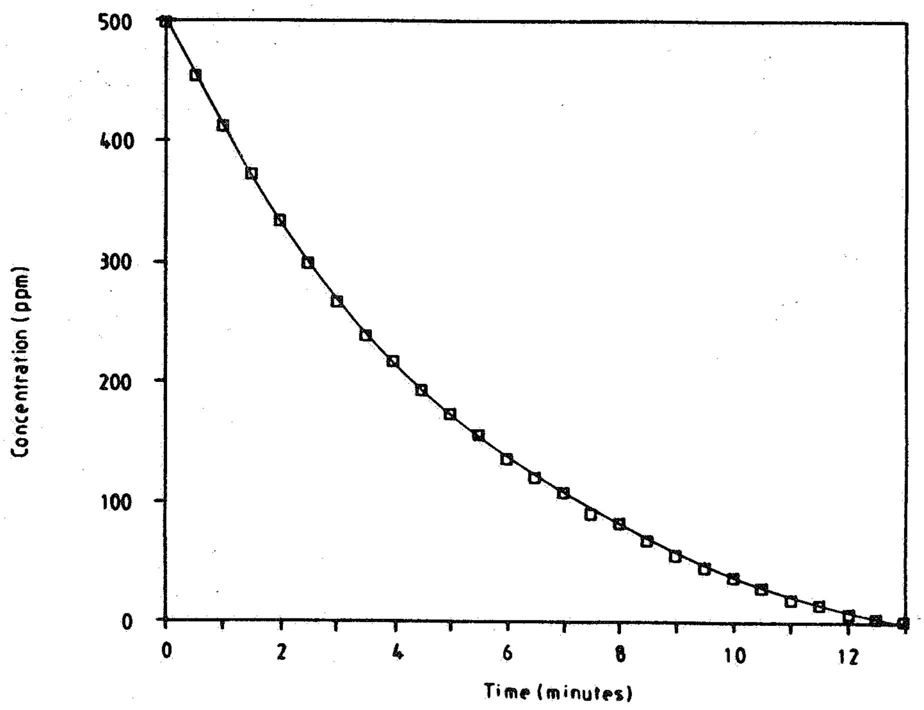

Figure 3 Variation of tracer gas concentration with time, U_{ref} = 2.9 m/s

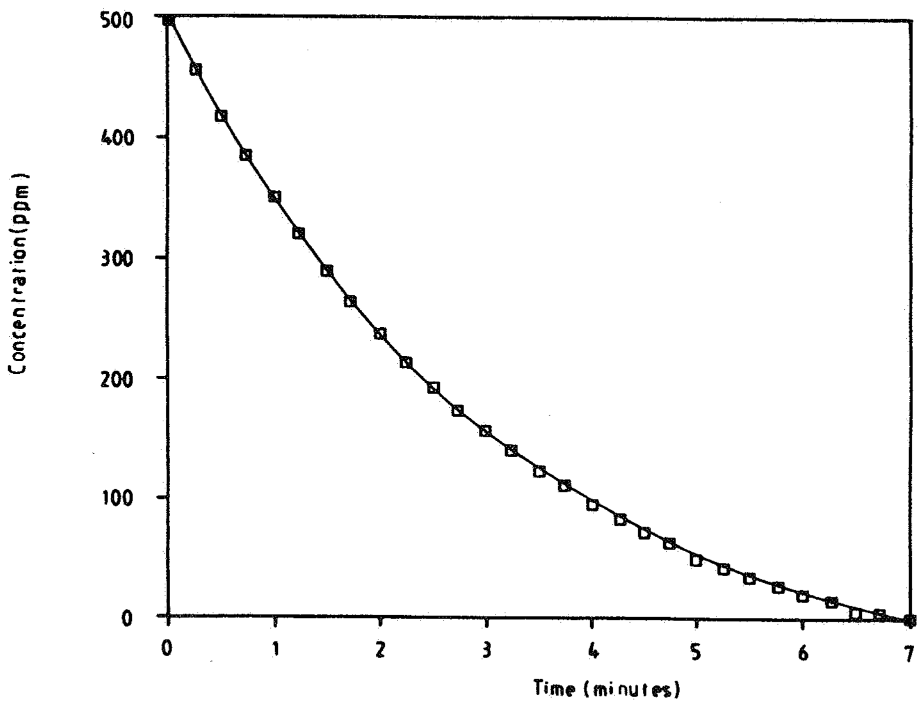

Figure 4 Variation of tracer gas concentration with time, U_{ref} = 4.4 m/s

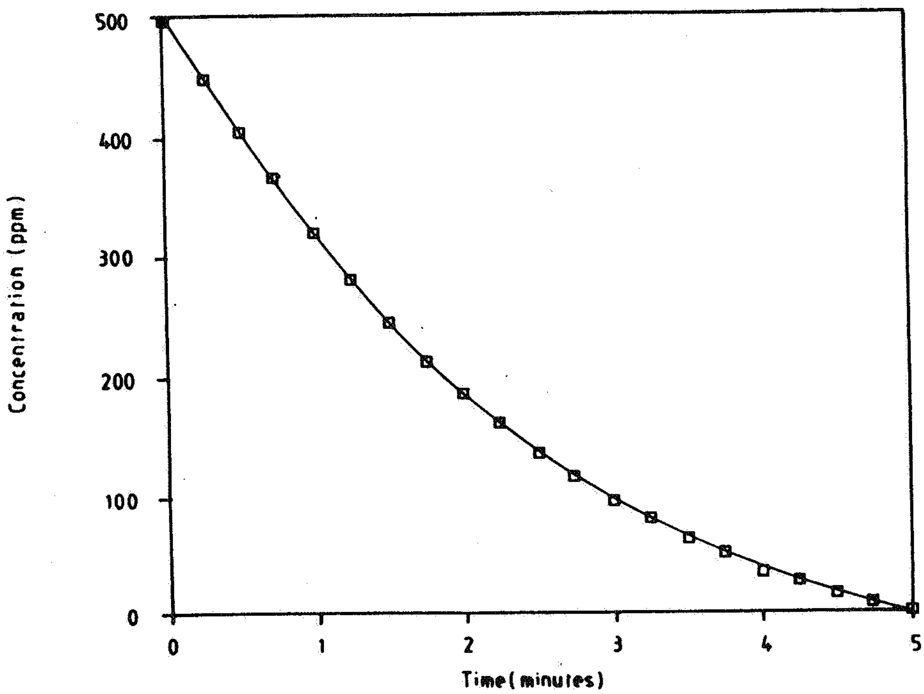

Figure 5 Variation of tracer gas concentration with time, U_{ref} = 6.2 m/s

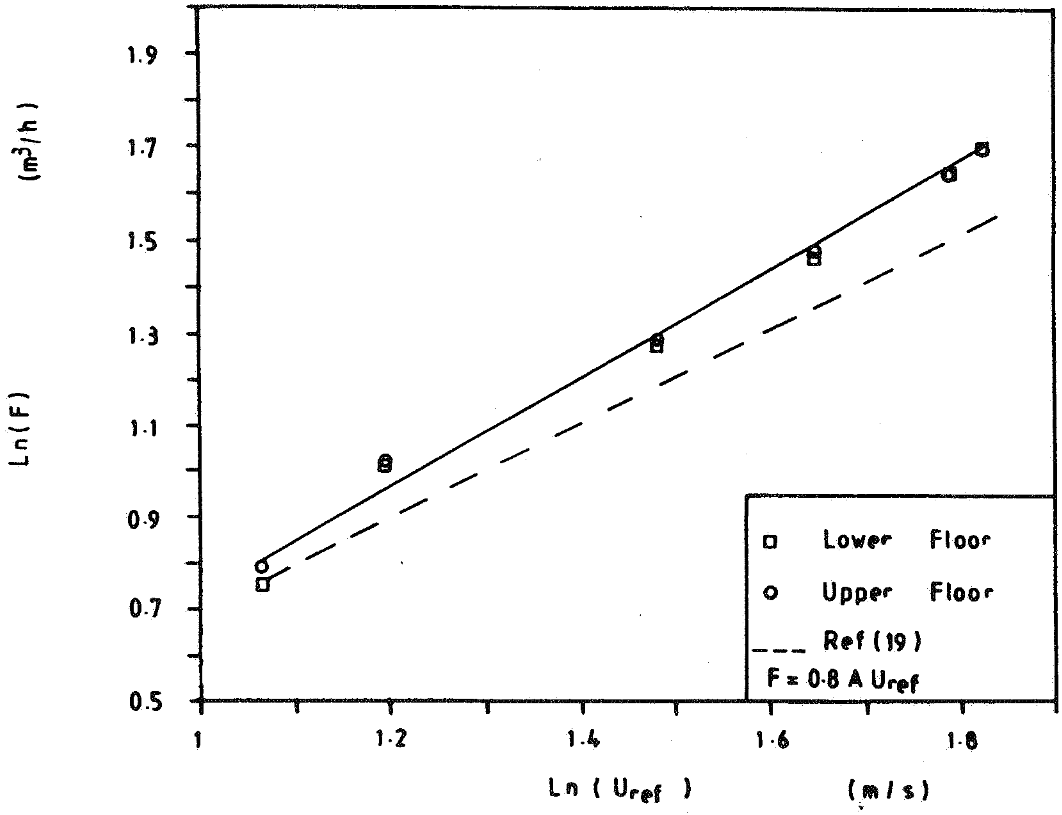

Figure 6 Variation of flow rate through the windows in the lower and upper floors as a function of wind tunnel speed

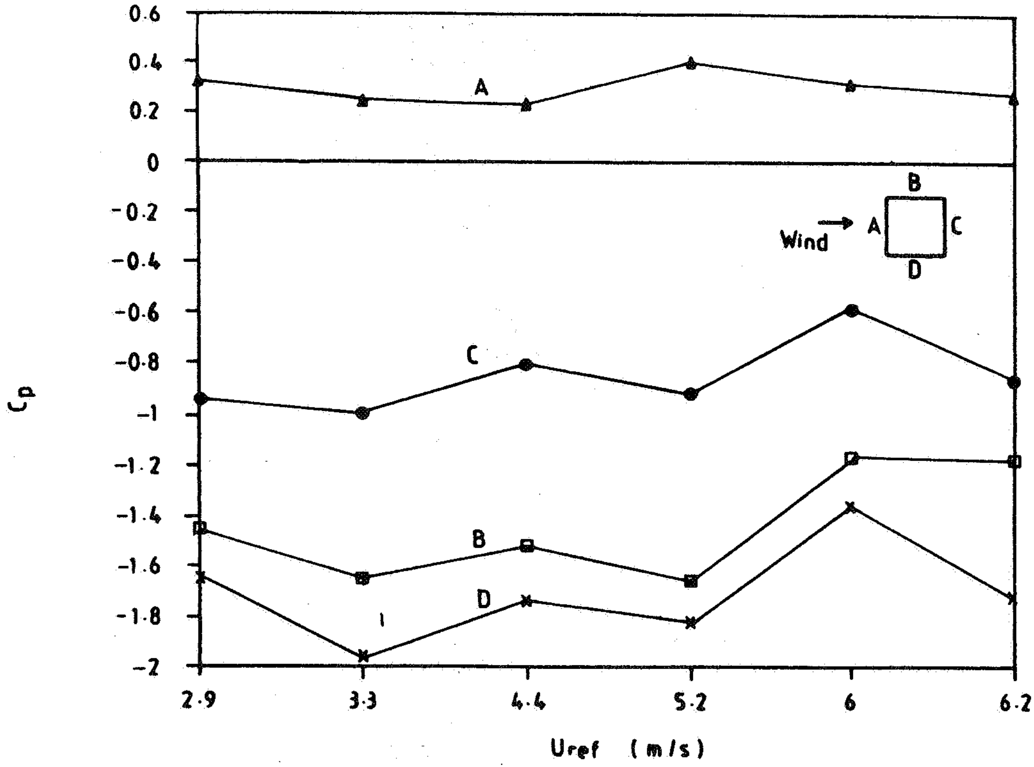

Figure 7 Variation of the average pressure coefficient for the surfaces A, B, C and D with wind tunnel speed

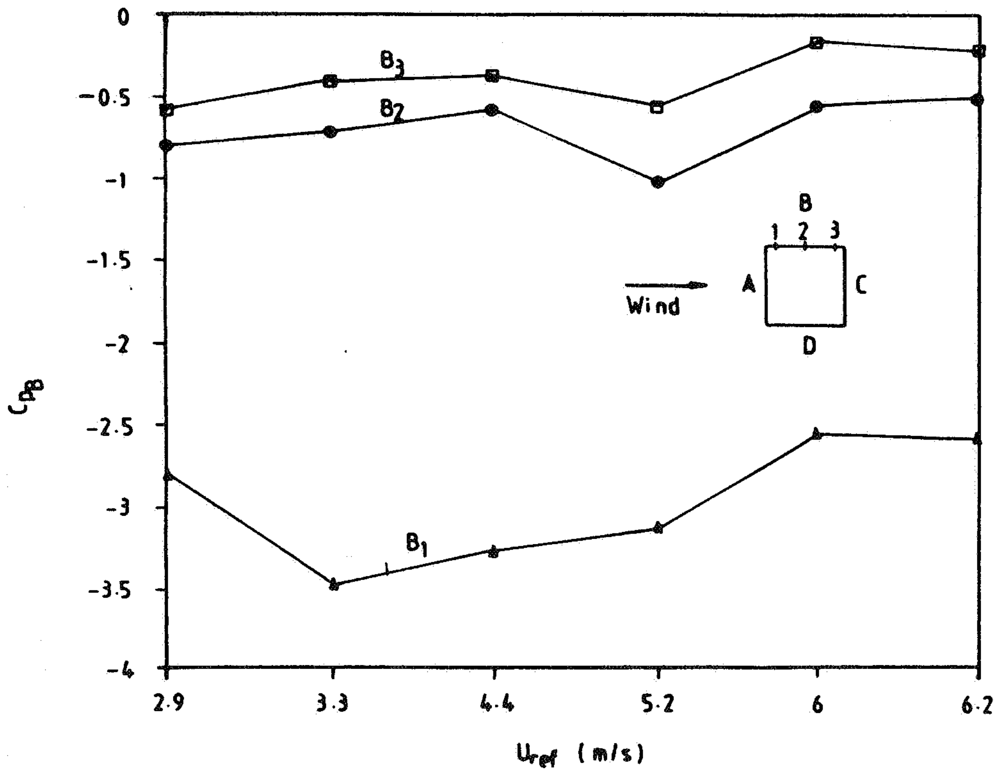

Figure 8 Variation of local pressure coefficient for the surface B with wind tunnel speed

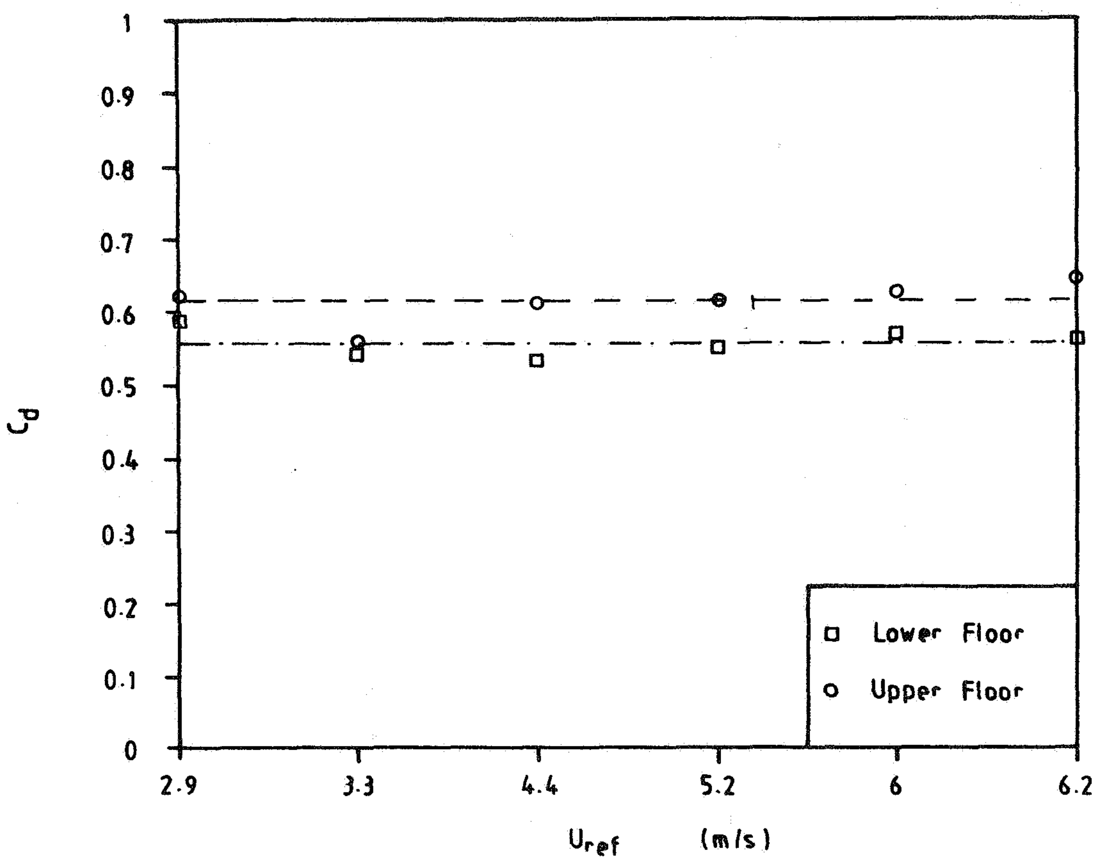

Figure 9 Discharge coefficient versus wind tunnel speed

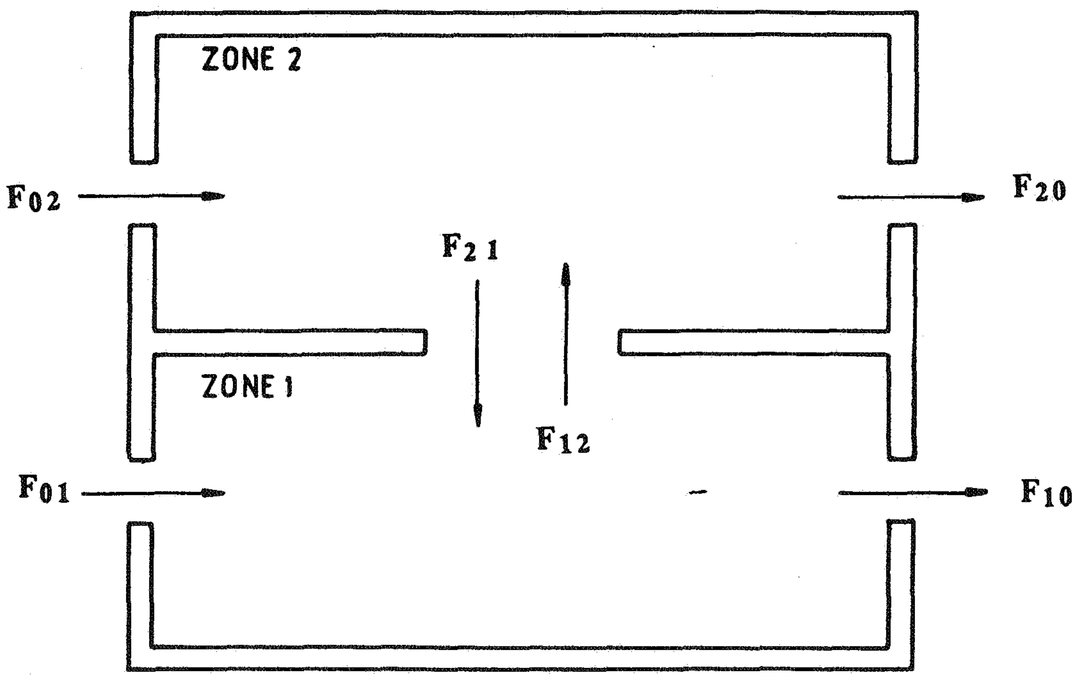

Figure 10 Air movement between two zones

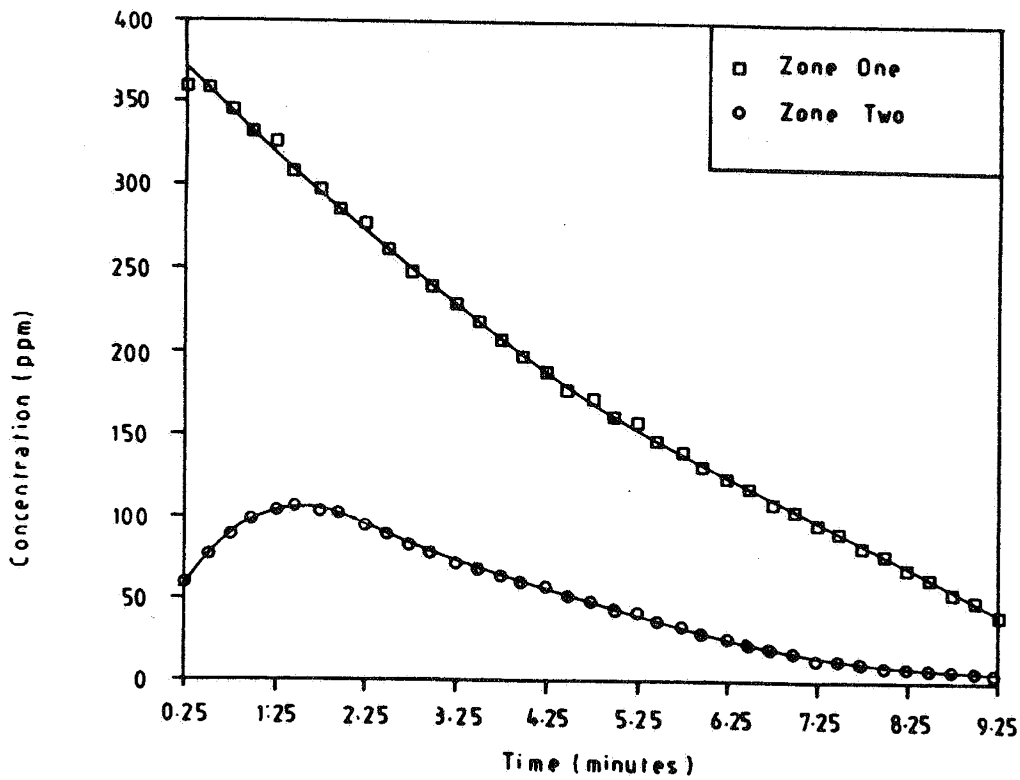

Figure 11 Variation of tracer gas concentration in the two zones with time, U_{ref} = 2.9 m/s

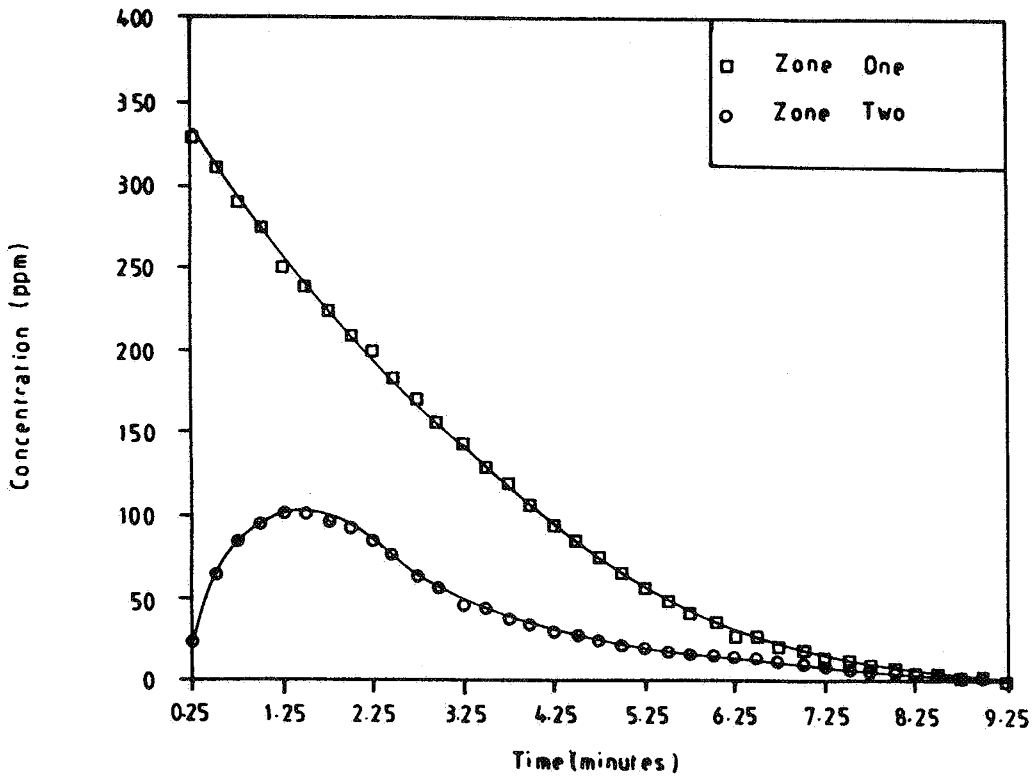

Figure 12 Variation of tracer gas concentration in the two zones with time, U_{ref} = 4.5 m/s

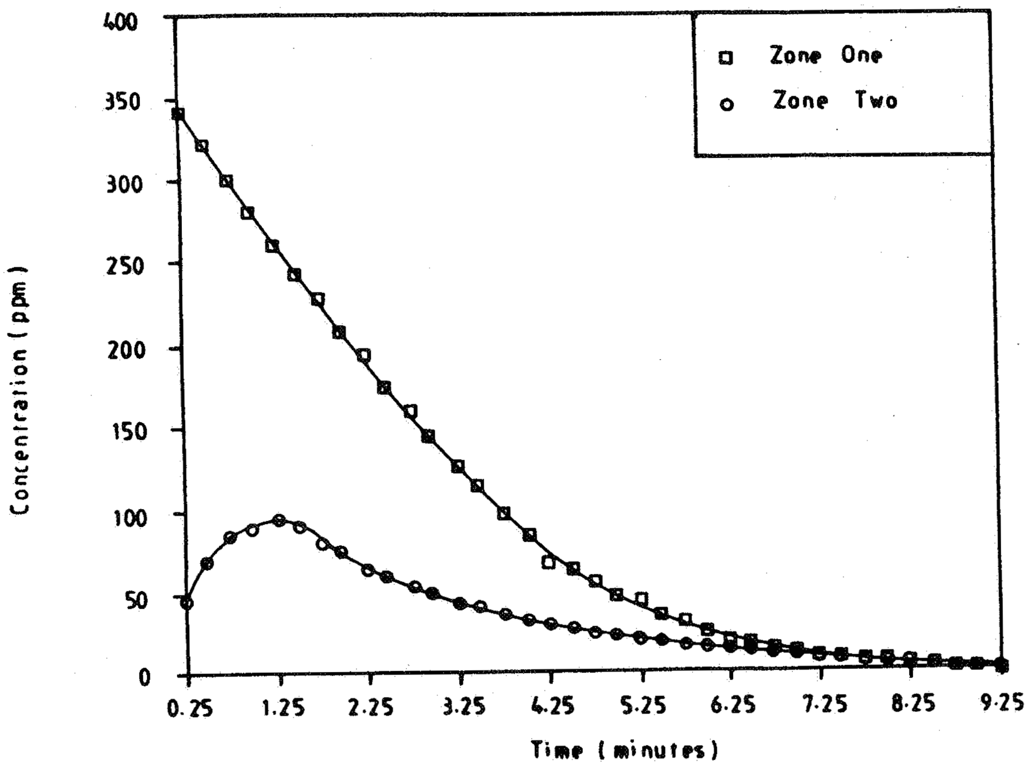

Figure 13 Variation of tracer gas concentration in the two zones with time, U_{ref} = 5.3 m/s

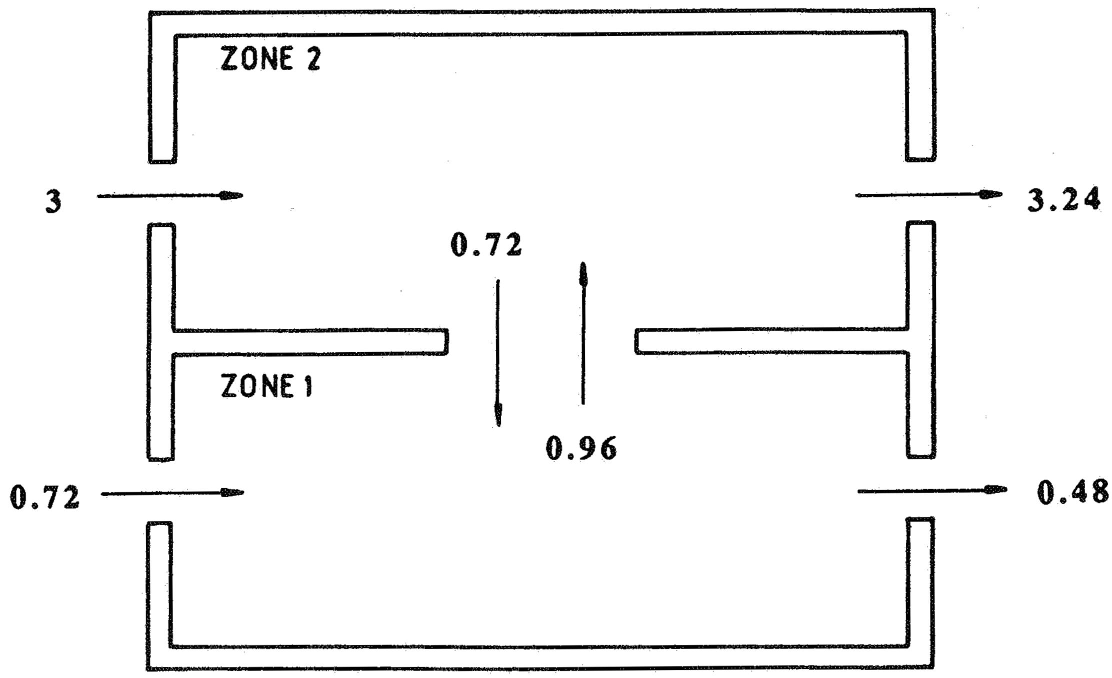

Figure 14 Calculated interzonal flow rates in m^3/h, U_{ref} = 2.9 m/s

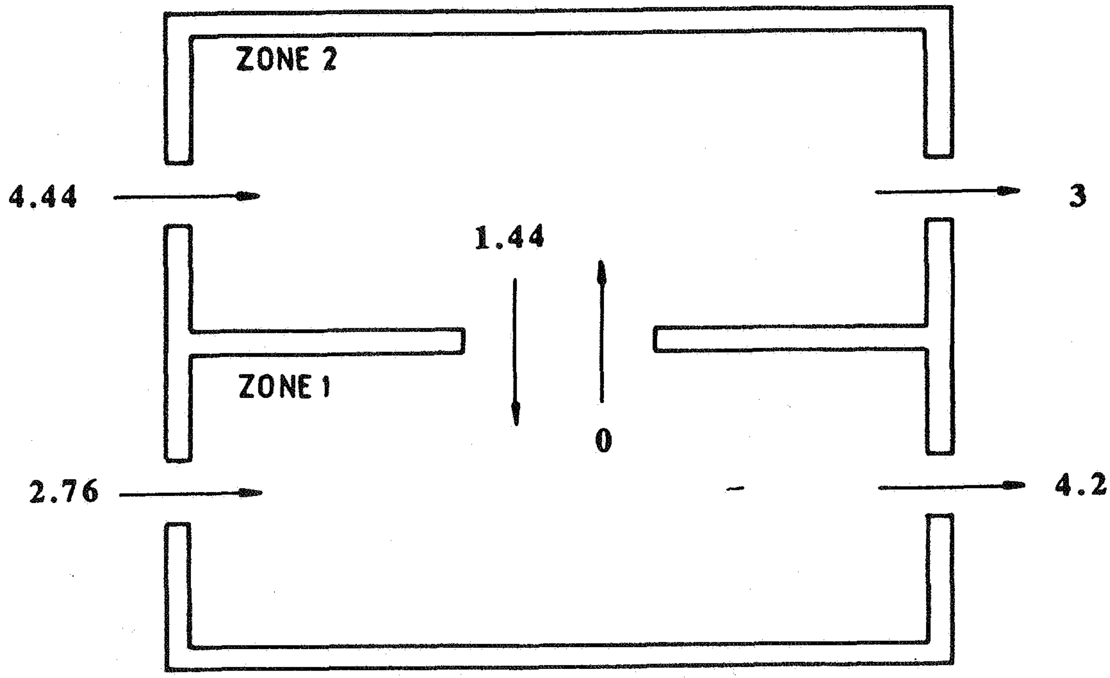

Figure 15 Calculated interzonal flow rates in m^3/h, U_{ref} = 4.5 m/s

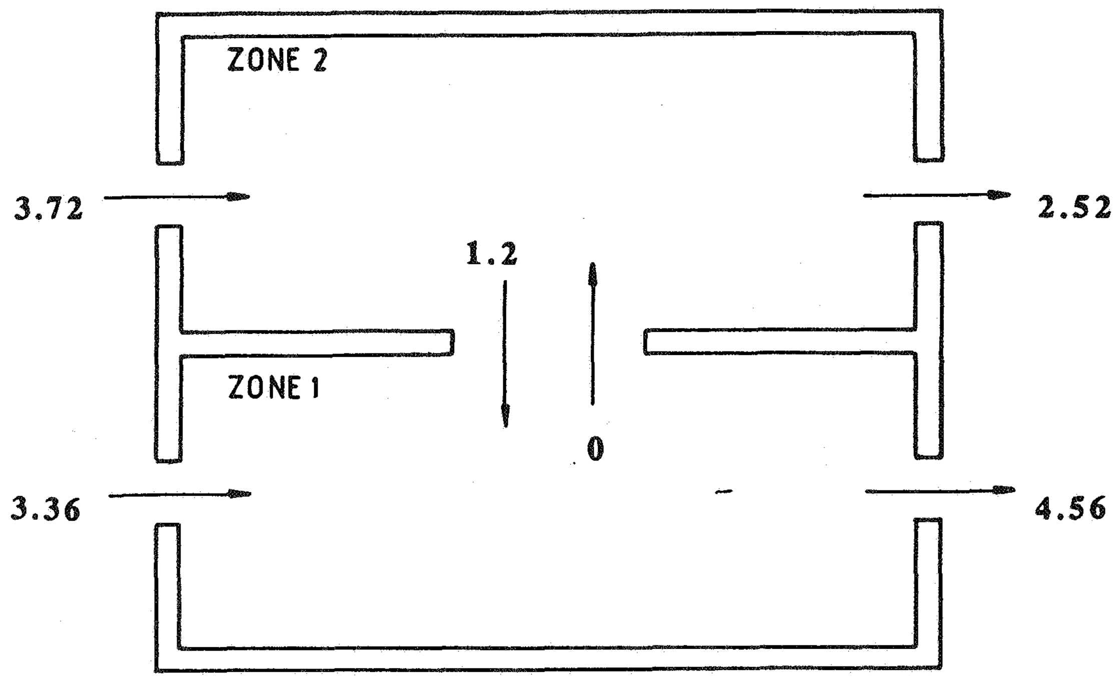

Figure 16 Calculated interzonal flow rates in m^3/h, U_{ref} = 5.3 m/s

PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

10th AIVC Conference, Dipoli, Finland
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Poster 4

METHODOLOGIES FOR THE EVALUATION OF VENTILATION RATES BY TRACER GAS COMPARISON

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1. INTRODUCTION

Ventilation in a building enables to renew the air it contains by means of a natural exchange of air (depending on weather conditions and climate) or a forced exchange using mechanical appliances. This exchange of air must range between minimum air purity and maximum economical limit of dispersion (ventilating means cooling) without causing currents of air, unbearable for the people in the room, which would worsen thermal comfort. It is difficult to envisage the exchange because it depends on numerous variables:

- airtightness of the building;
- difference between internal and external temperature;
- speed and direction of the wind which influences the difference between internal and external pressure.

The studies for the Energy Project at the ICITE the Central Institute for Building Industrialization and Technology of the National Research Council have enabled the development of a method to measure air exchange in homes with the aid of a gas electromechanical analyzer which examines the decreasing curve, in relation to time, of the gas concentration with which the home was saturated.

In this document the authors describe the method used and the results obtained by using two different types of gas: water vapour and oxygen.

The use of oxygen has given the most interesting results which have enabled significant statistical analysis repeatable with any type of climate.

The method used resulted effective although at present there is insufficient data to define the accuracy (estimated around 10%).

Experiments were carried out on a modular prefabricated housing unit built with highly insulating material complete with forced ventilation.

2. METHODS USED

The trial methods used are based on the criteria listed in standard ASTM E 741 - 80 " STANDARD PRACTICE FOR MEASURING AIR LEAKAGE RATE BY THE TRACER DILUTION METHOD ".

This standard describes the standars technique used to measure air exchange in buildings in natural conditions using tracer gas.

A small quantity of this tracer gas is inserted into the room and mixed with the air. The exchange is determined by the decrease in gas concentration in relation to time.

It is difficult to forecast analytically the number of hourly exchanges because it varies according to the airtight level of the building, to its configuration, to the difference in internal/external temperature, to the speed and direction of the wind and to the building's construction quality.

It is, therefore, difficult to quantify the exchange since it depends upon numerous variables.

The experiment was carried out using two separate methods with different traces: water vapour and oxygen.
In this specific case the container's finishing material (fiberglass covered in polyurethane paint) enabled the researchers to disregard the hygrometrical stabilizer effect caused by the container itself.
The possibility of using the humidity content level and its variation to determine the air exchange is in fact attractive because it would make the experiment easier.

3. AIR EXCHANGE MEASUREMENT USING WATER VAPOUR: INSTRUMENTS AND MEASURING METHOD USED

Wile maintaining a constact temperature level inside the module, the related degree of umidity was increased simply by vapourizing water. Vapourizing was interrupted and the humidity conditions inside the module were monitored for various hours.
Using the formula mentioned in the quoted standard we have calculated the rate of air exchange using as a time basis a period of 4 hours, taking into consideration the difference in starting and finishing absolute humidity in the air.
To monitor the temperature and internal and external humidity, we used a data collecting system which controlled temperature and humidity feelers positioned in various points of the module.
The levels were recorded at 5 minute intervals during the whole measuring time.
The 4 hour period was chosen so as to reduce the influence of external temperature and humidity variations on the measurement.
The recordings of the measured parameters are displayed in diagram 1, where we note the following:
- the internal room temperature during the trial varied by 1°C;
- the external environmental conditions in terms of absolute humidity were maintained around 15%;
- the decrease in the level of humidity was regular during the whole measuring time.

By applying the formula mentioned in the quoted ASTM standard we obtain a value referred to volumes/hour :

$$I = \frac{1}{t} * \ln \frac{Ci - Ui}{Cf - Uf}$$

where:
- I is the number of hourly air exchanges;
- t is the period of time considered;
- Ci is the internal starting absolute humidity;
- Cf is the internal final absolute humidity;
- Ui is the external starting absolute humidity;
- Uf is the external final absolute humidity;

The results obtained are summarized in chart 1.

EXPERIMENT	n° EXCHANGES/HOUR
	WATER VAPOUR METHOD
1	0.22
2	0.12
3	0.21
4	0.27
5	0.31
6	0.19

Chart 1

4. AIR EXCHANGE MEASUREMENT USING OXIGEN: INSTRUMENTS AND MEASURING METHOD USED

The instrument used for the trials is an electromechanical analyzer which controls the oxigen concentration in a sample flow.
The analyzer, which uses a sensor cell to measure the oxygen concentration, is equipped with a three range digital detector.
The sensor is self-feeding and works on a battery. It contains an anode, an electrolyte and an air cathode. In the cathode area the oxygen is reduced to hydroxyl ions which iodize the metal anode; the oxygen is therefore used up as soon as it reaches the electrode; the sensor power, exclusively related to the oxygen concentration sampled, is therefore read and translated into oxygen %.
A collecting system memorizes the data supplied by the analyzer, these values are susequently plotted (diagram 2).
As for water vapour, air exchange evaluation is based upon the principle of decrease in the oxygen concentration level in time.
Since the atmosphere is made up of a number of gasses, the composition of which remains stable, (except the value of CO2 which is influenced by various factors) we can assume thet the oxygen percentage in the external environment is also constant.
Past experience has enabled to evaluate this concentration which at sea level is near to 20,8 % ± 0,1% and this value is pratically constant.
The atmosphere in the room where the air exchange will be measured is enriched with oxygen up to 22% by using cylinders.
Subsequently we measure the concentration decreasing time. Also in this case the number of hourly air exchange is calculated by using the formula mentioned by ASTM:

$$I = \frac{1}{t} * \ln \frac{Ci - Ce}{Cf - Ce}$$

where:

I is the number of hourly air exchanges;
t is the period of time considered;
Ci is the initial internal gas concentration (after the release of oxygen);
Ce is the external gas concentration (constant);
Cf is the final internal gas concentration (after t time).

The results obtained are summarized in chart 2.

EXPERIMENT	n° EXCHANGES/HOUR
	OXYGEN METHOD
1	0.35
2	0.37
3	0.37
4	0.38
5	0.32
6	0.36

Chart 2

5. RESULTS AND CONSIDERATIONS

From the experiments carried out, the results of which are mentioned in chart 2, we can consider the following:

- despite the original hypothesis, water vapour measurements resulted unreliable. An example of this can be found in the influence of the external related humidity variation: a fact supporting the unreliability of this method was the condensation which was noticed in various parts of the building (eg. in the intrados openable joints in the floor and on the walls) during the humidification phase. Therefore, during the measured period we passed from the condensation phase to the vapour phase which was reflected in the recorded data with a lower exchange rate;
- judging on the basis of results obtained, oxygen measurements are more repeatable; therefore, the proposed technology can be applied effectively to evaluate the air exchange in buildings.

EXPERIMENT	n° EXCHANGES/HOUR	
	WATER VAPOUR METHOD	OXYGEN METHOD
1	0.22	0.35
2	0.12	0.37
3	0.21	0.37
4	0.27	0.38
5	0.31	0.32
6	0.19	0.36

Chart 3

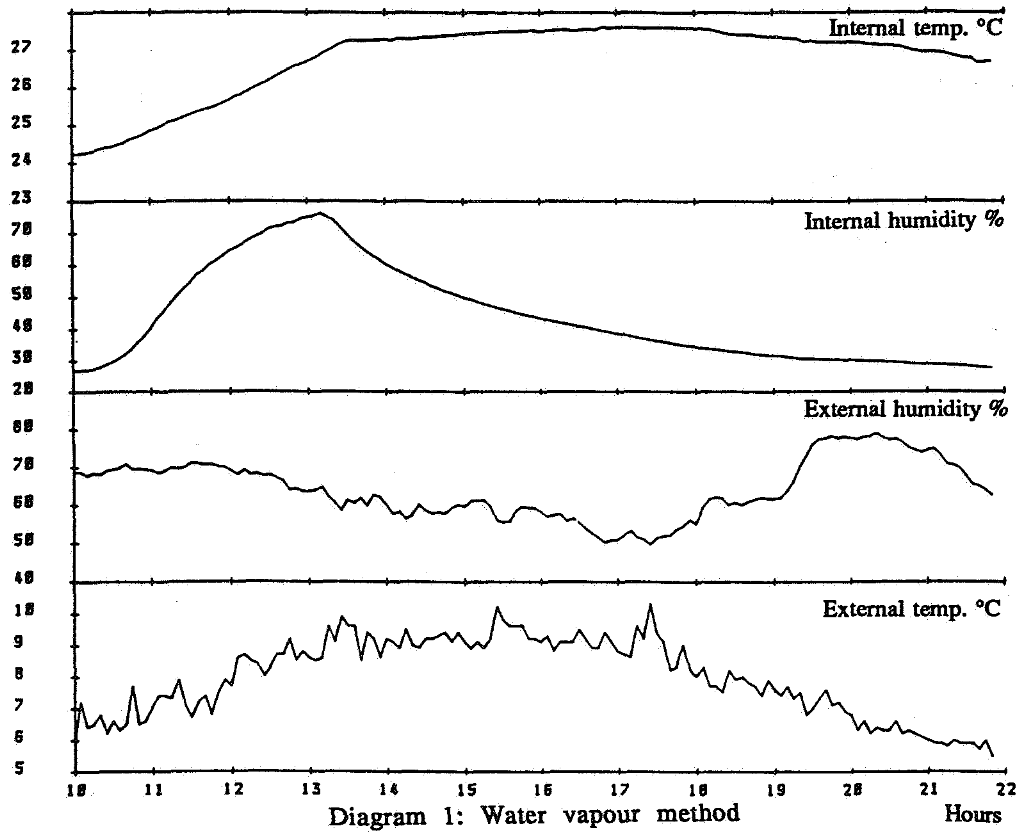

Diagram 1: Water vapour method

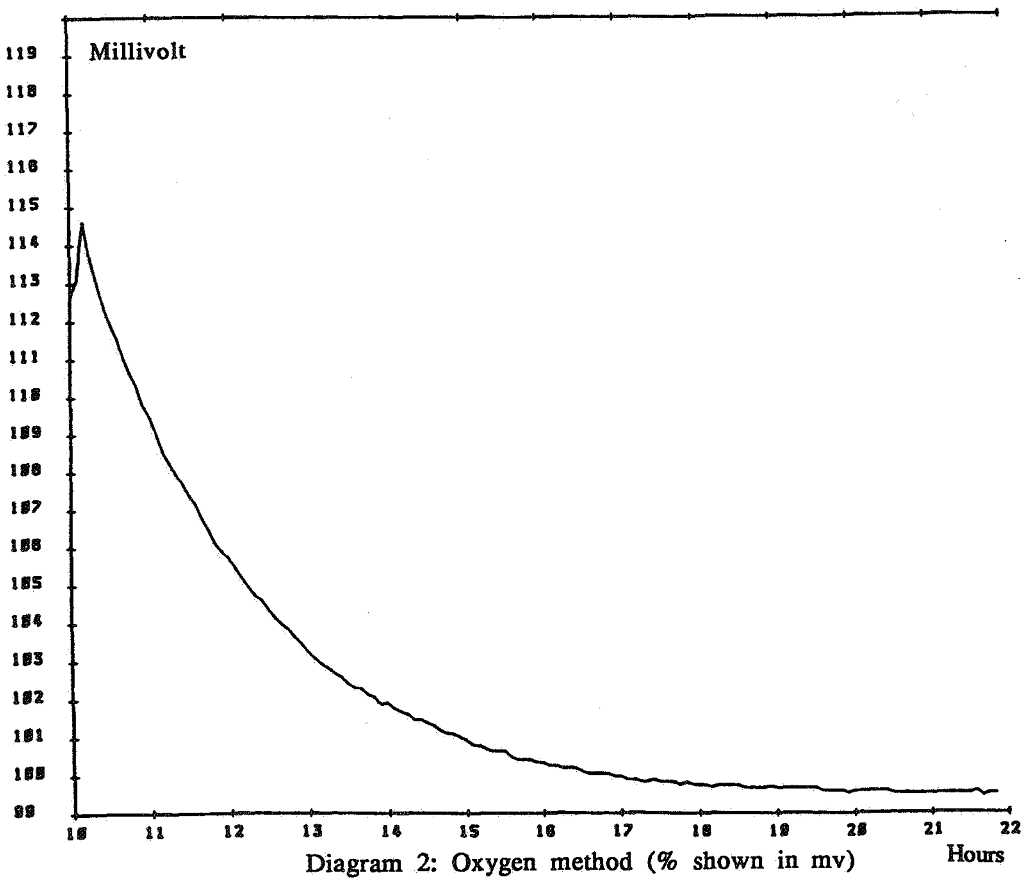

Diagram 2: Oxygen method (% shown in mv)

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Poster 5

A COMPARISON OF DIFFERENT METHODS OF CALCULATING INTERZONAL AIRFLOWS BY MULTIPLE TRACER GAS DECAY TESTS

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A COMPARISON OF DIFFERENT METHODS OF CALCULATING INTERZONAL AIRFLOWS BY MULTIPLE TRACER GAS DECAY TESTS.

C. Irwin and R.E. Edwards

Measurement methods based upon multiple tracer gas techniques have become an established branch of the study of air infiltration and interzonal air movements. Three general groups of techniques have emerged, namely constant concentration, constant emission, and decay.

Of the decay type group of techniques, several methods of deriving airflows from measured concentration/time curves have been suggested. Broadly speaking, these techniques can be classified into three types: numerical methods involving the use of concentration gradients; numerical methods involving the use of integration of concentration/time data; and thirdly, techniques based upon analytical solutions for the fundamental tracer gas equations. The favoured method of analysis at UMIST has been that of a simplified analytical solution in which the effects of tracer gas re-circulation are only taken into account if the degree of connection between zones if high. This method analysis has been successfully validated for the cases of two and three interconnected cells under controlled conditions in environmental chambers. However, up until now, no direct comparison with the results generated by other methods using the same raw concentration/time data has been made.

This paper describes an exercise in which site data for two and three zone regimes is analyzed by several different methods, and the results obtained by each method compared. It is demonstrated that, in particular, concentration gradient methods appear to be particularly ill-suited to dealing with site data which exhibits irregularities in concentration-time profiles caused by fluctuations in windspeed and wind direction. Integration techniques only appear to be marginally better.

(1) INTRODUCTION.

Indoor air quality and energy consumption in buildings are affected by air infiltration and exfiltration through the building envelope. The air movements between rooms thus induced will have a significant influence on the performance of the building in terms of occupant satisfaction and running costs. A means of accurately determining these airflows is therefore of great value.

The inherently complex nature of inter-cell airflows and their susceptibility to changes in environmental parameters such as windspeed, wind direction, and internal/external temperature difference makes their prediction by numerical techniques very difficult: however, several noteworthy efforts have been made to predict inter-cell air movements using network analysis.

For existing buildings, inter-cell air movements can be determined using tracer gas measurement techniques. Several comprehensive reviews of existing tracer gas techniques have been published. (For example Perera (1), Lagus (2), Charlesworth (3)) Three distinct groups of tracer gas techniques have emerged in recent years, namely constant concentration, constant emission, and decay. This piece of work concerns itself with the analysis of tracer gas concentration/time data derived from decay techniques.

(2) METHODS OF DATA ANALYSIS.

(a) The fundamental tracer gas equations. At this juncture, it would be useful to summarise the fundamental equations describing the variation of tracer gas concentrations with time in a multi-cell system. Consider the N-cell model in figure 1. It is assumed that:

(i) the system is composed of N cells in which air and tracer gas are perfectly mixed at the start of,and at all times during, the tracer gas decay test;

(ii) Q_{ij} and Q_{ji} are the volumetric airflow rates between cells i and j; (note that it is not neccessarily the case that $Q_{ij}=Q_{ji}$)

(iii) the volume of the i^{th} cell is denoted by V_i, and is given in cubic metres;

(iv) $C_{i(t)}$ is the time variation in the concentration of tracer gas in the i^{th} cell, and is given in parts per million;

(v) $f_{i(t)}$ represents the rate of production of tracer gas in cell i, and is given cubic metres per hour;

(vi) V_0 denotes the volume surrounding the structure, and is taken to be infinite-this implies that $C_{0(t)}$ is zero throughout the tracer gas decay test.

Consideration of the conservation of mass of tracer gas give

$$V_i dC_i/dt=f_i+[\sum_{j=1}^{N}Q_{ji}C_j(1-\delta_{ij})]-[Q_{i0}C_i+\{\sum_{j=1}^{N}Q_{ij}(1-\delta_{ij})C_i\}] \quad (1)$$

for i = 1, 2, ... N and j = 1, 2 ...... N

The Kronecker delta function (δ_{ij}) is defined as

δ_{ij} = 0 when i<>j

or δ_{ij} = 1 when i = j

A second set of equations is derived by consideration of the mass of air within the multi-cell system:

$$Q_{0i}+\sum_{j=1}^{N}Q_{ji}(1-\delta_{ij})=Q_{i0}+\sum_{j=1}^{N}Q_{ij}(1-\delta_{ij}) \quad (2)$$

If we define S_i as the outflow of air from cell i to the outside, by substitution for S_i in equation (2) we obtain

$$S_i = Q_{i0} + \sum_{j=1}^{N}Q_{ij}(1-\delta_{ij}) \quad (3)$$

There are (N^2-N) unknown values of inter-cell airflows Q_{ij} and Q_{ji}, plus 2N unknown values of Q_{io} and Q_{oi}. We therefore have (N^2+N) unknown values of airflows, and only N equations from equation (1) plus N equations from equation (3) from which to solve for them. Using the N equations of the form of equation (3), N^2 unknown airflows are left, with only N equations of the form of equation (1) remaining from which to solve for them. This means that (N-1) independent sets similar to equation (1) have to be generated in order to be able to solve for all airflows.

(b) Methods of solving the problem.

There are several methods of generating the required (N-1) independent sets of equations. Three methods will be considered:

numerical differentiation, numerical integration, and the derivation of analytical solutions.

(i) Numerical Differentiation

The method of numerical differentiation, adapted to the analysis of equations (1) and (2), can be applied to any number of interconnected cells, the limiting factor being the number of suitable tracers available. The corresponding sets of equations derived from equation (1) will get considerably larger as the number of interconnected cells under consideration increase, and the use of a computer becomes essential for data manipulation.

Equation (1) is solved using the matrix method suggested by Sinden (4). If a single pulse of tracer gas is injected into each cell in the multi-cell system,then it can seen that $f_i=0$: substituting for equation (3) into equation (1) and expressing in matrix form,

$$[Y] = [A].[X] \tag{4}$$

where $[Y] = \begin{bmatrix} V_i dC_{Ai}/dt \\ V_i dC_{Bi}/dt \\ \cdots \\ \downarrow \end{bmatrix}$ for i = 1 to N

$$[A] = \begin{bmatrix} C_{Aj}\ (1 - \delta_{ij}),\ C_{Ai}\ \ldots\ldots\ldots \\ C_{Bj}\ (1 - \delta_{ij}),\ C_{Bi}\ \ldots\ldots\ldots \end{bmatrix}$$

for $j = 1$ to N

$(\delta_{ij} = 0,\ i <> j)$

$(\delta_{ij} = 1,\ i = j)$

$$[X] = \begin{bmatrix} Q_{ji} \\ -S_i \\ Q_{ji} \\ -S_j \end{bmatrix}$$

Airflow vector $[X]$ is found by calculating the inverse tracer gas concentration matrix $[A]^{-1}$

Equation (4) becomes:

$$[A]^{-1}.\ [Y] = [X] \tag{5}$$

When measurements of tracer gas concentration/time histories are made the quantities C_{Ai}, C_{Bi}, C_{Aj}, C_{Bj}, at time (t) are known. Cell volumes V_i can be attained by site inspection. The remaining unknown concentration gradients dC_i/dt, dC_{Bi}/dt are estimated at a specific time (t).

(ii) Numerical Integration

The fundamental tracer gas equation (1) can be rewritten in the form:

$$V_i\ dC_i/dt = fi + \sum_{j=0}^{N} Q_{ji}\ (C_j - C_i) \tag{6}$$

A detailed derivation of this equation is given by Penman (5). Integrating equation (6) between a time step (t_1) - (t_2) we obtain:

$$V_i\ [C_i\ (t_2) - C_i\ (t_1)] = (\sum_{j=0}^{N} \int_{t_1}^{t_2} (C_j - C_i)\ dt)\ Q_{ji} \qquad (7)$$

$f_i = 0$ when a single pulse of tracer is injected into each cell.

The integrals in equation (7) are evaluated using numerical integration of time variations in tracer gas concentration, obtained from site data for the period (t_i) - (t_2).

The tracer concentration curves can be divided into several time periods ($t_j = 1, 2, \ldots k$), the unknown airflows are found using a least squares approximation for $k \geq N + 1$ time periods. Where multiple tracer gas measurements are used, separate estimates of airflows are possible for each tracer gas.

(iii) Analytical Methods

Several attempts have been made to derive an analytical solution of the fundamental tracer gas equations. (See for example Sinden (4) and I'Anson.(6)) Problems occur because of the unknown time variations of tracer concentration $C_{(t)}$ in the connected cells under consideration.

However, from the work of Dick (7) a simplified analytical solution is available which enables estimates of intercell airflow to be made.

Equation (1) can be rewritten in the form:

$$dC_i/dt = \frac{1}{V_i} \sum_{j=1}^{N} Q_{ji}\ (1 - \delta_{ij}) - \frac{S_i}{V_i} \frac{C_i}{} \qquad (8)$$

By use of integrating factors, this first order differential equation can be solved for unknown airflows Q_{ji}, S_i, and becomes:

$$C_i\ e\left[\frac{S_i}{V_i} t\right] = \sum_{j=1}^{N} \frac{Q_{ji}}{V_i} \int_{0}^{t} C_j\ (1 - \delta_{ij})\ e\left[\frac{S_i}{V_i} t\right] dt \qquad (9)$$

A fully detailed account of the equations resulting from the solution of equation (9) is given by Irwin. (8)

(3) Experimental Details

The test house used is a two storey terraced property of low energy design. (Figure 2) Both two and three cell measurements were carried out in this house. For the two cell measurement, upstairs and downstairs were taken as the two cells, whilst for the three cell measurement, the kitchen was taken as a cell in its own right.

Tracer gas concentration measurements were made using the rapid response multiple tracer gas system developed at UMIST: this system is well documented (9, 10) and will not be described in detail here.

(4) Results and Discussion

Table 1 summarises the variations in estimated airflows for a typical two cell case, whilst table 2 shows variations in estimated airflows for a three cell case, using the three methods of data analysis previously discussed. Figure 3 shows the "goodness of fit" with a set of concentration/time data for the two cell case. As can be seen the predicted time variation (equation (1)), using the airflows estimated by numerical differentiation has a poor correlation with the measured data. This is hardly surprising as the method of analysis is reliant upon the estimate of concentration gradients, from two distinct tracer gas concentration/time data points. The uncertainties in the data and their effect on airflow estimates can be clearly seen in figure 3.

There are similar problems with the airflow rates estimated using numerical integration techniques. The "goodness of fit" between measured data and predicted concentration time histories (shown in figure 3) are little better than for the numerical differentiation case. The reason for this lies in the tracer gas concentration difference term shown on the left hand side of equation (7) ie.

$$(C_i\ (t_2) - C_i\ (t_1)) \tag{10}$$

The method of analysis is reliant upon pairs of site data points in constructing the linear equations to solve for the unknown airflows.

The third method using a simplified analytical solution of the fundamental tracer equations does reflect a better comparison between predicted and measured tracer concentration/time histories. This occurs because the analytical method uses all data collected rather than discrete pairs of data points. As a word of caution, the analysis and consequently the estimates of airflows from the three methods discussed in this paper are all vulnerable to extraneous variables.

(5) Conclusions

Comparison of the three methods of analysis for concentration/time data shows that, of the three methods, the simplified analytical solution as described by Irwin et al (9, 10) gives the closest fit to site measurements of concentration variations with time. It should be noted that the sets of data presented have been deliberately selected so that the effects of fluctuations in extraneous variables are minimal. As these fluctuations become more pronounced, the differentiation and integration techniques become even more inadequate, whilst the simplified analytical solution can still be used satisfactorily. This whole issue will be more fully discussed in reference (11).

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TABLE 1

Variations in Measured Airflows (2 Cell Case)

	Airflows (m^3/hr)			
	Q_{12}	Q_{21}	S_1	S_2
Numerical Differentiation	113	203	352	138
Numerical Integration	65	157	282	107
Simplified Analytical Solution	190	215	263	254

TABLE 2

(Three Cell Case)

	Q_{12}	Q_{13}	Q_{21}	Q_{31}	Q_{23}	Q_{32}	S_1	S_2	S_3
Numerical Differentiation	122	18	65	88	105	129	258	288	270
Numerical Integration	89	88	188	55	98	111	299	212	241
Simplified Analytical Solution	106	75	113	77	110	82	325	220	234

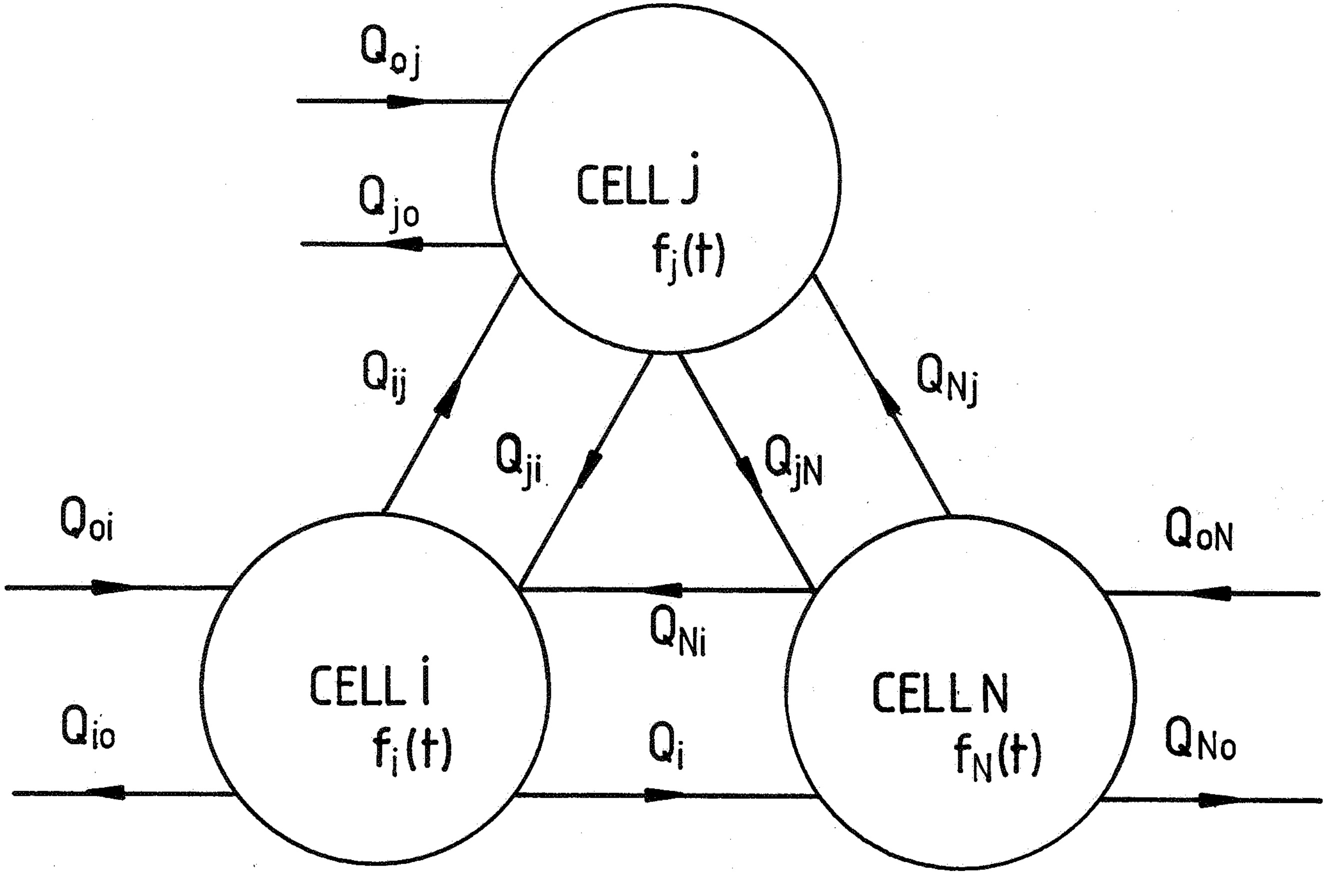

FIGURE 1: Test cell system

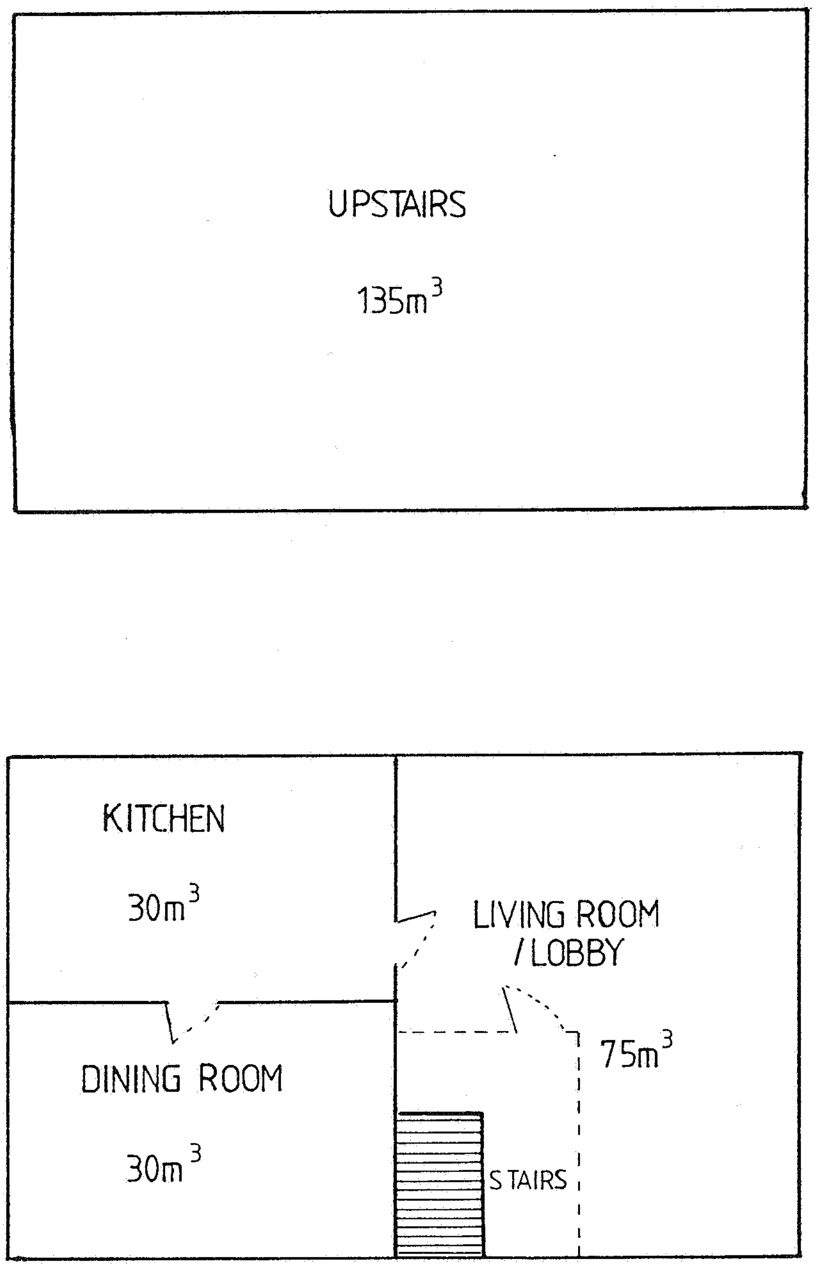

FIGURE2:Plan of test house

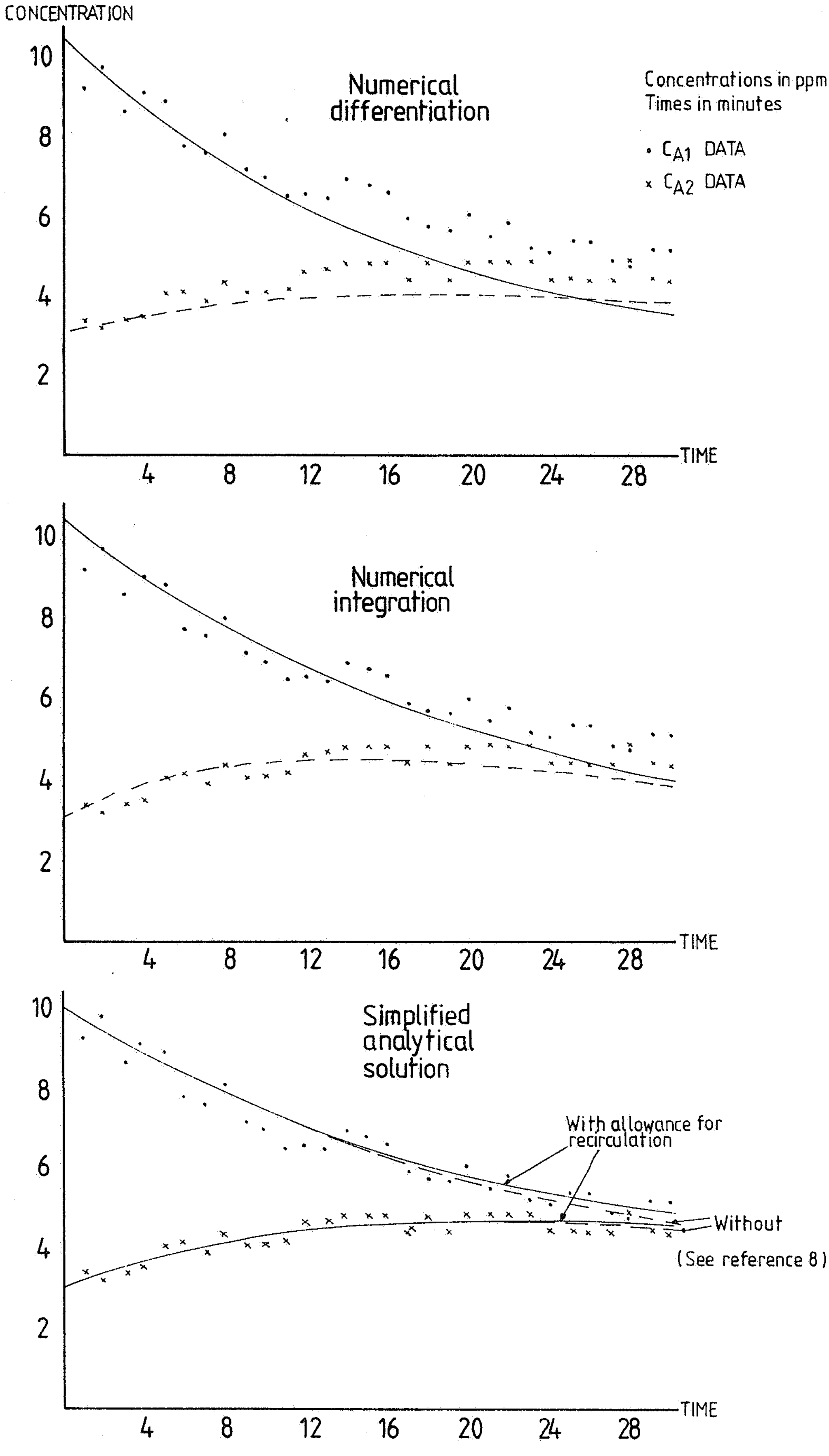

FIGURE 3: GOODNESS OF FIT COMPARISON

PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

10th AIVC Conference, Dipoli, Finland
25-28 September, 1989

Poster 6

A MODERN CONCEPT FOR OFFICE BUILDINGS: ENERGY SAVING AND GOOD INDOOR CLIMATE ARE NO LONGER CONTRADICTORY

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ABSTRACT

An up-to-date design concept for office buildings results in a very low energy consumption and provides a better indoor climate at the same time. This new concept is based mainly on two design features: An *extremely well insulated building envelope* decouples the indoor climate from the outside climate to a high degree during all seasons and weather conditions. The second element of this new design concept is the HVAC-system: The *source-dominated displacement ventilation* provides a better comfort and, as a cosequence of its high effectiveness, is very economical.

Normally windows are the thermal leaks of the building envelope. In order to overcome this deficiency, Geilinger Ltd in Winterthur, Switzerland developped the *High Insulation Technology (HIT)*. Typical glazing-U-values are around **0.65 W/m²K**. The consequences on natural air flow and on the energy balance are enormous: radiator heaters are not anymore necessary in a HIT room and the air change rate of a mechanical ventilation can be reduced considerably compared to the requirements of conventional air conditioning installation.

Modern office buildings usually produce considerable internal loads (occupants, lights, computers, office equipment) which compensate the heat transmission losses in winter. In summer, the excellent insulation protects the building from outside heat during the day. At night free cooling can be used.

Measurements of energy consumption and investigations on thermal comfort were made at different HIT-buildings. One case is presented in this paper.

1. INTRODUCTION

During the last few years it has become widely accepted, that new ways for energy saving in building management must be found. Many people do automatically associate this requirement with a decrease of the user comfort. In this paper it will be shown that the contrary can be achieved. A *new concept*, combining high insulation windows and displacement ventilation, leads to a *higher user comfort and at the same time reduces the energy consumption*. Several office buildings have already been constructed following this concept. Comfort and energy *measurements have been made* [1]. One case is presented on the next pages.

2. A MODERN CONCEPT FOR OFFICE BUILDINGS

2.1. High Insulation Windows

A few years ago, Geilinger Ltd in Winterthur, Switzerland, has developed a new window system which is called *High Insulation Technology (HIT) window.* Glazing-U-values of **0.65 W/m²K** are typical. The basic idea is to minimize all possible heat transfer mechanisms. Considering the glazing, this means to keep the heat conduction by air as well as the convective and the radiative (infrared) transfer rates at low levels.

Measurements and calculations have shown, that the surface temperature of the inner glas pane of a HIT glazing unit is only slightly below the mean indoor air temperature [2]. This gives a rather symmetrical radiation field and only a small deviation of the radiative temperature from the inside air temperature. A second effect of the small difference between room air temperature and surface temperature of the window is the absence of downdraft. As a matter of fact, HIT-buildings do not need radiators at all. (The small amount of heat, which has to be supplied at cold winter nights can be transported by air).

2.2. Mechanical Ventilation and Displacement Principle

Present office work is characterized by relatively high internal heat loads from equipment (computers, copying-devices) and lights. Typical values range from 20 to 50 W/m². It will be shown in the following, that this covers the heat transmission loss, provided that the whole façade is highly insulated. During winter time working hours, the mechanical ventilation has only one task: to supply the office user with fresh air. Heat recuperation is used in the ventilation system for energy saving. During the warmer periods of the year excess heat may be removed by the ventilation system (other means are also possible). But still, the maximum *air exchange rate can be reduced considerably* compared to earlier requirements. This point is of major importance since investment and operating costs are determined mainly by the rate of air changes.

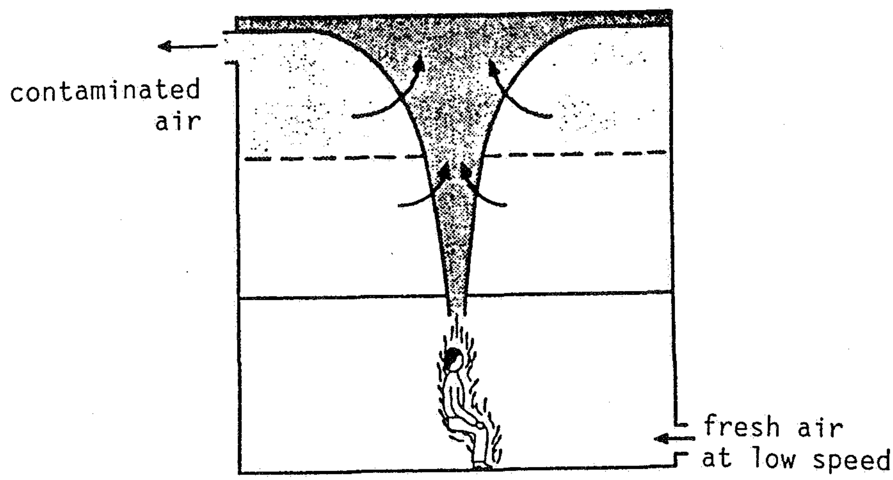

Figure 1: Displacement ventilation. Schematical picture of the air flow with air supply at the bottom, user in the fresh air zone, natural convection of polluted air, entrainment of ambient air, upper zone with polluted air, outflow near the ceiling.

Since the early eighties, a new type of air flow in rooms has been developped in the Skandinavian contries. Fresh air is introduced at low velocity near the bottom. A stable stratification from the bottom to the top devloppes (figure 1). Yet the temperature difference in the occupied zone remains small. The polluted air is removed from the source by natural convection to the polluted zone at the top of the room and from there to the outlet near the ceiling. This system is called *"displacement ventilation"* [3,4]. It combines the advantage of low air change rates and a high level of comfort: fresh air comes directly to the zone where it is needed and draft does not occur.

2.3. Control Concepts

Modern concepts of building management include control facilities which precisely match the features of the HVAC system, the heat production in the offices and the heat transfer through the envelope.

An optimum in comfort and energy consumption may be attained with a hierarchical system ending up with a variable, room individual air volume control. During winter days, heat recovery is applied. At night, the ventilation system is kept in operation only, if heat must be transported: either supplied (cold winter night: internal circulation) or removed (summer night: free cooling).

3. CASE STUDY: ONE YEAR OF REGULAR PERFORMANCE OF A HIT-PROJECT

Detailed investigations were made in 1988 at an office building in Winterthur, Switzerland.

3.1. A conventional building extended with a HIT-part

Up to the fourth floor, the building is built in a conventional way (insulated double glazing windows), with natural ventilation. On top of this building, two floors were erected in 1987 based on the HIT concept (high insulation windows produced by Geilinger Ltd., displacement ventilation with roomindividual control). Geometry is identical for the new and the old part. All windows can be opened and effective sun shades can be lowered individually. - The situation is ideal for comparisons.

3.2. Mechanical ventilation

The ventilation system of the HIT-part of the building operates with temperature demand controlled variable air volumes of 1.0 ... 5.0 air changes per hour. The outside air is heated up to 19°C by heat recuperation and by a water heater (connected to the central heating system of the older part of the building). On top of this, the air can be heated by a room individual electrical unit, if necessary. During summer nights, the excess heat is removed by free cooling. Since the system is not equipped with a cooling unit, the mechanical ventilation is turned off at outside temperatures over 22°C in order to prevent unwanted heating. Users are then recommended to make a reasonable use of the openable windows.

3.3. One year of observations

The energy consumption was investigated by weekly readings of different meters for electricity, heat supply and operating hours.

Further, two campaigns were conducted during winter 1987/88 and summer 1988, respectively. 30 parameters were recorded at time intervalls of 10 minutes. The objectives were:

- user comfort in a HIT room with displacement ventilation and comparison with a conventional room
- efficiency of free cooling
- suitability of the control concept of the ventilation system
- monitoring of the mechanical ventilation

Both rooms (HIT and conventional) were oriented to the east.

4. USER COMFORT DURING THE SUMMER

4.1. Comparability of measurements under real conditions

The temperature and energy performance of a room is up to a certain degree influenced by the user behavior and the weather. In general, with high insulation envelopes the overall energy consumption is very low, but the relative importance of the users influence is more important compared to conventional buildings. (This aspect has clearly been demonstrated by our computer simulations. The model calculation is based on hourly weather data for one year and includes building features as well as the user behavior.)

Nevertheless, observation of real buildings has important advantages compared to laboratory modelling (e.g. influence of thermal mass, long term run, real weather and solar radiation, users). General conclusions can be drawn and give important information to engineers and architects.

Fortunately, in summer 1988 several distinct warm weather periods occurred. The users were instructed *not* to take special care of the measurements, but to behave as they normally do. A reasonable use of sun protection and window opening could be stated in both rooms under observation. The internal heat loads were also comparable.

4.2. Room air temperatures during working hours

Figure 2 shows the statistical distribution of the room air temperatures of the HIT-room and the conventional room, respectively. The statistics includes a periode of 12 summer weeks (530 working hours). The time average doesnot give much information, but the **number of hours with "high" room temperatures** is of importance for the user comfort: there are 4 times less in the HIT-room then in the other one (t > 27°C during 19 hours in the HIT-room, 111 hours in the conventional room).

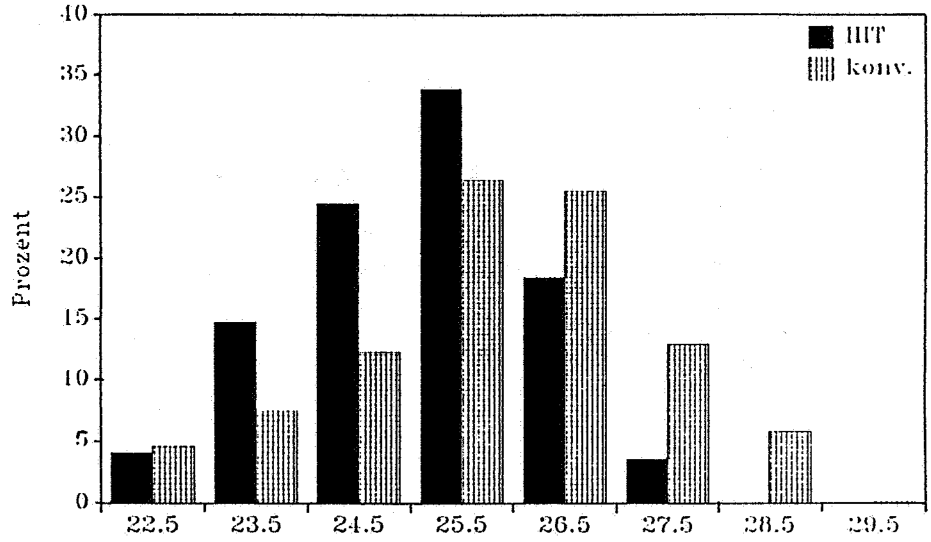

Figure 2: Statistical distribution of the room air temperatures during working hours.

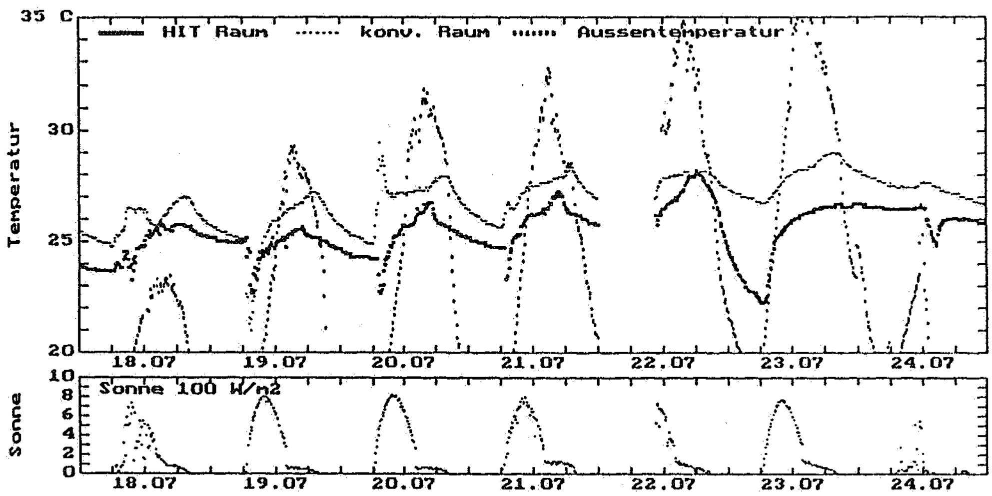

Figure 3: Air temperatures in the HIT-room (exceptionaly no free cooling), the conventional room and outside. The sun protection (louvers) were used every day. AT the bottom: solar radiation (global, vertical).

Room air temperatures during a hot summer week are shown in figure 3. Due to an operating error, the free cooling did not work during that week, which in turn reveals the dynamical characteristics of the room itself. At all times the room air temperature in the HIT-room was lower then in the other one. Quickly after the end of the working time, the room air temperature of the HIT-room began to drop, although the outside air temperature still remained higher. The reason is the effectiveness of the thermal mass (floor, walls, furniture). Contrary, the conventional room is strongly influenced by the course of the outside air temperature and can, therefore, only cool down if the temperature of the outside lies below.

The summer dynamics of a room with HIT-windows is characterized by two important factors:

- The high insulation also protects the interior against midsummer heat.
- The type of HIT-window used in the observed building show selective properties: a comparably large difference of light transmission rate (τ_{vis} = 58%) and total solar energy transmission rate (g = 42%). This results in an effective protection against the unwanted solar heat and, at the same time, an efficient day light.

Generally speaking, the HIT concept means that *inside and outside climate are separated as much as possible.*

4.3. Free Cooling

A mechanical ventilation system offers the opportunity to run a free cooling mode during summer nights: the excess heat, which has been accumulated in the thermal masses during the day, is being removed by the cold nocturnal air. Figure 4 demonstrates the effect of free cooling during a warm week at the end of August 1988: the temperature drop in the HIT-room is quite distinct.

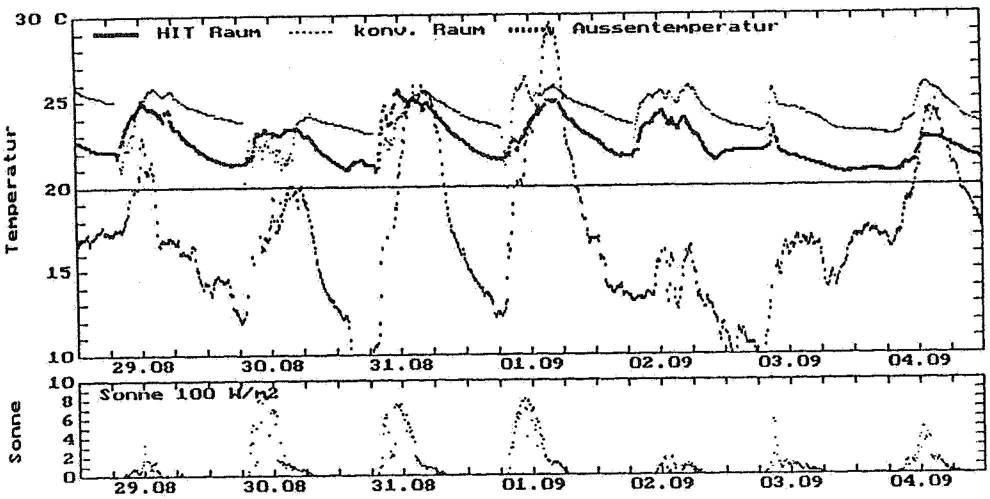

Figure 4: Air temperatures in the HIT-room (with free cooling), the conventional room (no mechanical ventilation) and outside. The sun protections were used every day. At the bottom: solar radiation (global, vertical).

Moore insight into the cooling process can be gained through the calculation of the energy transport. The night of the 1./2.9.88 is selected as a typical case (table 1). The calcuations are based on the measurement of the air temperatures and velocity in the air ducts (energy transport by free cooling) and the air temperatures in the rooms and outside (transmission). With transmission and free cooling, an amount of 10864 kJ was removed from the HIT-room. This means a drop of the floor temperature of around 0.7 K. Besides, the case shows that free cooling (9102 kJ) is more efficient than the cooling by natural transmission (7373 kJ) in a conventional room.

	HIT-room	conv. room
transmission	1762 kJ	7373 kJ
free cooling	9102 kJ	-
total	10864 kJ	7373 kJ

Table 1: Removal of the excess heat from the two observed rooms, both with 24 m^2, during the night of 1./2.9.88, from 20.00 to 05.00.

In this case, the energy consumption of the ventilators was 6000 kJ (related to the office room). This amount should be seen as part of the total, extremely low energy consumption (see chapter 6).

5. WINTER PERFORMANCE

5.1. Working hours

During working hours, HIT rooms are supplied with fresh air at a temperature of 19°C. Normally, the air change rate is at a minimum rate of 0.6 to 1.0 air changes per hour. In case of excess heat, the air volume is increased.

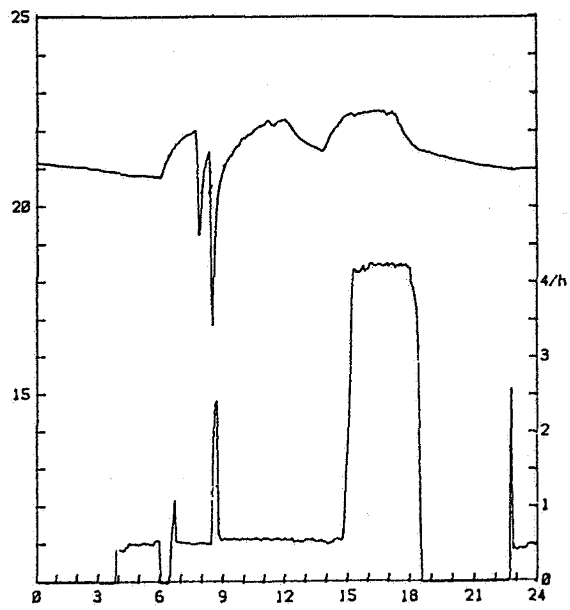

Figure 5: Room air temperature and air change rate of the mechanical ventilation during an overcast winter day (13.1.88).

Figure 5 shows the room air temperature and the air change rate of the mechanical ventilation during an overcast winter day with outside temperatures around 0°C. The user obviously left his office several times for shorter or longer intervalls (temperature drops). In the afternoon, the ventilation reacted to the room air temperature of 22°C with an input of the maximum air volume rate. This means that a larger amount of heat is removed (cooling).

Regarding the user comfort, the advantages of the HIT-concept during the winter working time may be summarized as follows: the surface temperatures of the HIT-windows are only a few degrees below the temperatures of the interior walls and the room air temperature. Consequently, the thermal radiation field is well-balanced: an important factor for comfort. As a second effect, the surface boundary layer at the window has approximately the same temperature as the room air. Therefore, downdraft does not develop and radiators below the window are not necessary. The comfort gain of the displacement ventilation results from the constant supply with fresh air without any draft (low air speed *and* low turbulence degree).

5.2. A HIT-room during winter nights

During moderately cold winter nights, a HIT-room needs not to be heated at all, even if no internal loads are present. An example is given in figure 6. The mechanical ventilation was turned off from 18.30 to 06.30. During this time the mean room air temperature dropped about 0.5°C. Totally different is the behavior of the conventional room. There, the radiator was on during the whole night (at a reduced level). The temperature dropped to 19°C.

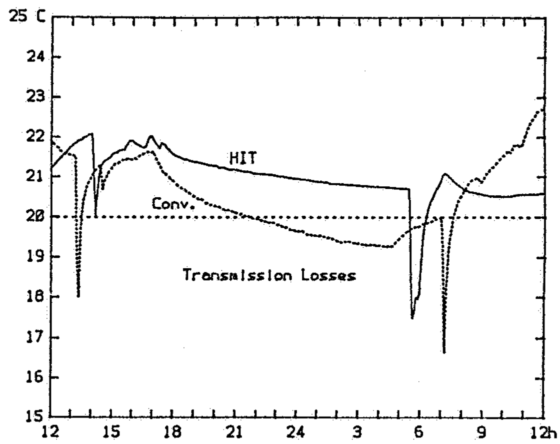

Figure 6
Room air temperature during a winter night in the HIT room and the conventional room, respectively. The mechanical ventilation of the HIT room was off from 18.30 until 06.30. The conventional room was heated at a low level by the radiator.

During very cold and clear winter nights, the situation is slightly different. Computer simulations and practical experience led to the conclusion that a little amount of heat should be added to the HIT room. This can be done by the mechanical ventilation (internal circuit). The idea is, to keep the overall room temperature (air, floor, furniture) at a level of about 20°C in order to have optimal conditions in the morning.

6. ENERGY SAVING

6.1. One year of observation

Some results of the detailed energy readings are presented in table 2 and in figure 7.

In "lights + machines" the direct consumption by the user is given. Monthly deviations are small, which means that the portion for machines is dominant.

For the ventilators more energy has been consumed in summer than in winter. This is partly due to free cooling.

Most of the "heat" energy has been used for central air pre-conditioning. In other words: heat is not used to cover the transmission losses, but for the fresh air supply of occupants.

	MJ/m²a	%
lights + machines ventilators heat	150 64 118	45 19 36
total	332	100

Table 2: Energy consumption 1988 at the 5. floor (HIT)

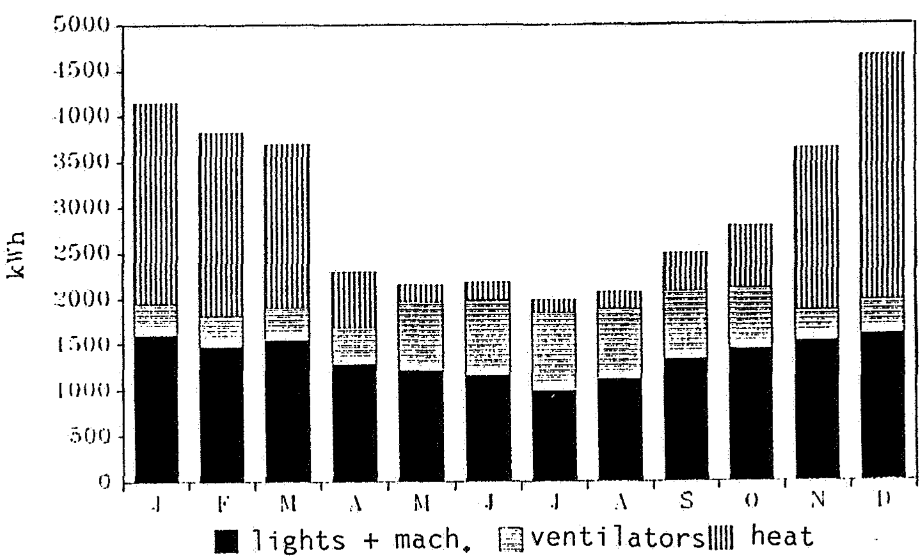

Figure 7: Monthly energy consumption at the 5. floor (HIT).

The most impressing feature is the extremely low total energy consumption, and besides, the small fraction (36%) used for heat. This is demonstrated with the comparisons in table 3.

	heat MJ/m²a	%	total energy MJ/m²a
average 1988 (*)	575	70	825
nominal value (**)	240	58	415
HIT 5. floor	120	36	332
HIT 5. + 6. floor	170	47	360

Table 3: Consumption of heat and total energy.
(*) Existing swiss buildings (buildings with heavy deficiencies are excluded), [5].
(**) based on the recommendations of the Swiss Society of Engineers and Architects (SIA), [6].

Although conditions for the extension of the reported building were not optimum (certain heat leaks such as outside columns could not be avoided), the energy consumption is far below the nominal value for Switzerland.
At the 6th floor more heat and total energy was consumed than at the 5. floor: reasons are the 67% larger transmission loss factor due to the bulding roof and an accidentally lower consumption of electricity.

6.2. Energy balance during winter

During three winter months (Dec. 87 - Feb. 88) with an avarage outside temperature of 2.8°C, detailed energy measurements were made. Gains and losses were calculated. Comprehensive energy balances for the fifth floor (HIT) and the third/fourth floors (conventional) resulted and are presented in table 4.

Gains/Input	HIT	conv.	
Sun + occupants	2.3	4.0	W/m²
Light + machines	5.5	7.8	W/m²
Heat	10.1	21.0	W/m²
Total	16.9	32.8	W/m²

Losses	HIT	conv.	
Transmission	7.1	21.4	W/m²
others (*)	6.2	-	W/m²
Air	3.6	11.4	W/m²
Total	16.9	32.8	W/m²

Table 4: Energy balance for the periode Dec. 87 - Feb. 88.
(*) Losses due to thermal leaks inherent to the existing building design. For comparison purposes "Transmission" values should be considered only.

Two important conclusions can be drawn from these measurements:

1. *A high insulation office room obtains enough heat energy by machines, lights, sun and occupants to balance the thermal transmission losses.*

2. If an adequate ventilation concept is used, ventilation losses are much lower compared to natural ventilation (crack flow and window opening).

7. CONCLUSION

The High Insulation Technology (HIT) - concept is based on decoupling the inside thermal climate from the outside climate by a high insulation building envelope. This provides a good radiative climate by saving large amounts of energy. The air quality is maintained at a high comfort level running a mechanical ventilation based on the displacement principle. Excess heat can to a certain extent be removed by the mechanical ventilation: free cooling can be recommended.

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PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

10th AIVC Conference, Dipoli. Finland
25-28 September, 1989

Poster 7

AIR CHANGE IN FLATS WITH NATURAL VENTILATION: MEASUREMENTS AND CALCULATIONS

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SYNOPSIS

The air change rate in existing, older blocks of flats having natural ventilation has been measured by the tracer gas method. Measurements were made in the buildings in as-found condition. The average air infiltration rate was 0.26 air changes/h (with the ventilation ducs closed), with an overall ventilation rate of 0.47 air changes/h. The average overall ventilation rate is very close to that recommended on the basis of health requirements, although values both above and below this are encountered in many flats.

A comparison has been made between measured air change rates and calculated air change rates using the JK-CIRCUS computer program. The results show that airtightness of wall and windows in older flats is equivalent to an average n_{50}-value of 1.5 air changes/h. n_{50} is the air change rate at a differential pressure of 50 Pa between interior and exterior. The building regulations specify an n_{50} air change rate of less than 1.0 air changes/h in new blocks of flats.

Calculations using the JK-CIRCUS program show that, for a flat having a natural ventilation system, improving the airtightness of the windows will result in a maximum reduction in air change rate of 0.1 air changes/h for an external wind speed range of 2-8 m/s.

LIST OF SYMBOLS

A	Admittance	m^3/sPa
a	Flow rate coefficient	m^3/sm^2Pa^b
B_o	Fluid permeability coefficient	m^2
b	Flow exponent	-
h	Chimney height	m
n	Air change rate	h^{-1}
p	Pressure	Pa
q	Air flow rate	m^3/s
u	Wind speed	m/s
ε	Surface roughness	m
η	Dynamic viscosity	Ns/m^2
θ	Temperature	K
λ	Friction factor	-
ν	Kinematic viscosity	m^2/s
ξ	Loss factor	-
ρ	Density	kg/m^3
μ	Form factor (wind)	-

1. AIR CHANGE RATE MEASUREMENTS - VENTILATION IN FLATS

A number of Swedish blocks of flats, built between 1860 and 1960, have been investigated in respect of various behaviourial characteristics, including air change rates. The measurements were made in as-found condition in 29 flats having natural ventilation.

1.1 Measurement procedure

The air change rate of a flat indicates how many times the air inside the flat is replaced by outdoor air per unit of time.

Air change rate measurements were made using the tracer gas method, involving the use of a gas analyzer which measures the concentration of laughing gas (N_2O) discharged inside the flat. The rate at which the gas concentration declines depends on the ventilation rate: the higher the ventilation rate, the more rapidly the gas concentration declines.

The measurement period for each flat has normally been of the order of 4-8 hours. Wind speed and indoor/outdoor temperature difference play a part in determining the ventilation rate, and have therefore been noted in connection with each set of measurements. The measurements have mainly been made during the winter, when the temperature difference between the indoor and outdoor air was greatest.

The air change rate has been investigated for two different cases: infiltration and normal ventilation. Infiltration (uncontrolled ventilation) is represented by the condition encountered when all ventilation fittings are sealed so that the only ventilation resulting is that caused by air leaks through the building envelope. This is, of course, an abnormal condition, and it is not possible simply to add the infiltration rate to the intended ventilation rate. However, the measurements do provide an indication of the airtightness of the building envelope. Normal ventilation conditions are those when no ventilation fittings are sealed, but are opened as they normally should be.

1.2 Measurement results

The results of the measurements are presented in Table 1. The table shows the as-measured values, i.e. no corrections have been applied for wind or temperature differences at the time. The values have been grouped in accordance with the ages of the buildings.

Table 1 As-measured values of ventilation air change rates in flats in Malmö, Eksjö and Gävle, and having natural ventilation.

Flat no.	Year of building	Wind speed, m/s	Wind direction	Temperature difference, indoor/outdoor, K	Infiltration air changes/h	Normal ventilation, air changes/h
1	1835	0.25- 0.75	N	21-22	0.92	1.00
2	1884	0.1 - 0.3	N-NE	20	0.23	0.29
3	1894	0 - 1.3	S	23.5	0.46	0.69
4	1897	1.6 - 2.0	SE	3	0.16	0.18
5	1898	0.3	N	24	0.87	1.00
6	1898	0.5 - 0.8	E	25	0.24	0.36
7	1898	0.1 - 3.9	W	9.5	0.26	0.30
8	1899	0.2 - 1.4	W	20-22	0.29	0.54
9	1905	1.5	E	19-23	0.36	0.35
10	1920	1.2	N	17	0.35	0.60
11	1922	1.8 - 2.6	N	15	0.15	0.24
12	1927	0.1 - 0.3	W	17.5-23	0.16	0.23
13	1927	0.4 - 1.1	W	20-25	0.38	0.55
14	1930	2.3 - 2.6	E	17	0.22	0.38
15	1934	2.2 - 3.8	SE	13	0.17	0.33
16	1934	2.3 - 2.7	S	21	0.21	0.50
17	1938	2.0 - 2.8	SE	18.5	0.27	0.47
18	1938	1.2 - 1.5	SE	13-19	0.29	0.69
19	1938	0.8	SE	26	0.16	0.53
20	1939	1.1 - 1.6	SW	22.5-23.5	0.09	0.64
21	1941	1.0 - 1.5	NE	23-24	0.23	0.92
22	1945	0.6	W	29-33	0.25	0.71
23	1946	1.0 - 1.2	SW	19-20.5	0.12	0.24
24	1946	1.6	N	19	0.12	0.21
25	1948	0.8 - 1.7	S	13-15	0.03	0.11
26	1949	2.2 - 2.4	SW	24	0.14	0.42
27	1950	1.3 - 2.0	SE	22.5-26	0.10	0.63
28	1954	0.8 - 1.2	N	20-21.5	0.06	0.35
29	1955	1.5 - 3.0	W	25-26	0.15	0.26
Mean value					0.26	0.47

The mean value of infiltration is 0.26 air changes/h, while that of normal ventilation is 0.47 air changes/h. This mean value of normal ventilation is quite close to the value recommended on the basis of health requirements (0.5 air changes/h), although there is a wide spread, and many flats have either too high or too low ventilation rates. This can also be seen clearly in Figure 1, which shows histograms of infiltration and normal ventilation air change rates.

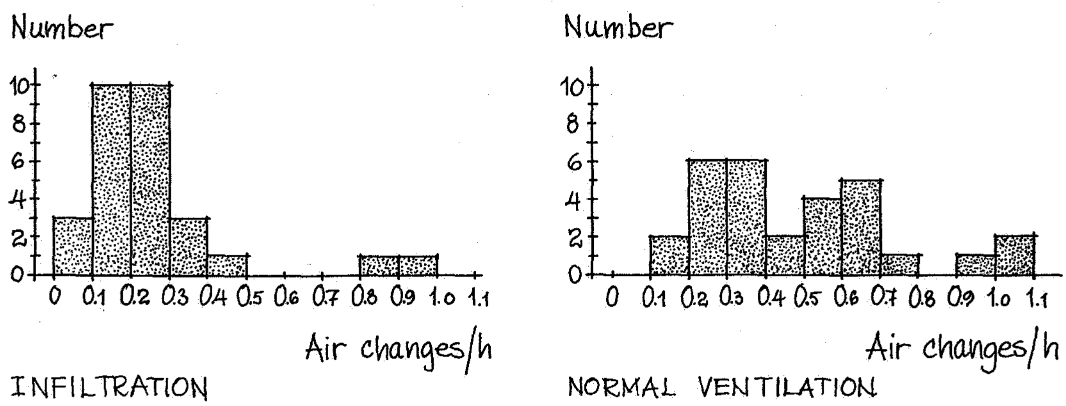


Figure 1 Histograms of as-measured infiltration and normal ventilation air change rates for 19 flats having natural ventilation.

Table 1 shows that the air change rate varies widely from one property to another, but that the ages of the properties do not seem to have any direct effect on normal ventilation. The ages do, on the other hand, seem to have some effect on infiltration. In order to verify this, the relationship between the year of building and ventilation rate has been calculated using the method of least squares. For infiltration, the coefficient of correlation is so high (0.73) that it does seem reasonable to postulate the relationship. This relationship means that infiltration tends to increase with increasing age of buildings, while normal ventilation is not affected by the ages of the buildings, for which the coefficient of correlation is low at 0.30.

2. COMPUTER CALCULATION OF VENTILATION

The measured values of normal ventilation rates in naturally-ventilated flats have been compared with theoretical values, as calculated by computer using the JK-CIRCUS program.

2.1 The JK-CIRCUS program

This program, developed by J. Kronvall[1], performs the following calculations:

- divides the flow geometry up into finite parts - components,
- calculated the admittance, defined below of each component,
- calculates the potentials, p (Pa), at all nodes, together with the flow rates, q (m^3/s), through all components.

In the case of (air) flow problems, a component may be either:

- a pressure difference between two nodes (active component),
- a piece of permeable material (passive),
- a section of duct, in the flow direction (passive),
- a single resistance, e.g. entrance, exit or bend loss (passive),
- a potential flow $q = a \cdot \Delta p^b$ (passive).

The principle is that the air flow in the flat is described by means of a network or flow diagram, in which the various parts are referred to as components, in a manner analogous to that for an electric circuit containing a network of resistances. Each component in the flow diagram having two connections is referred to as a branch, while the connection points are referred to as nodes. Several branches can be connected to the same node.

The designations used for parts of materials or ducts are illustrated in Figure 2.

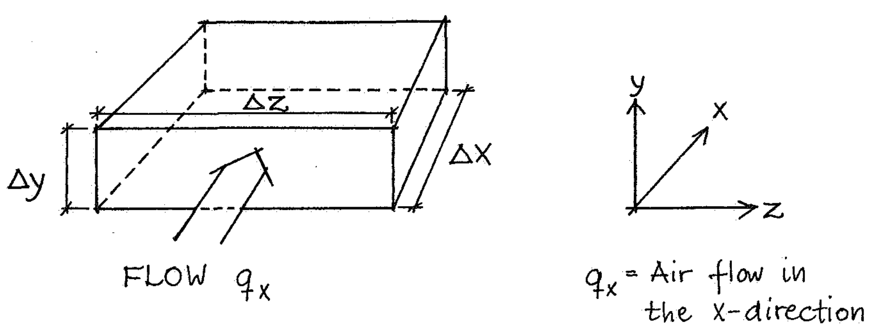


Figure 2 Definition of geometrical quantities of a component.

The computer program works with admittances. The admittance, A, of a component is defined as:

$$q_x \ (m^3/s) = A_x \ (m^3/sPa) \Delta p_x \ (Pa)$$

where Δp_x = the pressure difference across the component in the direction of flow (Pa)

According to Kronvall[1], the admittance, A, of permeable material is given by:

$$A_x = \frac{B_o}{\eta} \frac{1}{\Delta x} \Delta y \ \Delta z$$

where B_o = permeability (m^2)
η = the dynamic viscosity of air (Ns/m^2)

while, similarly, the admittance, A, of crevices and ducts is given by:

$$A_x(q) = \frac{4\Delta y^3}{\rho q_x} \frac{\Delta z^2}{\lambda \Delta x}$$

where ρ = the density of air (kg/m^3)
λ = the friction factor.

In general, the admittance of a duct depends on the flow, q, through it. With laminar flow (Reynold's number $Re < 2300$), the friction factor, λ, is inversely proportional to the flow,

$$\lambda = \frac{96}{R_e} = \frac{96 \cdot \Delta z \nu}{2q}$$

which means that the admittance is constant.

Further, the admittance, A, of a single resistance is given by:

$$A_x(q) = \frac{2\Delta y^2}{\rho q_x} \ \frac{\Delta z^2}{\xi}$$

where ξ = the loss factor.

The admittance, A, of the potential flows is given by:

$$A_x(\Delta p) = a \Delta y \ \Delta z \ \Delta p^{(b-1)}$$

where a = flow coefficient ($m^3/s \ m^2 Pa^b$)
b = flow exponent

2.2 Prerequisites

The computer calculations were made for a naturally-ventilated flat having a floor area of 80 m². The flat has two opposing exterior walls, and is situated in the centre of multi-storey building, as shown in Figure 3. Ventilation is provided by outlets in the kitchen and in the bathroom, both connected to the main ventilation riser, the height of which is 8 m.

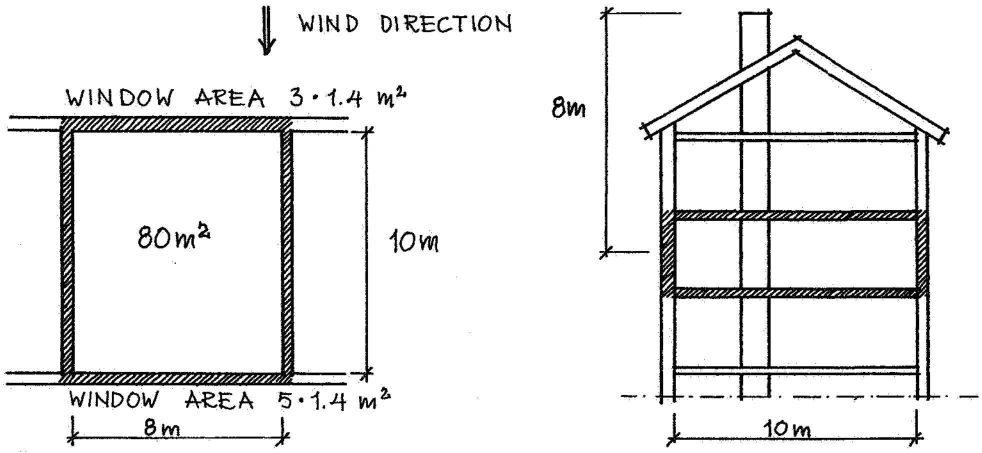


Figure 3 Dimensions and position of the flat considered in the computer calculation

The network for computer calculation of the above flat is illustrated in Figure 4:

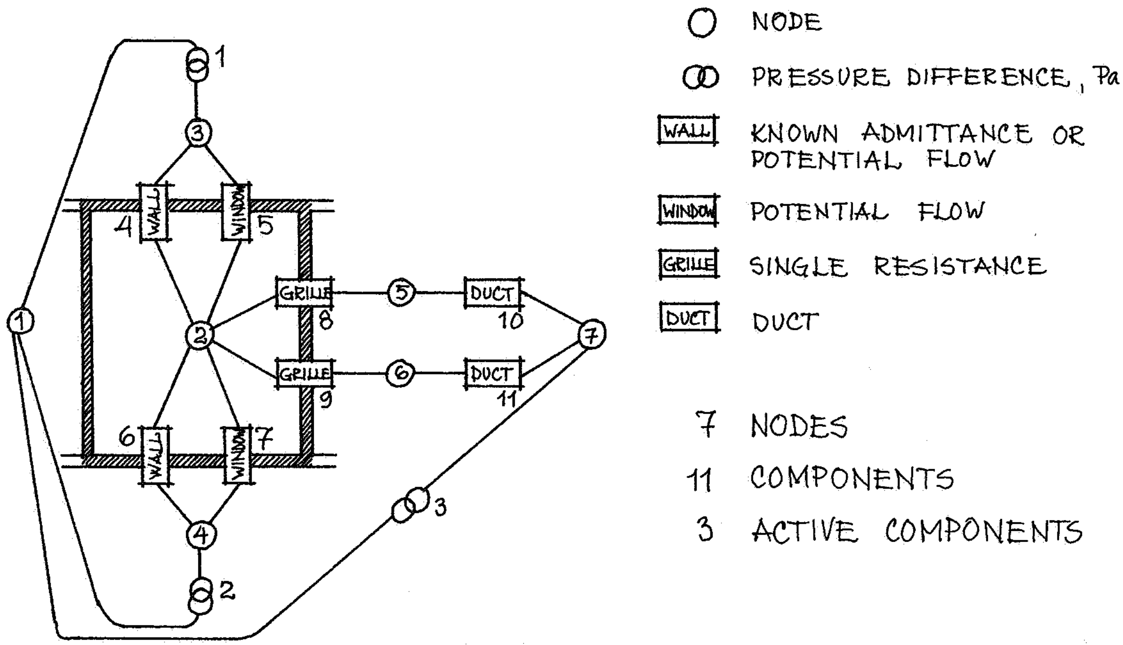


Figure 4 Computer network model of the flat shown in Figure 3.

Wind pressure has been calculated on the basis of a wind velocity of 4 m/s. Form factors of 0.7 and -0.5 respectively have been assumed for the windward and lee walls, equivalent to values assumed for wind loads in design standards. A form factor of -0.4 has been assumed for the chimney opening, as given by Liddament[2]. The wind pressure, p, can be calculated from the formula:

$$p = \mu \rho u^2/2 \quad \text{(Pa)}$$

where μ = the form factor
ρ = the density of the air (kg/m^3)
u = the wind velocity (m/s)

The difference in indoor and outdoor air temperatures results in the chimney or thermal effect, by which a difference in air pressure arises in the indoor and outdoor air. The differential pressure, Δp, can be calculated from:

$$\Delta p = 0.043\ h\ \Delta\theta \quad \text{(Pa)}$$

where h = chimney height (m)
$\Delta\theta$ = indoor/outdoor temperature difference (K)

For the purpose of the calculations, the thermal driving force has been assumed to be 8 Pa, equivalent to a temperature difference of about 23 K between the indoor and outdoor air with a chimney height of h = 8 m.

The facade surfaces are influenced both by wind pressure and thermal effects, while the top of the chimney is influenced only by wind pressure.

The ventilation fittings in the kitchen and bathroom are assumed to be of the grid type, and to have a size of 0.15 m x 0.15 m. The loss factor, ξ, has been assumed to be 10. The cross-sectional area of the ventilation duct from the kitchen is 0.125 m x 0.250 m, while that of the duct from the bathroom is 0.125 m x 0.125 m, both with lengths of 8 m. Surface roughness, ε, is assumed to be 0.01 m.

Computer calculations have been made for airtight windows, for which the airflow through them at a pressure difference of 50 Pa (q_{50}) is 1.7 m^3/m^2h, i.e. the value as assumed in the Building Regulations for new windows. However, if we consider the values of actual measurements of air leakage around older windows, Olsson-Jonsson[3], we find that few windows are actually as airtight as this. Calculations have therefore also been made for less airtight windows, for which q_{50} = 5 m^3/m^2h. The flow exponent, b, of windows having good airtightness has been given a value of 0.67, while for less airtight windows, measurements indicate that the value of b is around 0.80.

Walls, too, have been investigated in respect of their airtightness. The admittance of walls with a good level of airtightness has been assumed to be 10^{-10} m^3/sPa, i.e. around zero in practice. In the case of walls that are less airtight, there will be a potential flow through them. The flow coefficient, a, has been calculated so that the air change rate in the flat for an interior/exterior pressure difference of 50 Pa (n_{50}) is 1.5 air changes/h with poorly airtight windows as above. The flow exponent is assumed to be 0.7.

2.3 Results

Figure 5 shows the air flows into and out of the flat, and the air change rate, as calculated by the computer.

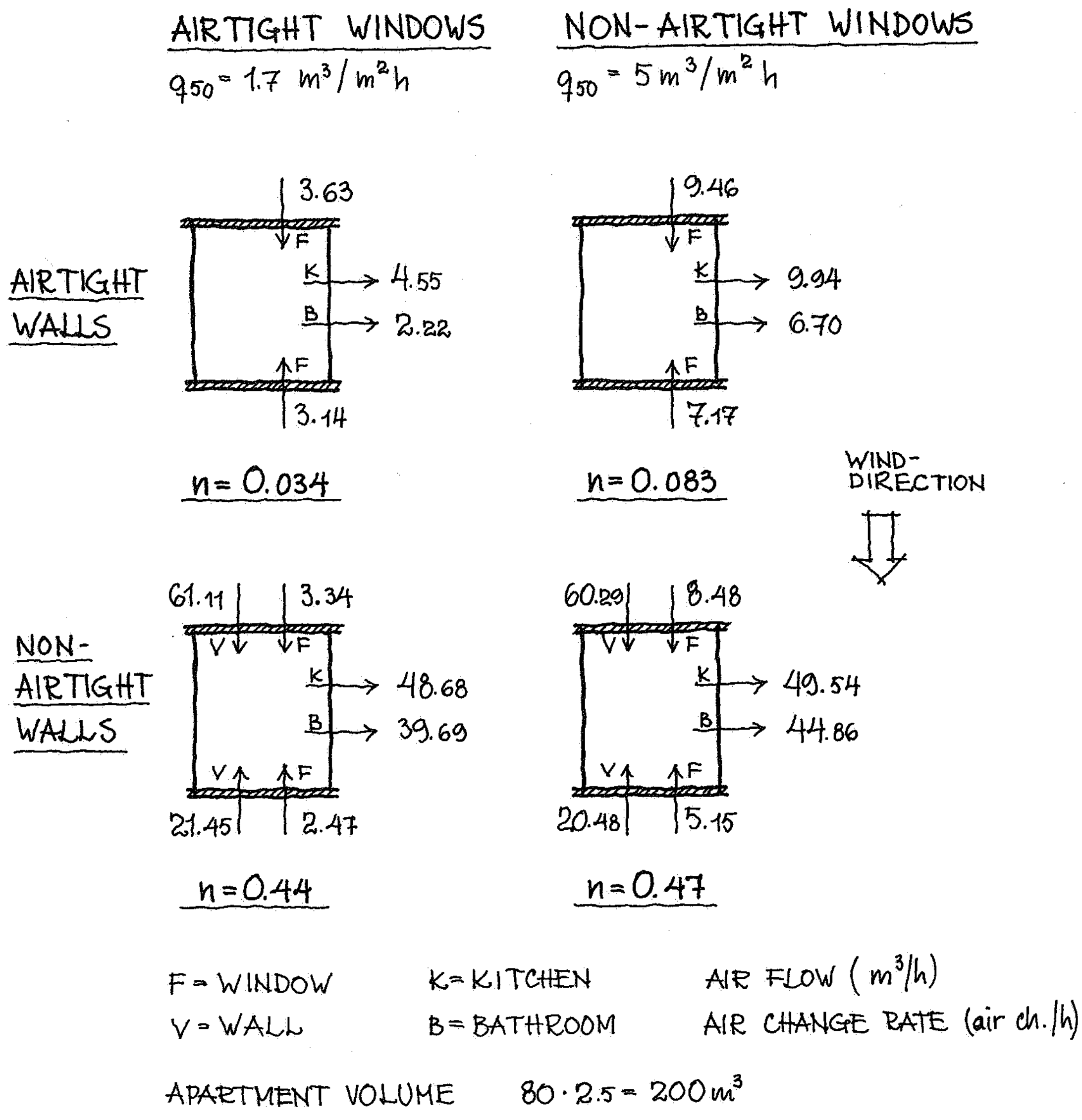


Figure 5 Results of the JK-CIRCUS computer calculation for a naturally-ventilated flat in a block of flats. Wind velocity = 4 m/s.

The mean value of as-measured normal ventilation in the naturally-ventilated flats is 0.47 air changes/h, i.e. approximately the same value as shown in the figure above for poorly airtight walls and windows. It therefore seems reasonable to assume that older flats have approximately the airtightness of walls and windows as used in the calculation case, and for which the air change rate at a differential pressure of 50 Pa (n_{50}) is 1.5 air changes/h. Putting it another way, this means that the airtightness of walls and windows in older flats is such that the n_{50}-value is about 1.5 air changes/h, which should be compared with the standard n_{50}-value for a block of flats of 1.0 air changes/h.

It can be seen from the figure that the air change rate in the flat is very little affected by the airtightness of the windows. This means that, if a badly-fitting window as above is weather-stripped so that its airtightness at a differential pressure of 50 Pa meets the Building Regulations requirements, then the air change rate in the flat would be reduced by only 0.05 air changes/h or 0.03 air changes/h respectively, depending on the airtightness of the walls. For windows with even poorer airtightness, q_{50} = 10 m^3/m^2h, the air change rate would be 0.16 air changes/h if the walls were airtight and 0.52 air changes/h if they were not. Weatherstripping the windows to meet the requirements of the Building Regulations in this case would result in reductions of air change rates of 0.13 air changes/h and 0.08 air changes/h respectively, i.e. somewhat more than before, but nevertheless not very much.

The air change rate is therefore largely dependent on the airtightness of the walls. That this is so is due to the fact that the wall area is large in proportion to the window area, with the result that a wall with poor airtightness will have more effect than a window with poor airtightness.

For the flat in the worked example, the air change rate at a differential pressure of 50 Pa, n_{50}, can be expressed as a function of the airtightness of the walls and of the windows as:

$$n_{50} = 0.144\ Tv + 0.056\ Tf$$

where Tv = airtightness of the wall (m^3/m^2h)
Tf = airtightness of the window (m^3/m^2h)

It can be seen from the expression that any change in wall airtightness will have a greater effect on the air change rate than an equal change in window airtightness.

3. THE EFFECT OF OUTDOOR CLIMATE ON AIR CHANGE RATE - COMPUTER CALCULATIONS

3.1 Prerequisites

The ventilation in naturally-ventilated buildings depends on current wind and temperature conditions. Wind pressure and chimney effect vary with wind velocity and indoor/outdoor temperature difference respectively.

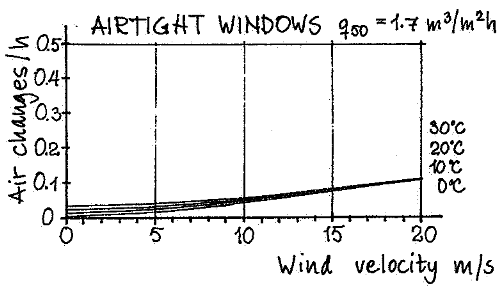
In order to investigate how wind and temperature influence ventilation, the air change rate in the flat illustrated in Figure 3 has been calculated using the JK-CIRCUS program for different wind velocities and different indoor/outdoor temperature differences. Other data are the same as before.

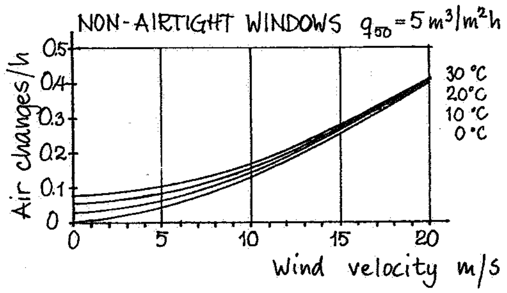
3.2 Results

The results of the calculations are illustrated in Figure 6. It can be seen that, at low wind velocities, it is the temperature difference (chimney effect) that has the greatest effect on air change rate. The greater the temperature difference, the higher the ventilation air change rate. At high wind velocities, on the other hand, the wind pressure becomes dominant, with the air change rate being almost independent of temperature difference.

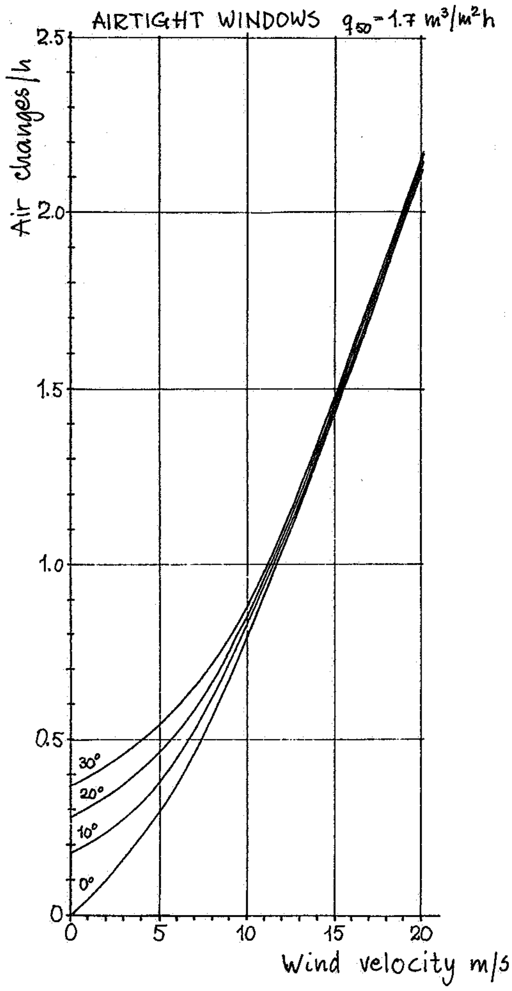
Poorly fitting windows increase the air change rate by up to 0.1 air changes/h when the wind velocity is low, i.e. less than 10 m/s. At higher wind velocities, poorly fitting windows result in increases in air change rate of 0.15-0.3 air changes/h, depending on the airtightness of the walls.

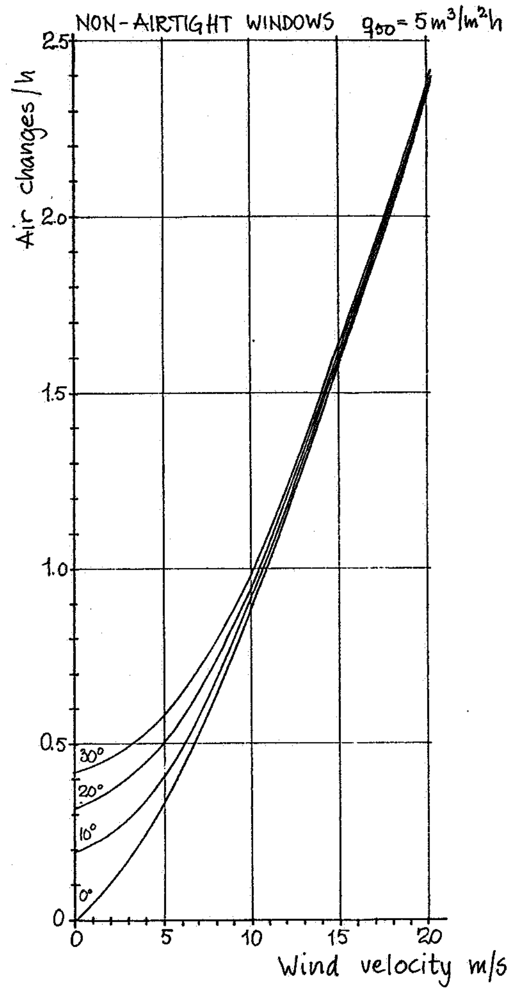
If we consider the routes taken by the airflows through the flat, we can see that, at low wind velocities, air leaks in through both the windward and lee sides, varying slightly depending on the airtightness of the walls. This inward leakage increases with greater temperature difference. As the wind velocity increases, so inward leakage of air through the windward wall increases, but decreases through the lee wall. At high wind velocities, above 10 m/s, inward leakage occurs only through the windward wall, with air leaking out instead through the lee wall, due to the fact that the effect of wind pressure is greater than the thermal chimney effect. This results in draughts through the apartment, although air is being evacuated through the ventilation ducts all the time.





A. AIRTIGHT WALLS





B. NON-AIRTIGHT WALLS

Figure 6 The effect on air change rate of wind velocity and indoor/outdoor temperature difference for airtight and non-airtight walls and windows. Natural-draught ventilation in the flat as shown in Figure 3.

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PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

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25-28 September, 1989

Poster 8

OUTDOOR AIR INLET WITHOUT DRAUGHT PROBLEMS

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1. SYNOPSIS

This paper presents a new technique for supply of outdoor air directly through external walls into a dwelling room without any draught problems.

A new type of air inlet unit has been developed based on the experience from the study of indoor climate in the "Stockholm Project".

This Swedish experiment including six new residential buildings, where the indoor climate together with different solutions for ventilation is evaluated, shows that draught from air inlets is one of the greatest problems with bad indoor climate.

Further results show that for other ventilation aspects except draught, there are less problems in buildings with mechanical exhaust systems than in buildings with other ventilation systems.

This paper presents an air inlet principle that is integrated into the external wall construction. Measurements carried out show that a test unit works without draught problems and with a lower heat demand for cold down draught protection than with ordinary ventilators and radiators.

2. INTRODUCTION

The experience from the Stockholm Project shows that almost all types of ventilation systems have draught problems from air inlets (Ref. 1).

The new building code in Sweden has increased the demand of air changing in dwellings. The minimum rate is stated to 0.35 $l/s, m^2$ floor area. For bedrooms an additional condition states a rate of 4 l/s per bed, which in most dwellings gives a higher rate than 0.35 $l/s, m^2$. This means that there will be a higher demand on the air inlets, especially on direct outdoor air inlets, to prevent draught problems.

In Sweden it is now becoming more and more common to build apartments with only exhaust air fans due to installation costs and operation and maintenance problems.

With this background we can see that there is a great need to develop and improve the function of outdoor air inlets.

With support from the Swedish Council for Building Research (BFR), a new air inlet principle has been developed and a test unit has been measured in a climate chamber at the Swedish Institute for Building Research (SIB).

The research and development work has been carried out by a group of ventilation engineers, energy engineers and building engineers from the consulting firm AIB in Stockholm.

One of the basic ideas for the air inlet concept is to integrate the ventilation technique into the building construction.

The aim of the air inlet principle is to supply the room with outdoor air without any draught problems. It has also to be kept at a low cost construction level as there is a very low cost limit for this kind of installation in apartment buildings.

3. THE PRINCIPLE FOR OUTDOOR AIR INLET WITHOUT DRAUGHT PROBLEMS

In solutions of ventilation with ordinary air inlets the air flow has a rather high velocity which makes cold air pass through the warm convective air stream from the radiator. This cold air stream will often cause a direct draught feeling.

The cold air can also flow down to the floor level which gives cold floors. Even the cold down draught from windows will increase when it mixes with an extra cold air stream.

To reduce this kind of draught problem the basic principle for the new air inlet concept is to have a very low velocity of the incoming air flow, lower than 0.02 m/s. This will be possible by letting in the air through a large surface in the external wall.

The idea is to enable the convective air stream from the radiator to "catch" the cold air flow and mix with it. This will also increase the air stream from the radiator and improve the prevention of cold down draught from windows.

In the test unit the air inlet is made by a hard mineral woolboard (insulation board) that is placed in a frame and integrated into the wall. The surface is just a little smaller than the radiator (in this case about 0.6 m^2.) Behind the board in the wall there is an air space and a connection to the outdoor air through an ordinary slit in the facade. (The outdoor air can hereby also get a very good filtration when it is necessary in polluted areas.)

Due to the low air velocity the air flow is very uniformly distributed over the air inlet surface.

A test unit has been built after a principle construction shown in Figure 1. The test in the climate chamber has been measured in order to analyse the draught problems.

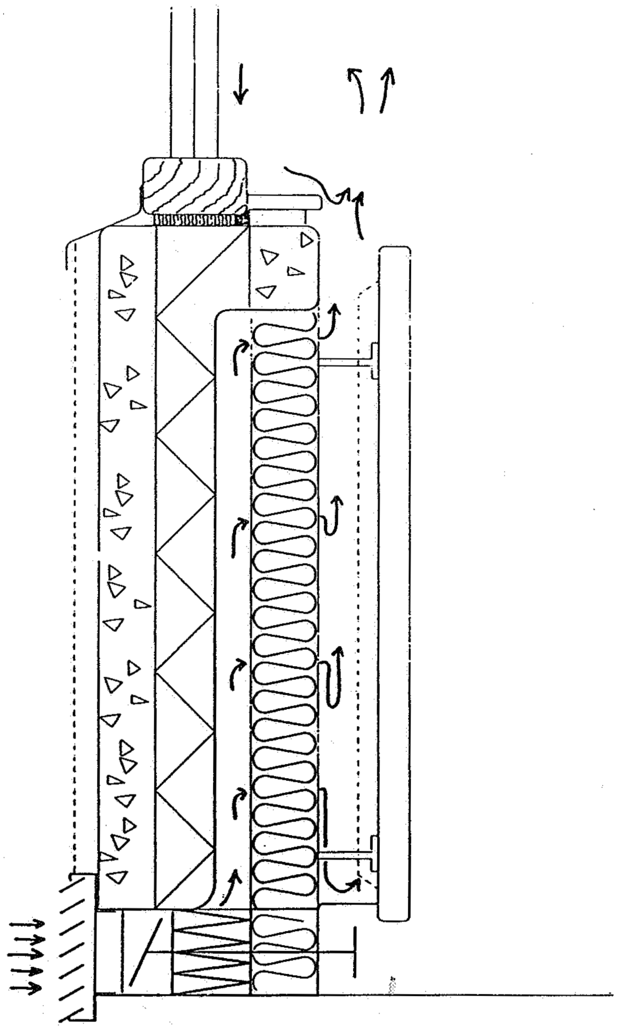

Figure 1 The figure shows the principle for the outdoor air inlet construction

4. MEASURING AND TESTING METHOD

The test unit has been tested in a climate chamber where outdoor temperatures at -20°C and ±0°C were simulated (Figure 2).

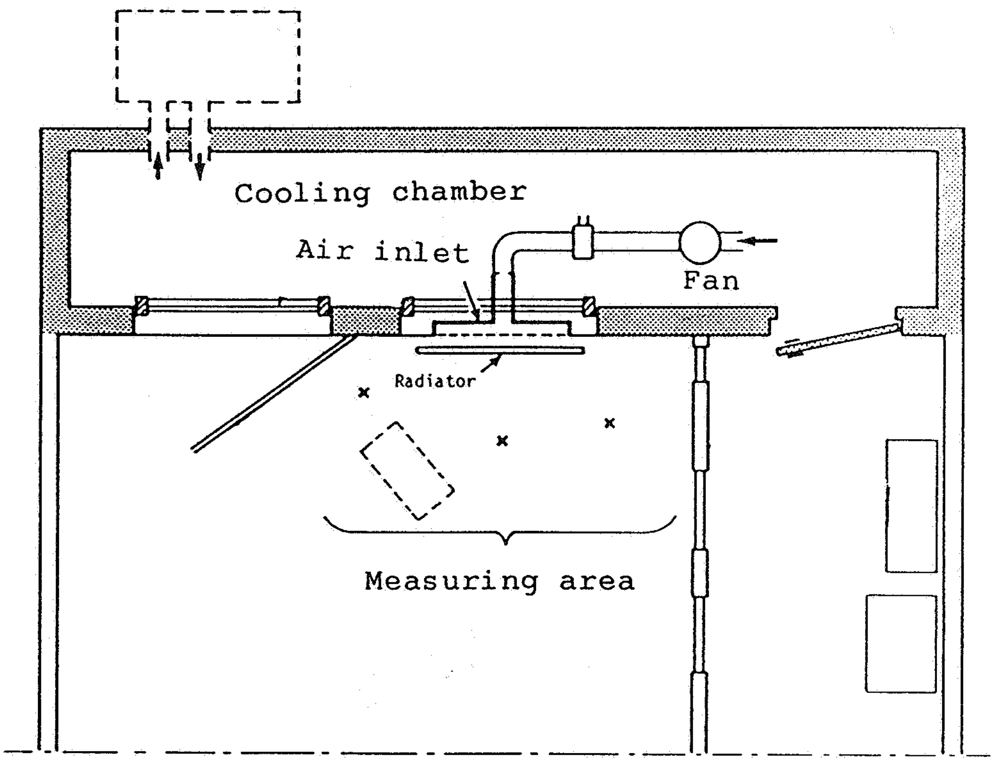

Figure 2 Climate chamber and test room at SIB

Room temperatures in the living space were registered at different levels with thermoelements. Air movements were studied with smoke and recorded on video tape.

The air velocity was also registered with an anemometer (type Disa 54N50).

The air flow through the air inlet was 25 m^3/h which could be a normal air flow for a bedroom in a dwelling in Sweden according to the new building code.

The surface area of the air inlet was 750 mm x 600 mm = 0.45 m^2. This gives an inlet velocity of 0.015 m/s.

The pressure drop over the air inlet with the air flow of 25 m^3/h was 12-14 Pa.

The heat load on the radiator was regulated with the water temperature to the radiator.

Tests were made with different heat loads on the radiator at the same outdoor temperature with the intention to evaluate at which level the draught problem was to be prevented.

The same tests were carried out to investigate the freezing risk of the radiator water if e.g. the heating system will be out of order.

5. RESULTS

The test results are shown in Figures 3-6.

The draught tests show that with the air flow of 25 m^3/h and an outdoor temperature of -20°C there is a need for 600 W heat load on the radiator to prevent cold air to flow out on the floor and cause draught problems.

If the outdoor temperature is at a level of ±0°C, a heat load of 300 W will be enough.

The experience at SIB from other tests with ordinary outdoor air inlets indicates that these results show a relatively low heating demand.

In the tests with ordinary air inlets 1000 W was normally required at -20°C, respectively 500 W at ±0°C to prevent draught problems.

In the tests to investigate the freezing risk of the radiator water it was established that there is no risk of freezing if there is a minimum basic water flow of 5 l/s through the radiator, even if the water temperature is only +20°C.

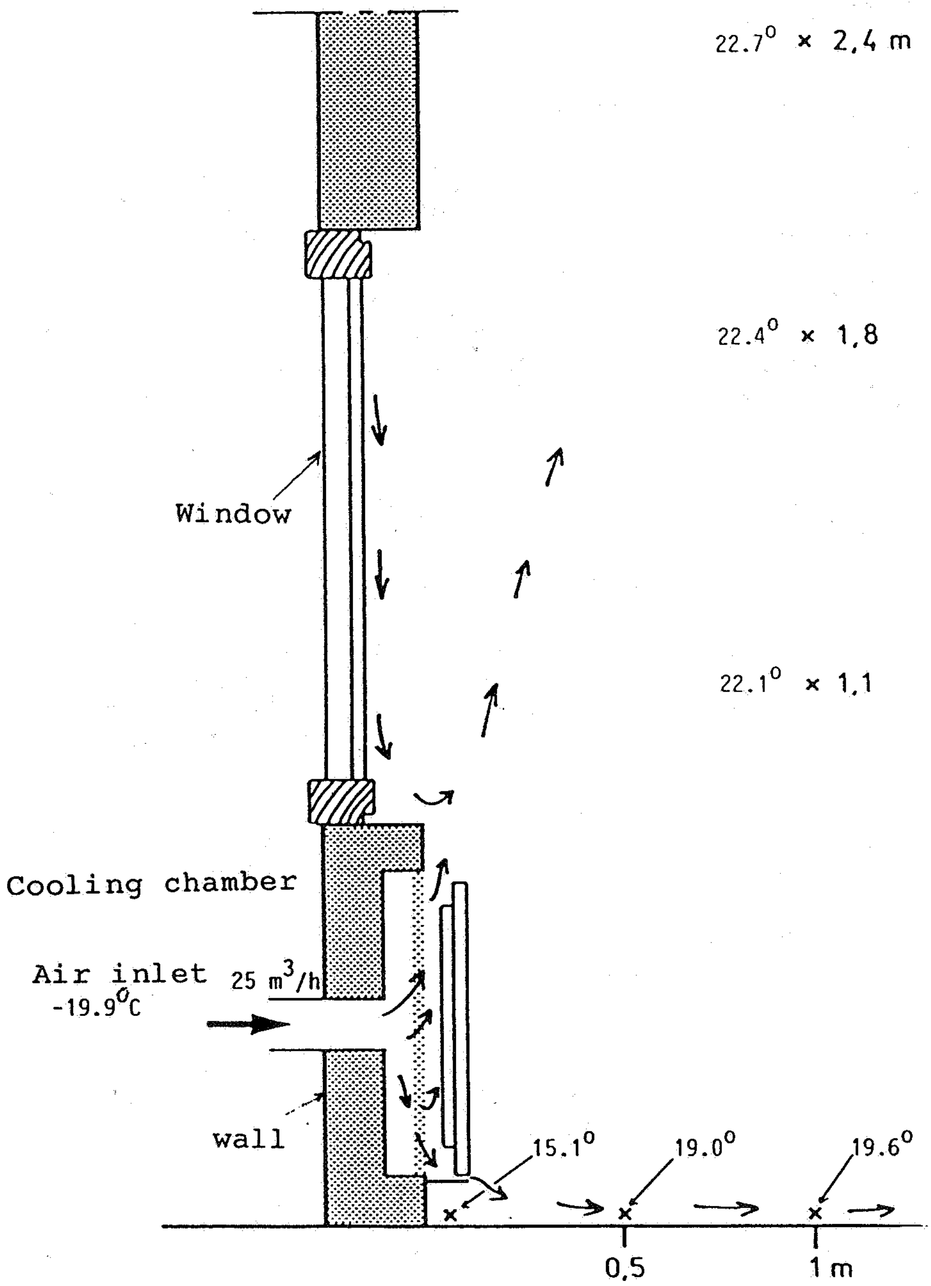

Figure 3 Test No. 1

The figure shows the air flow and the temperatures at outdoor temp. = -19.9°C and radiator heat load = 430 W.

It can be seen that the cold draught at floor level is not prevented.

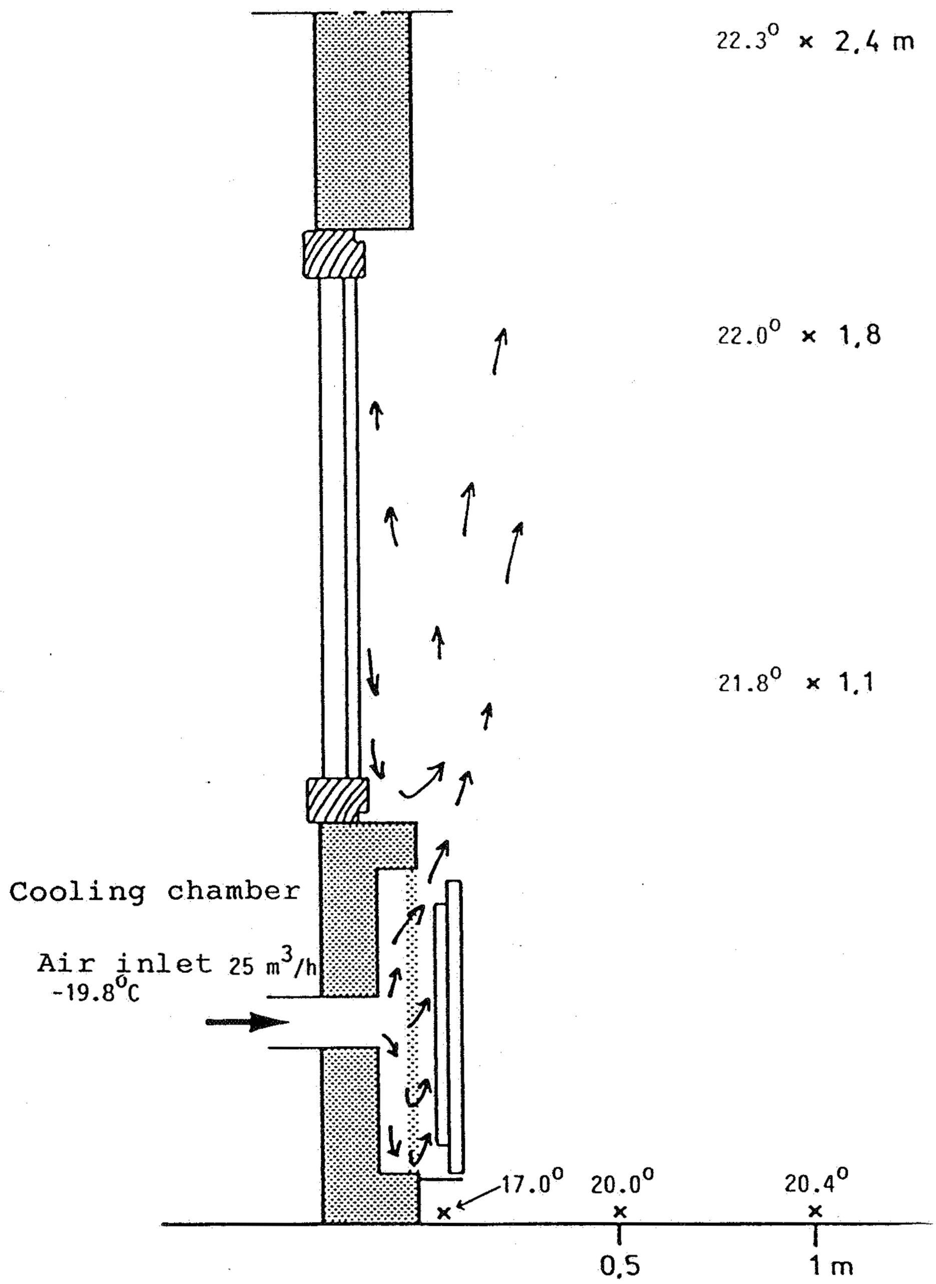

Figure 4 Test No. 2

The figure shows the air flow and the temperatures at outdoor temp. = -19. 8˚C and radiator heat load = 582 W.

It can be seen that the radiator takes care of all the outdoor air from the inlet and that there are no draught problems.

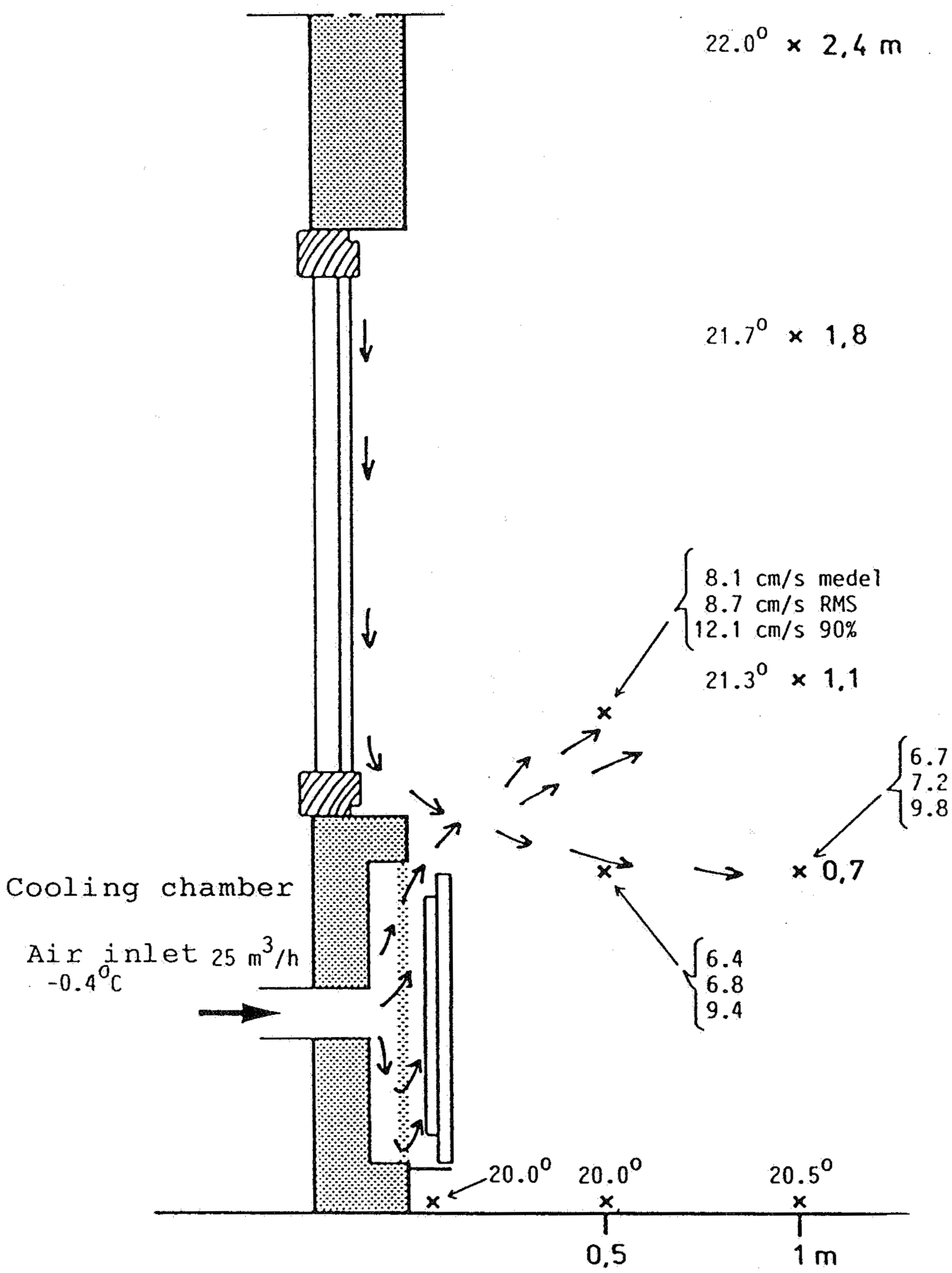

Figure 5 Test No. 3

The figure shows the air flow and the temperatures at outdoor temp. = -0.4°C and radiator heat load = 206 W.

It can be seen that there are down draught problems.

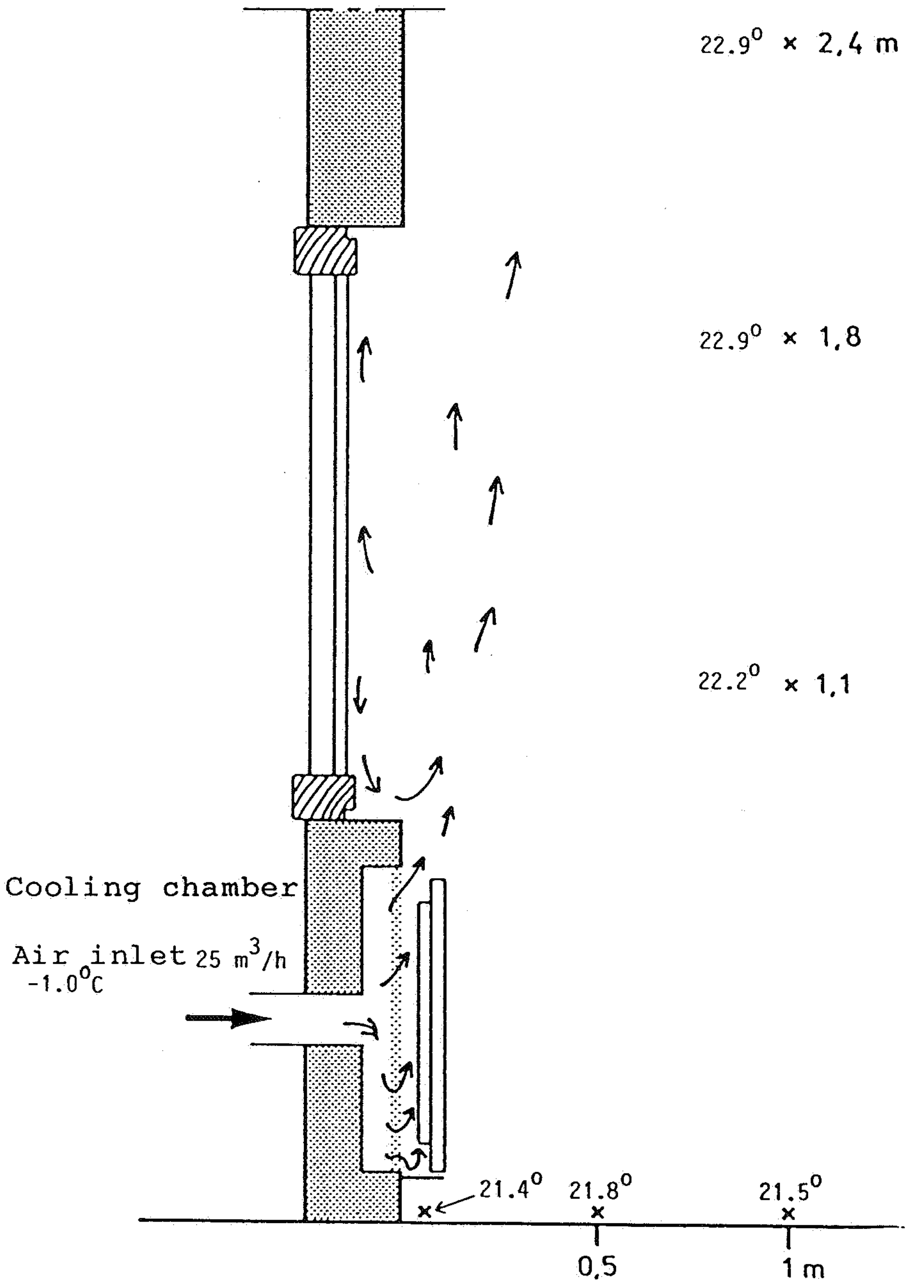

Figure 6 Test No. 4

The figure shows the air flow and the temperatures at outdoor temp. = -1.0°C and radiator heat load = 302 W.

It can be seen that there are no down draught problems or problems with cold air at floor level.

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"Air heating systems in airtight multifamily residential buildings"
Proceedings from 9th AIVC Conference, Gent 1988

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PROGRESS AND TRENDS IN AIR INFILTRATION
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Poster 9

ENERGY USE FOR TRANSPORT OF VENTILATION AIR

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1. SYNOPSIS

In the "Stockholm Project", different blocks of multi-family buildings have been extensively monitored for about three years. Temperatures, airflows and electricity use have been registrated each hour. As an additional base to this examination, ten fan units in the buildings have been intensively studied.

The results show that the specific use of power for transportation of ventilation air varies between approximately 1 and 4 kW per m^3 and second.

The results from the measurements indicate a notably low level of installation efficiency. The total efficiency of the ten units varies between 15 and 57 percent and has a mean value of 32 percent. The loads of the units have been comparatively low and consequently they have had an influence on the power factor. The operative conditions related to the loads also tends to have a disadvantageous influence on the efficiency.

The regained energy from heat recovery plants in two of the installations, compared to the total electric energy for operating the fans in the system, gives an approximate relation of 2 to 1. Regained energy compared to transformed energy specifically related to drops of pressure in the heat exchanger gives the relation 6 to 1.

Because of heat losses from ducts and inability to utilize the extracted energy from the exhaust air, the mean value for the useful recovered heat has been considerably below the efficiency of the plant.

The cause of the variaton in specific use of power for the examined units, mainly depends on the total pressure drops and the dimensioning of the fan motors. The power demand for the electric motors has in general been overestimated. 90 percent of the examined motors have an output power exceeding the requirement of power for the fan. The loads vary between 38 and 97 percent, with an average of about 64 percent.

One of the main reasons why the power of the motors exceeds the normal requirement, is the design of regulating extra needs of airflow on limited occasions, such as cooking hours. During normal operation the air flow is reduced by a damper. This gives an extra pressure drop for the main part of the running hours, which causes essential losses of energy.

The combination of high pressure drops and low efficiency in ventilation plants create unnecessary use of electric energy.

With a careful design of the system, well insulated ducts and an efficient regulation of air flow, the possibilities to improve the mode of application of the recovery system and decrease the use of electricity seem to be substantial.

2. BACKGROUND

In Sweden today, the uncertainty of the future situation of using and producing energy is noticeable. In this matter the questions of production and environmental effects have been dominating. The use of energy in buildings is considerable. Therefore, the potential of improving the efficiency of plants and building technology is of great interest.

The Swedish Council for Building Research encourages the development of new energy-effective techniques and systems for new and existing buildings. The "Stockholm Project" is the first larger joint experimental project for the evaluation of low energy consumption in multi-family buildings. The purpose of the project was to test new buildings that will lead to a lower requirement of purchased energy.

The technical evaluation of the project has been carried out by the Energy Conservation in Buildings Group - EHUB, a department of the Royal Institute of Technology - KTH, in Stockholm.

One of the results from the "Stockholm Project" is that advanced technique in multi family-buildings tends to increase the use of electricity, although the total level of purchased energy is comparatively low. The aspects of how different technical systems coordinate are of great interest and have been one of the main purposes of this evaluation.

In this paper the relation of use of heating and electricity for ventilation has been studied in six of the buildings of the project. Power to fans and state of pressure in ducts have been measured in ten different fan units. In two of the buildings the use of electric energy to the fans has been related to the energy balance of the ventilation system. The sacrifice of electric energy to regain heat from exhaust air has been calculated.

3. BRIEF DESCRIPTION OF BUILDINGS AND ENERGY SYSTEMS

Building A and B

The intention was to demonstrate the possibility of reducing the energy consumption by using conventional building methods. The buildings have been constructed with application of stringent air-tightness and insulation requirements.

Operating costs of the buildings in use have been tried to be kept low by concentration on quality during the building process. Improved construction documents, training of and information to the building workers and the provision of operating and maintenance instructions for users, have been important elements.

The heating system is a two-pipe, low-temperature system, operating at a maximum supply temperature of 55 °C and supplying not only radiators, but also the ventilation air heaters. Domestic hotwater, which is metered separately for each apartment, is supplied at a temeperature of 45 °C. The heated floor area of the two buildings together is 4282 m^2.

The buildings have a balanced ventilation system with mechanical supply and exhaust. The system also has heat exchanger of the air to air recuperative type.

Building C

Building C is a conventional Swedish apartment building. Each floor level contains two apartments consisting of five rooms plus kitchen. The total heated area is 1943 m^2. The structure is of cellular type with outer walls of 40 cm thick light-weight concrete blocks. The windows are triple-glazed.

The purpose was to reach a low requirement of purchased energy through utilization of solar energy gained by solar collectors integrated with the roof structure. The heating will be provided by means of a forced air heating system.

The building has a mechanically balanced ventilation system. Air is supplied to the building through an air/air heat exchanger in which the incoming air is preheated by the exhaust air. When the air through the solar collector reaches a sufficiently high temperature to be able to warm up the incoming air, the air is passed through the collector. Each apartment also has its own separate heating unit supplied by district heating.

Building D and E

Buildings D and E have a total heated floor area of 5336 m^2 and consist of 57 apartments. The buildings have been costructed of prefabricated sandwich type elements of concrete. The construction of balconies is carried out especially to reduce thermal bridges.

The concept with the energy system is to preheat the incoming air by passing the air through the external walls. The air inlets are designed as ducts in the concrete elements. During summertime the incomimg air passes through a separate outdoor air inlet. To reduce the peak power demand for heating, the structure is heavier than in conventional Swedich buildings.

The ventilation system uses mechanical exhaust air with the supply air system as described. The heating is provided by a heat pump and district heating. The heat pump recoveres energy from the exhaust air and supplies heat to the domestic hot water and the radiator systems.

Building F

Building F has a total heated floor area of 7265 m^2 and consists of 71 apartments. The object, which contains a glazed atrium, has been built with an in-situ-cast main concrete structure, with light- weight prefabricated external wall elements. Ducts for the heating and ventilating system have been built into the floor/ceiling slabs.

The building incorporates a forced air heating and ventilation system. Each apartment has an airheating unit in a wardrobe in the entrance hall. The heat is supplied by the domestic hotwater circuit, which therefore serves two purposes.

The ventilating is mechanically balanced with fans for exhaust and supply air. The incoming air is passed through two air-to-air heat exchangers, installed parallelly.

4. METHODS AND INSTRUMENTS OF MEASUREMENTS

The measurements have been carried out by the Energy Conservation in Buildings Group - EHUB, in cooperation with the consulting company AIB Anläggningsteknik AB, in Stockholm. The measurements include air flows, pressures in ducts and electric power to fan motors in the ten ventilation plants described in section three.

The air flows in the ducted systems have been determined by using a pitot static tube and a micro manometer. A traverse measuring has been carried out at a plane in a section of the duct. The traverse has been done in two diameters with nine points on each line. The method used is based on the recommendations of the NVG[5] (Nordiska Ventilations Gruppen).

The velocity profile has been plotted on the basis of the velocity pressure and the average velocity in the duct has been calculated. Corrections for errors in instruments and readings have been considered.

Total and static pressures have been measured by connecting the applicable tapping at the tail of the Pitot tube to the appropriate connection at the manometer. The connection used to measure the velocity pressure has then been left open to the atmosphere. Differences in static pressure across components in the system such as dampers, filters, heat exchangers and fans have been measured.

The manometer used is manufactured by FURNESS CONTROLS LTD.,s/n FM 2513 and was calibrated in connection with the measurements to an accuracy of 2.5 %.

A clip-on power instrument, EB 1286 MICROVIP MK1, has been used to measure power to the fan motors. The instrument indicates active power, current, voltage, frequency and power factor. The instrument was calibrated in connection with the measurements to an accuracy of 2 % measuring active power at the range 0 to 36 A.

The instrument can be used for measuring loads of one and three phase power. When measuring three phase power each phase has been measured separately to ensure that the load has been symmetric and if not, considered in the determinations.

Long term monitoring of the examined heat recovery plants in two of the buildings has been carried out by the Monitoring Center of Energy Research - MCE, a department of KTH.

The monitoring system is operated by a desk computer of the type Hewlett&Packard 85 or 86. The sensors in the system are connected to the computer through instruments of high standards of accuracy.

Signals from the sensors are registrated every 5:th or 12:th minute and stored as mean values or sums every hour. The accuracy of temperature meuserements is stated to be less than 0.1 Kelvin and measuring of energies in air to an accuracy of 10 %.

5. RESULTS

The results from the measurements of the ten fan - installations are illustrated in Table 5.1. All fan motors are of the type three phase non-synchronous. The fans are V-belt driven and of radial type with the blades bent forward.

Column one in the table shows the different buildings and the associated type of installations with a submerged index, the index $_i$ for units transporting in-coming air and $_e$ for exhaust air.

L indicates the load, defined as the relation between measured and rated power of the fan-motor:

$$L = \frac{P_m}{P_r}$$

where P_r stands for the rated power of the motors, and P_m stands for the measured power.

PF_m shows the measured relation between useful power and apparent power, the power factor. Q_m shows the measured airflow in m^3/s and p_t the total drop of pressure in Pascal (N/m^2).

The total efficiency of the installations = n_t, is defined as the relation between the measured air flow, Q_m (m^3/s) multiplied by the total pressure drop, p_t (N/m^2) in the installation divided by the measured power to the fan motor, P_m (Nm/s).

$$n_t = \frac{Q_m * p_t}{P_m}$$

Table 5.1. The results from the measurements of ten installations of fans.

Fan-Unit	P_m (W) Measur. Power	L (%) Load	PF_m Power factor	Q_m (m^3/s) Airflow	P (N/m^2) Pressure	n_t Efficien
A_i	1000	70	0.62	0.67	500	34
A_e	1670	57	0.54	0.65	640	29
B_i	1040	69	0.61	0.68	640	42
B_e	1980	66	0.62	0.86	590	26
C_i	700	64	0.57	0.68	330	32
C_e	1560	*104	0.58	0.73	578	27
D_e	2130	97	0.57	0.83	1450	57
E_e	1140	38	0.42	0.50	333	15
F_e	5100	46	0.51	1.98	540	21
F_i	7110	65	0.68	1.84	1268	33

* Variable speed of C_e is controlled by a frequency converter, why the load can exceed the rated load.

The mean value of the total efficiency of the installations of fans has been approximately 32 percent, and varies between 15 and 57 percent. The difference of total drops of pressure for different installations has been about 1100 Pascal.

The low levels of loads of the motors is noticeable. The mean value is 64 percent. The load has an influence on the power factor which decreases with a decreased load.

One way of comparing the efficiency of transporting air, is by studying the power to transport a specific volume of air as a function of the total drop of pressure in the system. The results show that the specific use of power varies between approximately 1 and 4 kW per m^3 and second.

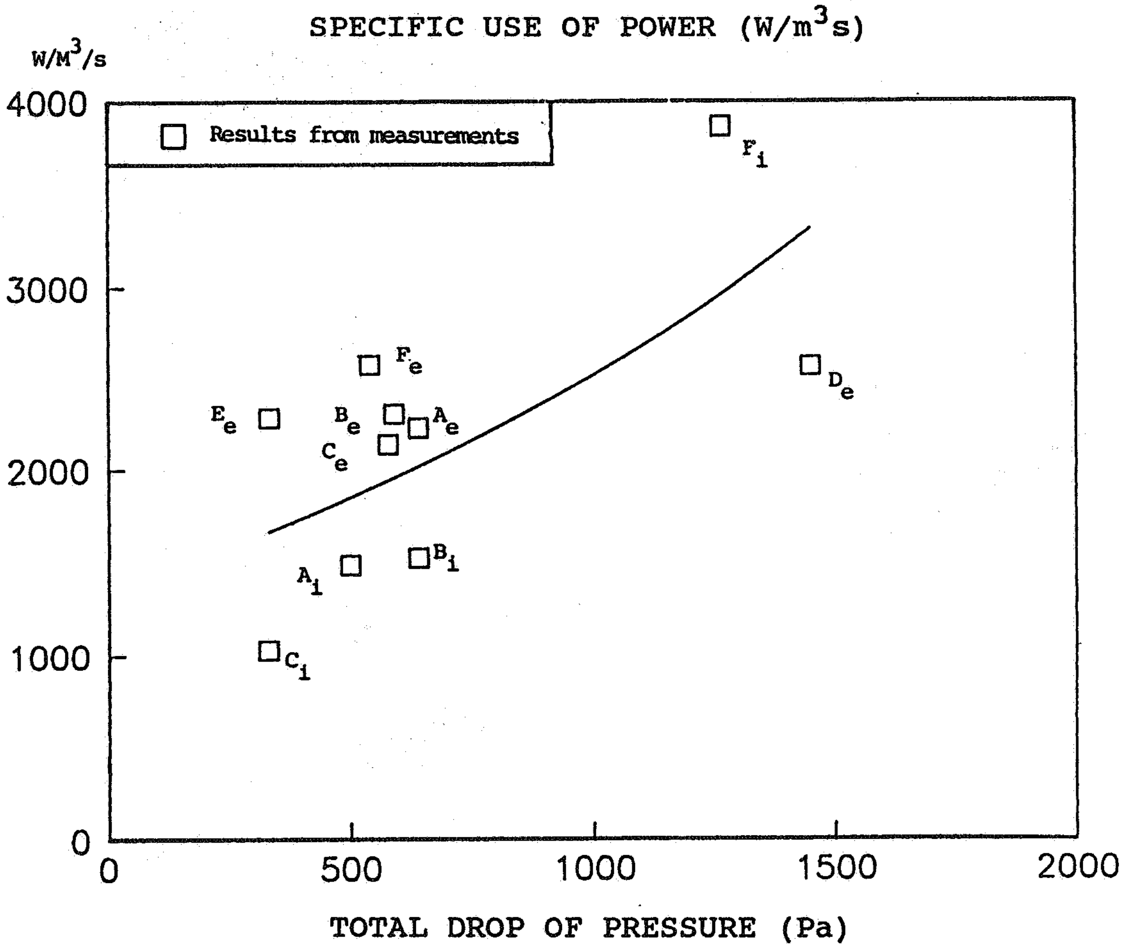

Figure 5.1. Specific power, defined as measured electric power to the fan motor at a certain airflow as a function of the total drop of pressure in the system.

It can be observed that the results from the calculations of specific power from the examined units are considerable scattered. Comparing the extremes the most significant digit is almost four times greater than the lowest.

The power demand to transport the same amount of air varies because of differences in drops of pressure and designs of systems. The main reasons for these variations are owing to the influence of adjustability and dimensioning of the electric motors.

The total electric energy to the fans over a year has been calculated by extrapolating test results from instantaneous measurements. The electric energy extensively related to the regaining system = P_{re}, has been calculated from drops of pressure across both sides of the exchanger including the filter for the exhaust air.

The values of pressure drops (N/m^2) have been multiplied with values of the air flows (m^3/s). The results have then been divided with the total efficiency of the fan unit, as:

$$P_{re} = \frac{Q_{mi} * P_{exi}}{n_{ti}} + \frac{Q_{me} * P_{exe}}{n_{te}}$$

Where Q_{mi} = Incoming airflow through the exchanger

P_{exi} = Drop of pressure across the exterior side of the exchanger

Q_{me} = Exhaust airflow through the exchanger

P_{exe} = Drop of pressure across the exhaust side of the exchanger including the filter.

n_{tx} = Total efficiency of the unit

Some of the energy that is transformed to heat by the drop of pressure can be used for heating. The transformed energy in the exhaust system has normally to be considered as losses.

The heat recovery units (consisting of air-to- air recuperative heat exchangers) and the fans in building A and B are placed in a plant room situated on the top floor. The ducts between the apartments and the units lead through unheated spaces. The ducts are insulated according to the Swedish Building Code.

The heat losses from the ducts depend on dimensions, insulation, length of ducts, difference in temperature outside and inside the ducts and the velocity of the transported air. The energy losses are proportional to drops of temperature.

In building A and B, mean values of the losses have been calculated to approximately five percent of the transported energy. These buildings were built under certain supervision to obtain greatest quality, why similar or worse conditions can be expected in the oridinary production of the same category.

DIFFERENCES IN AIR TEMPERATURES BETWEEN INDOORS AND THE RECOVERY PLANT °C.

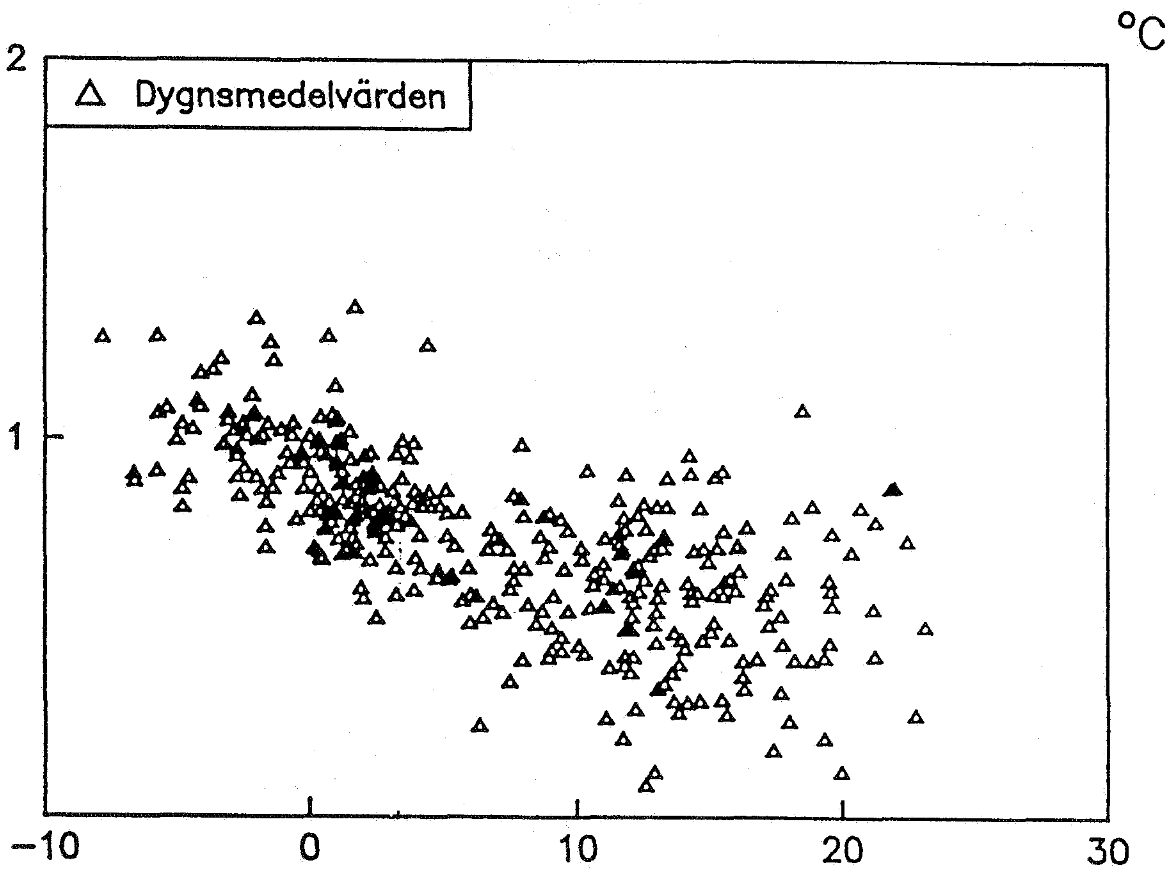

Figure 5.2. Differences between temperature in apartments and exhaust air passing the heat exchanger in building A, as a function of the outdoor temperature. The temperature difference is shown by daily averages for a year.

Some of the electric energy to the fans and heat losses from the ducts can be considered as sacrificed energy to provide the recovered energy. To study this relation, the influence of the internal heat-load on the possibilty to use the regained energy has to be calculated. This has been done by subtracting total regained energy over a year with the regained energy during the season without need of external heating, defined as the period between the 15:th of May and the 15:th of September.

Table 5.2. Illustrates the relation between regained energy and electric energy to fans in MWh during one year .

	Building A	Building B
Energy in exhaust air	100	152
Regained in exchanger	58	79
Losses from ducts related to regain	-3	-4
Regained during summer	-10	-14
Utilized	**45**	**61**
Total electric energy to fans	23	26
Transport through exchanger, P_{re}	8	9

The degree of efficiency for two of the heat exchangers has been measured to 58 and 52 percent respectively. Owing to heat losses from ducts and inability to utilize the extracted energy from the exhaust air during the summerperiod, the mean value for a year has been calculated to 45 and 40 percent respectively.

The regained heat, compared to total electric energy for running the fans in the ventilation system, gives an approximate relation of 2 to 1. Regained heat compared to transformed energy related to drops of pressure in the heat exchanger is about 6 to 1.

6. DISCUSSION

The results show that the coordination of different technical systems must be observed. The efficiency of an energy recovery plant cannot be considered as a single factor connected to a specific unit, without looking at the entire system. This approach is necessary, especially as the complexibility of new techniques tends to increase.

The studies of the relation between regained heat from exhaust air and the electric energy to transport ventilation air through different parts of the system, show that the degree of efficiency can vary significantly. The way of designing the system is of vital importance for the efficiency.

Owing to the strong influence of the pressure drops on the power demand to transport ventilation air, the pressure drops should be minimized. This can for instance be achieved by choosing a larger dimension of recovery plant. The exceeding investments for a larger recovery unit must be related to the reduced costs of power and energy.

The design of systems where air flows are increased during short periods, must be considered. A common design is a damper that can be regulated. The drop of pressure across the damper during normal operation can be great and the losses of electric energy noticeable. These losses cannot normally contribute to the heating.

The only examined fan unit equipped with a frequency converter to control variable speed, has a noticeably low degree of efficiency. The operation of this motor has been disturbed, probably because of overload. The frequency converter makes it possible to exceed the rated load.

Beside the drops of pressure in the ventilation system the dimensioning of the fan motors has an essential effect on the use of energy for the transport of air.

The efficiency of the motors varies according to the load. Due to the size and type, the motors in the examination have an indicated efficiency of approximately 70 to 80 percent at rated power. The efficiency is almost stable at the upper range of the load. When the load decreases below 50 to 60 percent of the rated power, a drastic drop of efficiency is indicated.

The load of the ten fan motors in the test varies between 38 and 97 percent and the results of the examination indicate a strong relation between load and efficiency of the fan unit. The total efficiency of the ten units has been plotted as a function of load, and has a linear correlation of 0.91 (Figure 6.1).

To create the moving force in a non-synchronous electric motor a rotating electro-magnetic force has to be generated. This creates inductance with an angle of phase-difference between voltage and current. Inductance and current create reactive and apparent power. The Power Factor = PF, is defined as the quota of active and apparent power.

Electromagnetic force, bearings and cooling of the motor create almost non-variable losses, which in relation to the total losses increase at reduced load.

Beside the low efficiency of the motor caused by operating on low load, the reactive power creates losses from the eletricity supply network. Furthermore, the network cannot be used at its optimum because it has to be dimensioned for the apparent power.

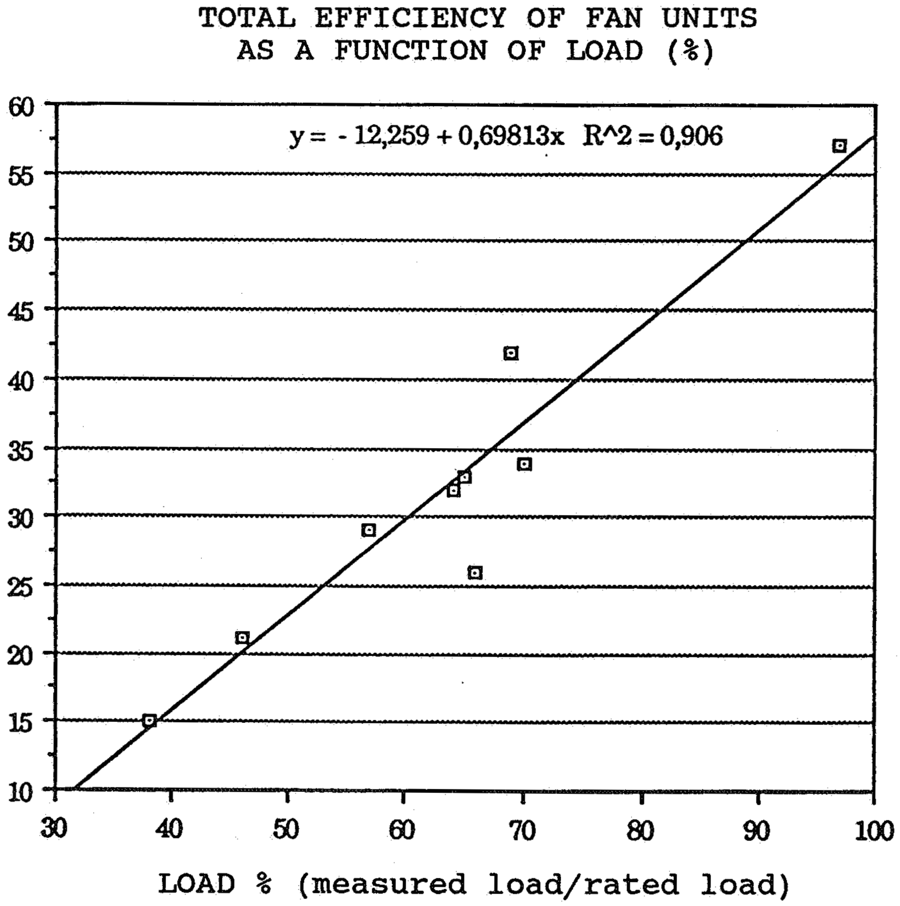

Figure 6.1. The figure illustrates the relation between the total efficiency of the fan units and the load. The influence of the load on the degree of efficiency indicates to be strong. The relation has a linear correlation of 0.91.

It is therefore of great importance that the motor is operating at the upper range of the load to achieve its rated efficiency. This demands accuracy in designing and operating the system.

The fan and the electric motor must be adjusted to the actual state of pressure in the system. To achieve an accurate adjustment the plant probably has to be in operation before final adjustment can be carried out.

The adjustment and the design of the fan units consequently have a great effect on the possibility to recover energy in an efficient way. The efficiency of the recovery plant must be compared to the energy related to its operation.

Besides electric energy for transporting air, the heat losses from the ducts transporting preheated air from the heat exchanger have to be calculated. The demands on insulation of ducts in the Swedish Building Code are not as strong as on the building envelope.

The total need of heating, influenced by climate, building technology and internal heat load, which strongly effect the possibility of utilizing the recovery plant on a yearly basis, must also be considered.

Owing to heat losses from ducts and inability to utilize the extracted energy from the exhaust air, the mean value for used energy from the heat exchanger has been considerably below the efficiency of the recovery unit. The values for two of the plants have been calculated to 45 and 40 percent respectively for a year compared to the efficiency of the heat exchanger unit which has been 58 and 52 percent respectively for the same period.

7. CONCLUSIONS

Adjustability and needs of increased air flows should not be designed with dampers. The losses of energy due to drops of pressure across the dampers during the main part of operating hours can be essential. If the dampers are installed in ducts for exhaust air, transformed energy from the drop of pressure cannot be used for heating.

Reducing the air flow with dampers during short periods of times creates operating conditions which are negative for the type of electric motors normally used in plants for ventilation. If the rated power of the electric motor exceeds the need for driving the fan, the influence on the use of energy for transport of air can be considerable.

Heat losses from ducts and inability to utilize the extracted energy from the exhaust air considerably decrease the efficiency of the plants. Therefore, the total need for heating effected by climate and internal heat in the building has to be considered when designing a recovery plant and making profitability analysis.

The examined systems were designed as recently as 1980, why similar or worse conditions can be expected in a large number of plants in operation.

With awareness of the importance of accuracy in designing systems and the co-ordination of techniques, there seems to be a considerable potential to improve the energy efficiency of plants for ventilation and heat recovery.

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PROGRESS AND TRENDS IN AIR INFILTRATION
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Poster 10

COMPARISON OF AIR INFILTRATION RATE AND AIR LEAKAGE TESTS UNDER REDUCTIVE SEALING FOR AN INDUSTRIAL BUILDING

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1. SYNOPSIS

The paper compares air infiltration rate measurements with air leakage measurements in a modern industrial building. In each case the tests have been performed firstly with the building 'as-built', and then with the major leakage components sealed.

The building investigated was of a cladding wall construction with U-values of 0.6 $W.m^{-2}.K^{-1}$ for both the walls and roof. It had a floor area of 466 m^2. The volume was 3050 m^3.

Tracer decay tests and constant concentration methods (both using N_2O) were performed in the building to establish the air infiltration rates. The air leakage of the building was determined by the fan pressurisation method.

The paper presents the results of the measurements and the discussion focuses on the variations of the air infiltration rate due to changes in internal and external conditions. The results from the three different techniques used are compared.

The results show that there was good agreement between the tracer decay and constant concentration methods when determining the air infiltration rate. There was also good agreement under reductive sealing between the reductions in measured air infiltration rate and measured air leakages.

The paper is a result from research work funded by the Building Research Establishment to investigate air infiltration rates and air leakage rates in Industrial Buildings.

2. INTRODUCTION

This paper describes the air infiltration and air leakage tests carried out in a modern lightweight industrial unit constructed in approximately 1986.

The main aims of the investigation were:

(i) To carry out tracer decay and constant concentration experiments to determine the air infiltration rate and its variation with wind velocity and internal-external air temperature difference.

(ii) To determine the air leakage rate at an induced internal-external static pressure difference up to 50 Pa.

(iii) To repeat (i) and (ii) above having carried out sealing measures to the loading door and roof ventilators.

The factory had a production space floor area of 466 m^2 and a production space volume of 3050 m^3. The factory walls had an inner leaf of masonry with a cladding external leaf. The roof was of a metal cladding construction with an inner leaf of fibreboard. Both constructions incorporated glass fibre quilt insulation to give a U-value of 0.6 $W.m^{-2}.K^{-1}$. The factory was built on a concrete slab base. There was an office space which was partitioned off from the production area.

There was 7.5%, by area, of glazing in the walls (all double glazed and tinted) and approximately 10% skylights in the roof. There were 4 louvered ventilators fitted in the roof, each of area 1.56 m^2. These were electrically operated by means of a single switch which controlled all four vents simultaneously.

The factory was fitted with a single standard roller shutter loading door of approximate area 16 m^2. The loading bay door faced north, the factory being orientated on an east-west axis.

Figure 1 contains a floor plan and section of the factory.

For factories of this size and construction, the recommended[1] design air infiltration rate for the purpose of design heat loss calculations is quoted at 1 $ac.h^{-1}$.

3. METHOD

3.1 Tracer Gas Test Equipment

Tracer gas tests were performed using the Autovent System developed and marketed by British Gas plc.

The system comprised of a gas analyser, 2 injection units each having 7 injection channels (6 for the primary tracer and one for the secondary tracer) and a 12 channel sample unit. The equipment was controlled by a personal computer via an interface. In addition there was the facility to monitor 12 internal air temperatures, the external air temperature, the wind speed and wind direction.

The gas analyser used was a Binos infra-red twin channel analyser that measured sulphur hexafluoride (SF_6) and nitrous oxide (N_2O) on dedicated channels and in the range 0 - 200 ppm. The analyser was fitted with an optical filter which prevented the N_2O measurement being affected by cross-sensitivity to water vapour. In addition to the two display meters (one each to display the measurements for SF_6 and N_2O), there was also an output signal 0-1 Volt dc that corresponded to the measurement range 0 - 200 ppm.

A Rexagan interface was used as the interface between the controlling computer and the sample/injection units. An IBM-XT personal computer was used as the controlling computer with an Epson FX-100 printer which permitted summary information in the form of tables to be printed out during the course of a run after a user-specified time interval.

The Autovent logging program permitted a constant concentration experiment to be run using the primary tracer gas (ie injecting on channels 1 to 12). In addition, 10 pairs of injection and release times could be entered for each of the two remaining channels (Channels 13 and 14). This permitted the running of a simultaneous decay experiment using a second tracer gas.

The tracer decay tests described in this paper were performed by preparing a program disc beforehand that set up the channels as sample only (ie no injection required). The factory was then brought up to target concentration in the usual way. Once a uniform concentration was achieved, the constant concentration run was aborted and the subsequent decay recorded using the sample only program disc.

Both the constant concentration tests and the tracer decay tests were performed using pure nitrous oxide (N_2O) as the tracer gas. An initial target concentration of 75 ppm N_2O was used for the tracer decay experiments. A target concentration of 50 ppm was used for the constant concentration experiments. The calibration of the gas analyser performed at the start of each test was achieved using a cylinder of 50 ppm N_2O in N_2.

Tracer decay tests were performed using a 120 second cycle time (ie the time taken to sample channels 1 to 12 inclusive). Subsequent data analysis was based on half hourly average readings.

Constant concentration tests CC01 – CC03 were performed using a 120 second cycle time. The remaining constant concentration tests (CC04 – CC08) were performed using a 360 second cycle time. Again, subsequent data analysis was based on a half hourly time base.

3.2 Air Leakage Test Equipment

The pressurisation unit employed for the tests at this factory was a modular system based on two fans of nominal diameter 24", namely a Woods 24 JL and a Eurofoil CA635. The unit was arranged such that the Eurofoil unit was mounted on top of the Woods unit and the whole arrangement fitted into a standard sized fire exit. The door exit was sealed by means of a plywood blanking plate fitted with bell mouthed inlets on the external face and flanges on the internal face. The fan system was offered up to the fire exit and linked to the flanges by means of flexible connectors.

The two fan ducts were each of 3m length. The volume flow rate through the ducts was measured using Wilson Flow Grids. Flow straighteners were fitted to eliminate swirl within the region of the Flow Grids.

Volume flow rate control was by means of speed controllers fitted to each fan. The Woods unit had a two speed controller whereas the Eurofoil fan unit had a five speed control unit.

The fans on maximum speed delivered 4.1 $m^3.s^{-1}$ and 4.3 $m^3.s^{-1}$ for the Woods and Eurofoil units respectively.

A schematic drawing of the fan pressurisation system is illustrated in Figure 2.

3.3 Work Programme

Air infiltration (tracer decay and constant concentration) and air leakage (fan pressurisation) experiments were carried out during July/August 1988.

The work programme for this study is shown in Table 3.1 below.

Table 3.1: Work Programme

Test	Configuration
Tracer Decay	
D01 and D02	As built
D03	As built. Vents open
Constant Concentration	
CC01 - CC06	As built
CC07	Loading door sealed
CC08	Loading door and roof vents sealed
Air Leakage	
P01 (a)	Loading door sealed
P01 (b)	Loading door sealed. Vents open
P02	Loading door and vents sealed
P03	As built

4. RESULTS

The results for the tracer decay, constant concentration and air leakage tests are summarised below.

The wind speed during the experimental period, averaged using a 30 minute time base, was recorded in the range from 0 to 7 $m.s^{-1}$. The wind direction was predominantly south and south west. External temperatures were recorded in the range 11 to 21 °C.

4.1 Tracer Decay

The tracer decay curves for the three experiments, namely Tests D01, D02 and D03, are shown in Figure 3. The corresponding log-linear transformations are shown in Figure 4. A half hourly averaging time base has been used in all cases.

The results of each test are summarised in Table 4.1 below.

Table 4.1 : Summary of Tracer Decay Tests

Test	Ventilation Rate $ac.h^{-1}$	$m^3.s^{-1}$	Wind Speed $m.s^{-1}$	Wind Direction	Stack $(°C)^{1/2}$	Configuration
D01	0.30	0.25	Low*	--	1.60	As-built
D02	0.39	0.33	4.1	SW	1.83	As-Built
D03	2.12	1.80	3.7	SW	1.89	As-built; Roof vents open.

* Although the windspeed and wind direction monitoring equipment was not operating for the duration of this test, on site observations noted that the windspeed decreased from moderate to very light during the course of the test. Examination of Figure 4, ie the log-linear plot of the concentration decays, suggests that after an elapsed time of four hours the infiltration rate changes. The tracer decay curve appears to consist of two distinct parts. The mean air infiltration rates were found to be 0.53 $ac.h^{-1}$ (0.45 $m^3.s^{-1}$) for the first part of the run (moderate windspeed) and 0.19 $ac.h^{-1}$ (0.16 $m^3.s^{-1}$) for the latter part of the run (light windspeed).

4.2 Constant Concentration

Constant concentration experiments were carried out for three factory configurations, namely as-built, loading door only sealed and loading door plus roof vents sealed.

The average values for each test are presented in Table 4.2 below.

Table 4.2 : Summary of Constant Concentration Tests

Test	Ventilation Rate $ac.h^{-1}$	$m^3.s^{-1}$	Wind Speed $m.s^{-1}$	Wind Direction	Stack $(^\circ C)^{1/2}$	Configuration
CC01	0.27	0.23	2.4	S,SW	1.9	As built
CC02	0.32	0.27	2.5	S	1.7	As built
CC03	0.42	0.36	3.0	S	2.9	As built and heating on
CC04	0.50	0.42	4.6	S	2.2	As built
CC05	0.48	0.41	3.7	S,SW	2.6	As built
	2.44	2.07	3.7		1.9	Roof vents open
CC06	0.33	0.28	2.4	S,SW	2.6	As built
CC07	0.27	0.23	0.9	S,SW	4.2	Door sealed and heating on
CC08	0.31	0.26	3.2	S,SW	2.9	Door and ventilators sealed

The data from Tests CC01-CC06 inclusive was amalgamated to form a single data set comprising of factory as-built data. The relationship of air infiltration rate with windspeed was examined for the factory as-built and factory with loading door plus roof vents sealed configurations. The windspeed range for the factory with door only sealed was too small to warrant any investigation of the relationship of air infiltration rate with windspeed. The analysis firstly considered all recorded wind directions together and then considered subsets of each data file by separating out the data into separate wind direction components, namely south and southwest directions for both factory configurations, there being little data recorded for any other directions.

Figures 5 to 7 inclusive show the air infiltration rate versus windspeed for both factory configurations for all wind directions, south winds and southwest winds respectively.

As the air infiltration rate appeared to be highly correlated with windspeed, a linear regression was performed on each set of data. A relationship of the following form was assumed to exist.

$$I = C_1 \cdot (WS) + C_2$$

Where

I = air infiltration rate (ac.h^{-1})
WS = windspeed (m.s^{-1})
C_1, C_2 = coefficients to be found from the regression analysis

The regression coefficients obtained are presented in Table 4.3 below together with the standard error of estimating the air infiltration rate, the correlation coefficient of the regression, (R^2) and the number of observations (N_{obs}). These regression are shown in Figures 8 to 10 inclusive.

Table 4.3: Regression Data For Air Infiltration Rate As A Function Of Windspeed Only.

Wind Direction	C_1	C_2	Standard Error	R^2	N_{obs}
As-Built					
All Directions	0.096	0.100	0.069	80.5%	375
South	0.076	0.145	0.063	65.7%	211
Southwest	0.110	0.056	0.063	88.4%	144
Loading Door Only Sealed					
All Directions	-0.003	0.274	0.039	0.3%	29
South	-0.057	0.308	0.041	17.5%	15
Southwest	-0.003	0.281	0.037	0.6%	10
Loading Door And Vents Sealed					
All Directions	0.062	0.110	0.053	67.8%	138
South	0.046	0.142	0.042	57.0%	91
Southwest	0.061	0.144	0.055	62.1%	47

To investigate the relationship of air infiltration rate with climate further, the stack effect was considered.

The measured air infiltration rates were normalised with respect to the stack and plotted against the windspeed normalised with respect to stack. Figures 11 to 13 inclusive show plots of (air infiltration rate ÷ stack) versus (windspeed ÷ stack) for the two factory configurations under study for all wind directions, south wind and south west wind components respectively.

A regression analysis was performed that took into account the stack effect. A relationship of the form;

$$I = C_3 \cdot WS + C_4 \cdot S \quad (\text{ac.h}^{-1})$$

was assumed where

I = air infiltration rate (ac.h^{-1})
WS = Windspeed (m.s^{-1})
S = Stack ($^{\circ}C^{½}$)
C_3, C_4 = Coefficients to be found from the regression analysis.

The data obtained from the linear regressions performed are presented below in Table 4.4.

Table 4.4: Air Infiltration Rate As A Function Of Windspeed And Stack.

Wind Direction	C_3	C_4	Standard Error	R^2	N_{obs}
As-Built					
All Directions	0.093	0.046	0.061	84.4%	375
South	0.081	0.055	0.056	72.8%	211
Southwest	0.104	0.032	0.059	90.0%	144
Loading Door Sealed					
All	0.018	0.061	0.036	13.4%	29
South	−0.028	0.069	0.037	34.8%	15
Southwest	0.022	0.060	0.036	75.2%	10
Loading Door And Vents Sealed					
All	0.066	0.033	0.054	66.8%	138
South	0.059	0.036	0.047	45.6%	91
Southwest	0.069	0.035	0.060	54.8%	47

The above regression fits were applied to the factory as-built and factory with door plus vents sealed configurations to determine the reductions in air infiltration rate effected by sealing. The air infiltration rate has been shown versus windspeed (using the regression equations obtained for all wind directions) and for two values of stack, namely 1 and 3 $^{\circ}C^{1/2}$ in Figures 14 and 15 respectively.

The potential reductions at these stack values are summarised in Tables 4.5 and 4.6 for two values of windspeed, namely 2 $m.s^{-1}$ and 5 $m.s^{-1}$.

Table 4.5: Predicted Air Infiltration Rates At Stack = 1 $^{\circ}C^{1/2}$.

Configuration	Windspeed 2 $m.s^{-1}$	5 $m.s^{-1}$
As-built	0.232	0.511
Door And Vents Sealed	0.165	0.363
Reduction	28.9%	29.0%

Table 4.6: Predicted Air Infiltration Rates At Stack = 3 $^{\circ}C^{1/2}$.

Configuration	Windspeed 2 $m.s^{-1}$	5 $m.s^{-1}$
As-built	0.324	0.603
Door And Vents Sealed	0.231	0.429
Reduction	28.7%	28.9%

4.3 Air Leakage

Three air leakage experiments, namely Tests P01, P02 and P03, were performed as detailed in the work programme, Table 3.1 for the loading door sealed, loading door plus roof vents sealed and factory as-built configurations respectively. In addition, for the loading bay door sealed configuration (Test P01), the roof vents were opened but a maximum internal-external static pressure difference of only 4 Pa was achieved (fan flow rate = 8.3 $m^3.s^{-1}$).

The air leakage plots for the three factory configurations are shown in Figure 16 together with an experimental line fit to the data in each case.

The curve fits to the experimental data shown were found by assuming a relationship of the form:

$$Q = C \cdot (\delta P)^n \quad (m^3.s^{-1})$$

Logarithmic transformation of the variables was then performed to find the flow coefficient, C, and the exponent, n.

The resulting equations for each factory configuration are given below.

<u>Test P01 (Door Sealed)</u>

$$Q = 0.446 \cdot (\delta P)^{0.667}$$

R-sq = 99.6% (7 data points)

<u>Test P02 (Door and Vents sealed)</u>

$$Q = 0.539 \cdot (\delta P)^{0.608}$$

R-sq = 99.4% (6 data points)

<u>Test P03 (Factory as-built)</u>

$$Q = 1.052 \ (\delta P)^{0.540}$$

R-sq = 99.1% (7 data points)

A summary of the air leakage rates at 50 Pa internal-external static pressure difference (ie Q_{50}) is presented in Table 4.7 below together with the Q_{50} air leakage normalised with respect to the envelope area of the factory.

The air leakage at 50 Pa quoted for the factory as-built case was an extrapolated value obtained from regression analysis performed on the logarithmically transformed variables. The extrapolation is considered valid[2] as the regression coefficient is greater than 99.0% and the flow exponent, n, has a value between 0.5 and 0.7.

The leakages at 50 Pa for the other two cases were also calculated from the regression fits, and can be checked graphically from the experimental data since pressure differences of up to 75 Pa were achieved.

Table 4.7: Summary of Air Leakage Tests

Test	Q_{50} ($m^3.s^{-1}$)	($m^3.s^{-1}.m^{-2}$)*	Configuration
P01	6.06	0.056	door sealed
P02	5.83	0.054	door and vents sealed
P03	8.70	0.079	'as built'

* The envelope area of the factory was estimated to be 1100 m^2. The area of the loading door was taken to be 16 m^2. There were 4 ventilators, each of area 1.56 m^2.

5. DISCUSSION

The tracer decay results show an increase in infiltration rate from 0.30 $ac.h^{-1}$ (0.25 $m^3.s^{-1}$) to 0.39 $ac.h^{-1}$ (0.33 $m^3.s^{-1}$) with increasing wind speed, from low to moderate. Opening the roof vents increased the ventilation rate to 2.12 $ac.h^{-1}$ (1.80 $m^3.s^{-1}$).

The constant concentration results show that for all wind speeds there was a good correlation of infiltration rate with wind speed (wind direction was almost without exception from the south/south west). This data is presented in Figures 5, 6 and 7. At low wind speeds (less than 1 $m.s^{-1}$) the infiltration rate was approximately 0.2 $ac.h^{-1}$ (0.17 $m^3.s^{-1}$). The correlation with stack was second order, as demonstrated by the strong correlation of air infiltration rate with windspeed (Figures 5, 6 and 7) and the relative magnitude of the windspeed and stack coefficients in the regression equations (Table 4.4). For the 'as built' configuration, including the stack in the regression improved the goodness of fit (ie. R-squared). However, for the sealed configuration there is a slight decrease in the goodness of fit, from 67.8% to 66.8%, when stack is included. Figures 11, 12 and 13 show air infiltration rate normalised with respect to stack versus windspeed normalised with respect to stack.

From the regression equations relating air infiltration rate to windspeed (all directions) and stack for the factory as-built and factory with door plus roof vents sealed, sealing the loading door and roof ventilators reduced the infiltration rate by approximately 28% (refer to Figures 14 and 15, Tables 4.5 and 4.6).

Comparing constant concentration runs with tracer decay for low and moderate wind speeds gave good agreement as shown in Table 5.1 below.

Opening the roof vents increased the infiltration rate to 2.44 $ac.h^{-1}$ (2.07 $m^3.s^{-1}$) as measured by the constant concentration method. This again agrees well with tracer decay results for roof vents open, (air infiltration rate 2.12 $ac.h^{-1}$), as summarised in Table 5.1 below.

Table 5.1 : Comparison of Tracer Decay with Constant Concentration Tests

	Tracer Decay			Constant Concentration		
	($ac.h^{-1}$)	($m^3.s^{-1}$)		($ac.h^{-1}$)	($m^3.s^{-1}$)	
Low Wind	0.30	0.25	(D01)	0.27	0.23	(CC01)
Moderate Wind	0.39	0.33	(D02)	0.48	0.41	(CC05)
Roof Vents open	2.12	1.78	(D03)	2.44	2.07	(CC05)

At moderate windspeeds, the mean air infiltration rate as determined by the constant concentration experiment (0.48 $ac.h^{-1}$) was higher than that determined by the tracer decay experiment (0.39 $ac.h^{-1}$). This was due to the mean value of the stack being higher for the constant concentration test (2.6 $^\circ C^{1/2}$) than for the tracer decay test (1.8 $^\circ C^{1/2}$) (refer to the Appendix, Tables A2 and A8).

The air leakage results show a decrease in air leakage rate with increasing sealing, from 8.70 $m^3.s^{-1}$ (0.079 $m^3.s^{-1}.m^{-2}$) (extrapolated value) to 6.06 $m^3.s^{-1}$ (0.056 $m^3.s^{-1}.m^{-2}$) to 5.83 $m^3.s^{-1}$ (0.054 $m^3.s^{-1}.m^{-2}$), for the 'as built', door sealed and door and vents sealed cases respectively. This represents a reduction in air leakage of 23% achieved by sealing the loading door, and a total reduction in air leakage of 33% achieved by sealing both the loading door and the roof vents.

The percentage reduction (33%) in air leakage effected by sealing the loading door and roof vents agrees reasonably well with the reduction in air infiltration rate (28%) as measured by the constant concentration experiments.

6. CONCLUSIONS

Infiltration rates were well correlated with windspeed for the wind directions (south and south west) under which measurements were taken. There was good agreement between the tracer decay and constant concentration tests results. This would imply that both methods are suitable for air infiltration measurements in this size of building.

On average, the air infiltration rate was measured to be 0.4 $ac.h^{-1}$ (0.34 $m^3.s^{-1}$). This is well below the CIBSE[1] design value of 1 $ac.h^{-1}$ for this size and construction factory.

Averaging the results obtained from tracer decay and constant concentration experiments showed that opening the roof vents provided a ventilation rate of 2.3 $ac.h^{-1}$ (1.95 $m^3.s^{-1}$).

The air leakage rate was 8.7 $m^3.s^{-1}$ (0.079 $m^3.s^{-1}.m^{-2}$) at 50 Pa static pressure difference (extrapolated value). On opening the roof vents with the loading door sealed, a maximum flow rate of 8.3 $m^3.s^{-1}$ produced an internal – external static pressure difference of 4 Pa.

Sealing the roof ventilators and loading door reduced the air infiltration rate by 28% as determined by the constant concentration method. This compares well with a reduction in air leakage of 33% at 50 Pa static pressure difference as measured by the fan pressurisation method.

The major component for air leakage as identified by the air leakage tests was the loading door. Sealing the loading door reduced the air leakage at 50 Pa by 23%.

7. ACKNOWLEDGEMENTS

The work described here formed part of a program of work funded by the Building Research Establishment (BRE) that is still in progress. The paper describes interim results and does not necessarily represent the views of the BRE.

We would like to acknowledge the help of the Welsh Development Agency in making the factory available for our tests.

8. REFERENCES

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   The Chartered Institute Of Building Services Engineers, London.

2 Determining the airtightness of buildings by the fan pressurisation method: BRE recommended procedure. p8.
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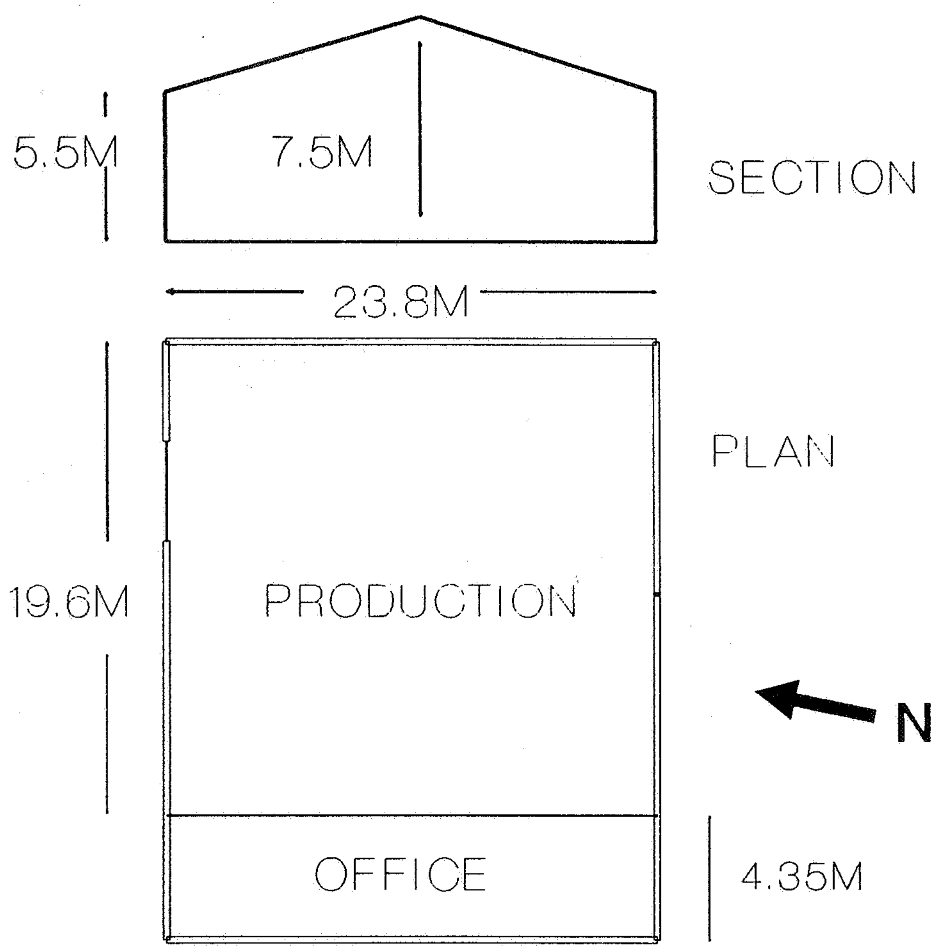


Figure 1: Section And Plan Of Factory

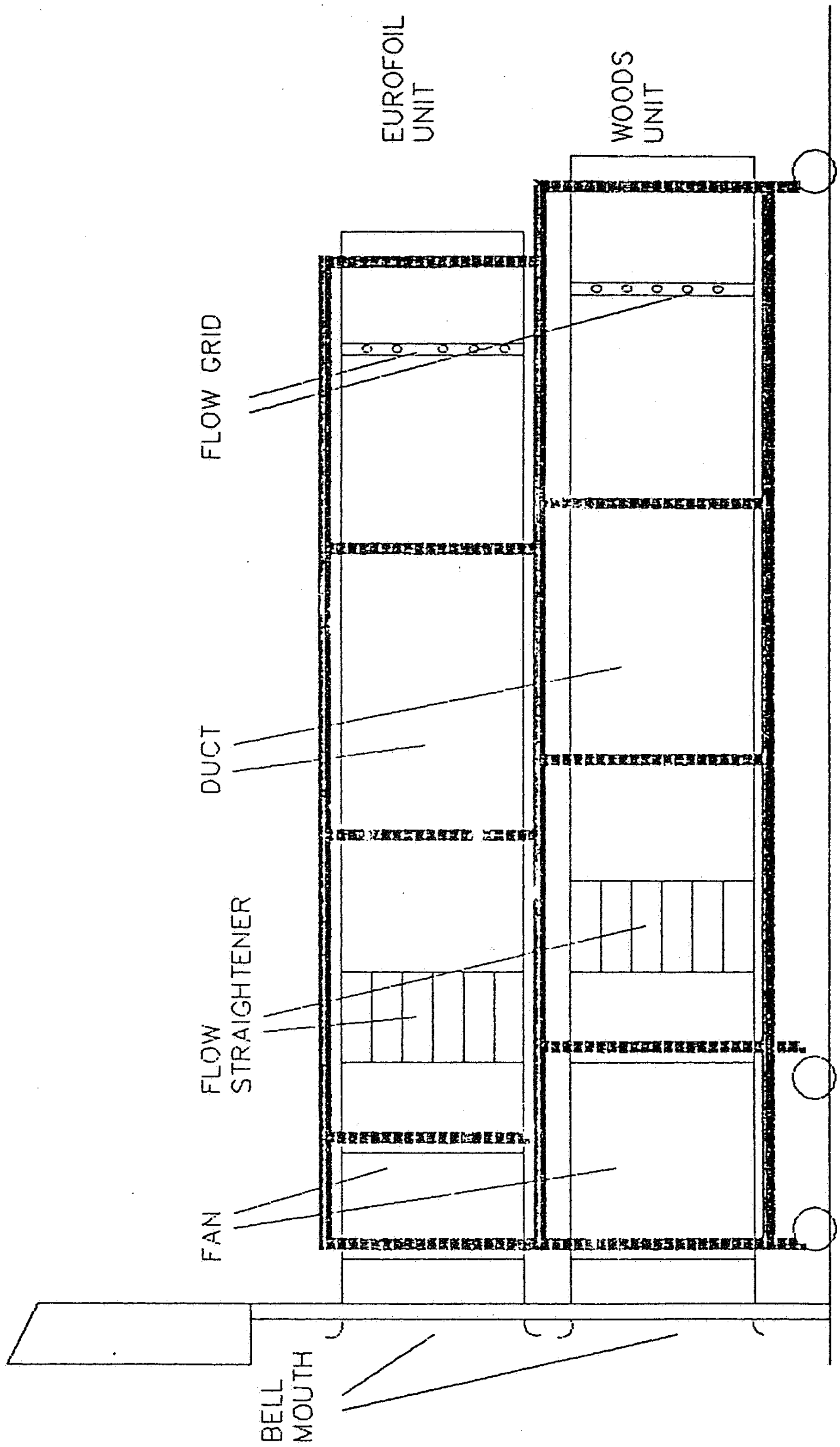


Figure 2: Fan Pressurisation Rig

Figure 3: Tracer Decays
Tests D01, D02 and D03

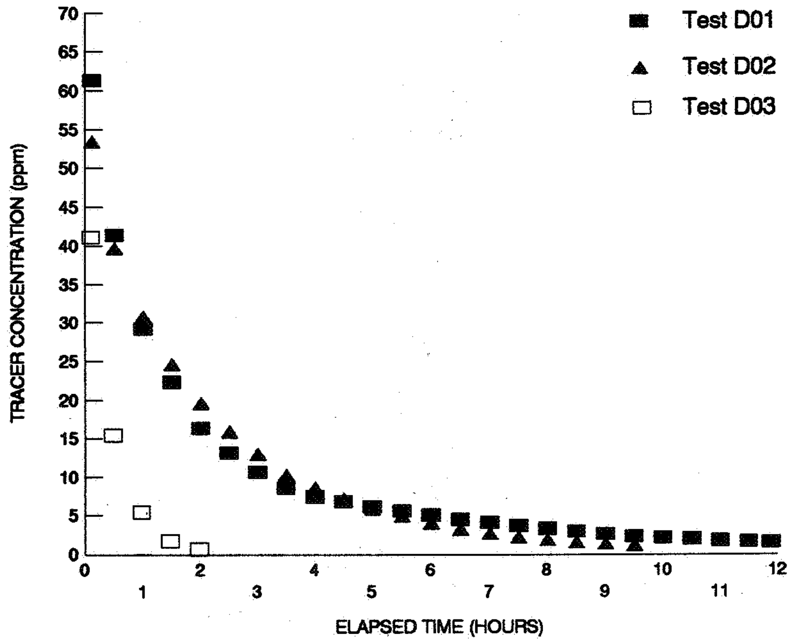


Figure 4: Logarithmic Tracer Concentrations
Tests D01, D02 and D03

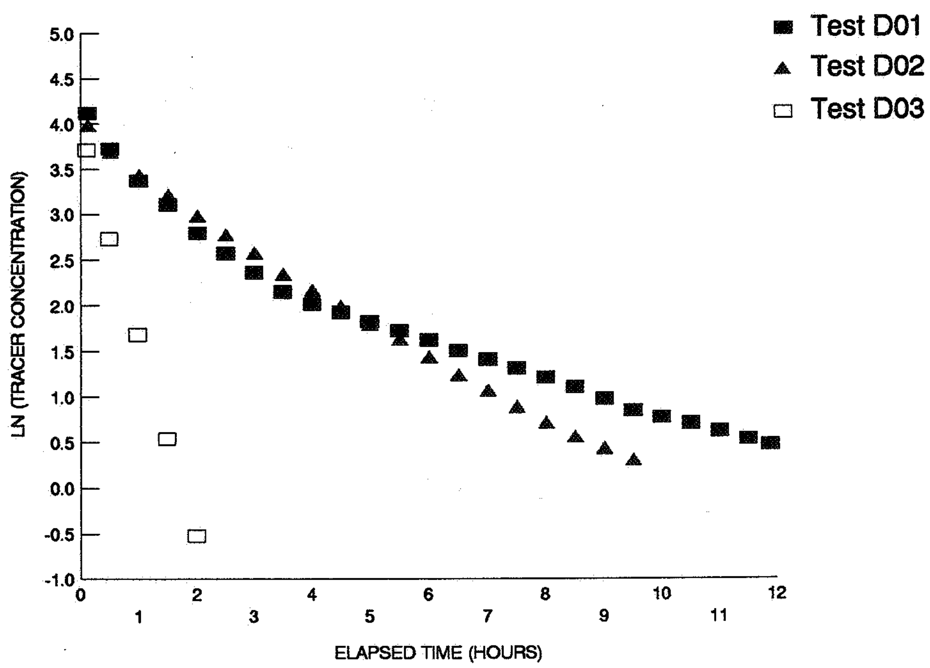


Figure 5: Air Infiltration Rate Versus Windspeed (All Wind Directions) For Factory As-Built and Door+Vents Sealed Cases

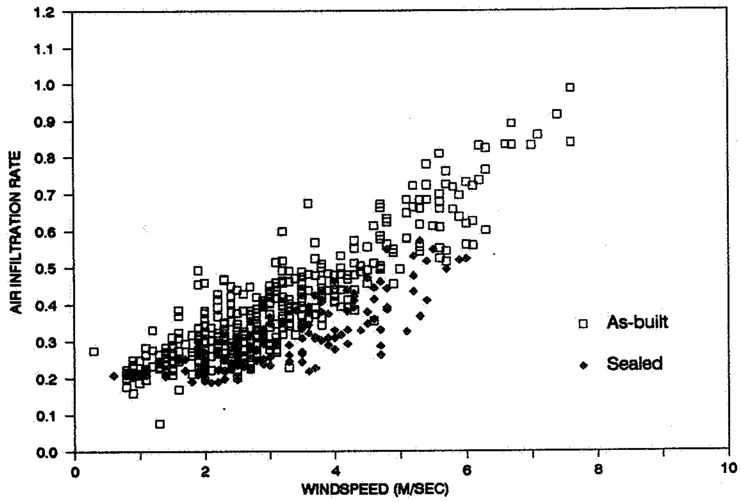


Figure 6: Air Infiltration Rate Versus Windspeed (South Only) For For Factory As-Built And Door + Vents Sealed Cases

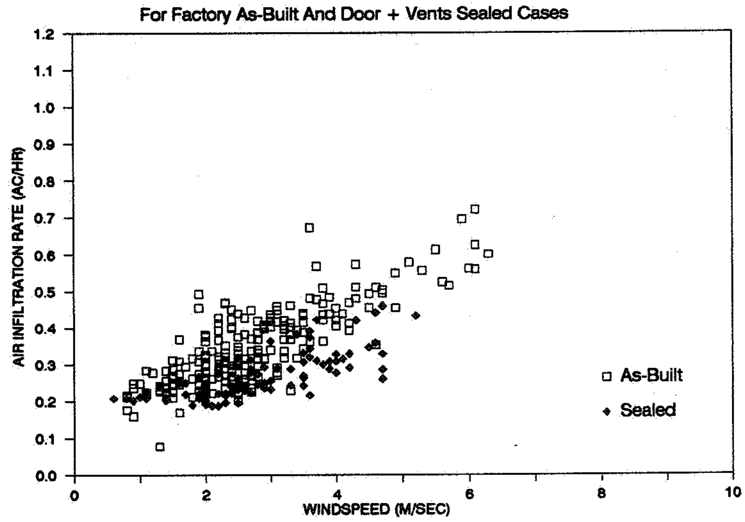


Figure 7: Air Infiltration Rate Versus Windspeed (South West) For Factory As-Built And Door+Vents Sealed Cases

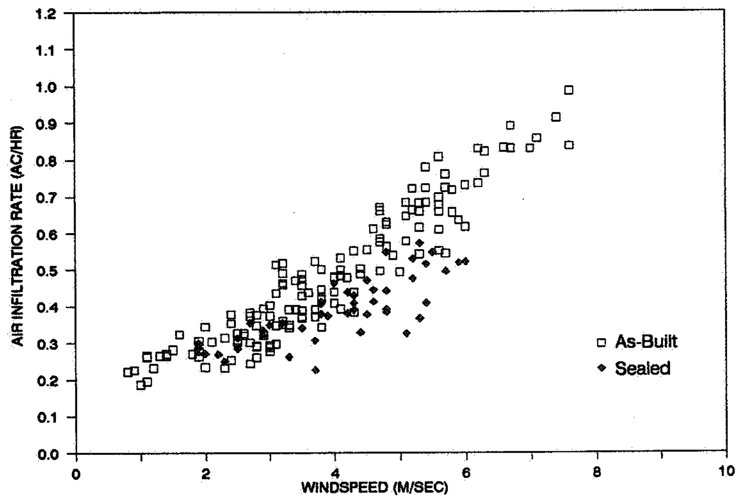


Figure 8: Regression Fits For Air Infiltration Rate Versus Windspeed
All Wind Directions

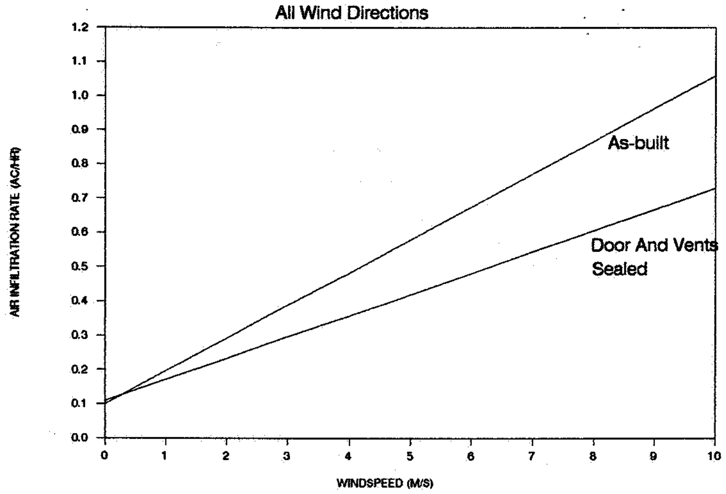


Figure 9: Regression Fits For Air Infiltration Rate Versus Windspeed
Wind Direction: South

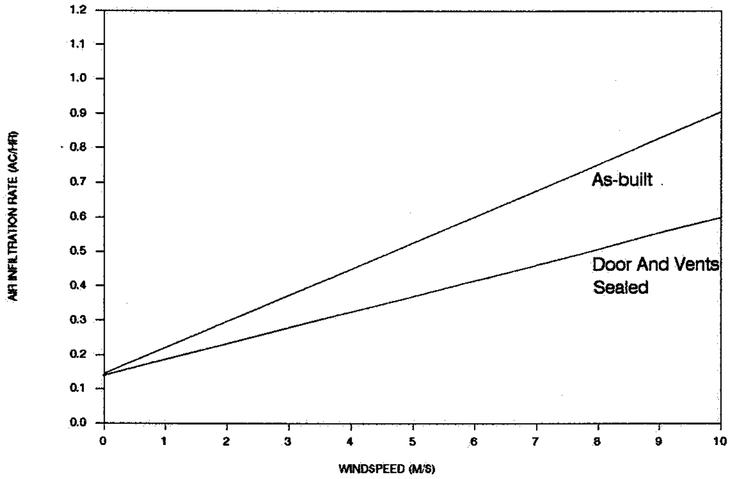


Figure 10: Regression Fits For Air Infiltration rate Versus Windspeed
Wind Direction: South West

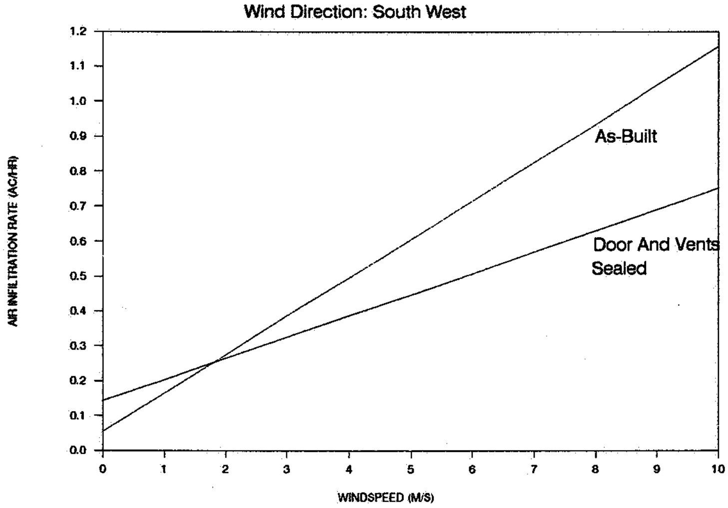


Figure 11: Air Infiltration Rate/Stack Versus Windspeed/Stack (All Directions)
For Factory As-Built and Door+Vents Sealed Cases

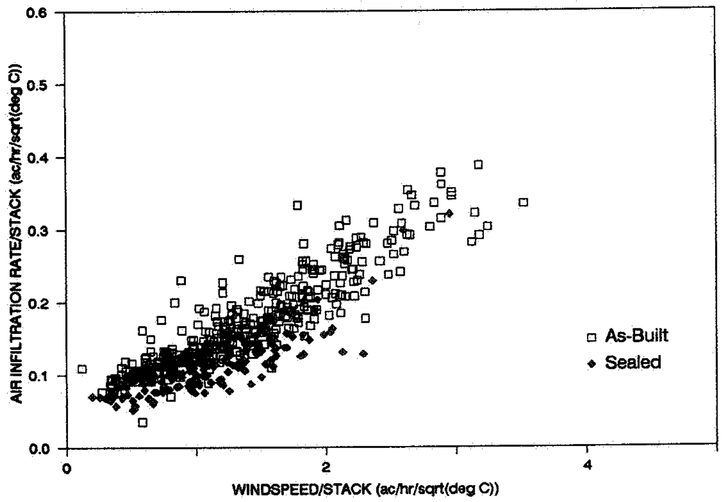


Figure 12: Air Infiltration Rate/Stack Versus Windspeed/Stack (South Only)
For Factory As-Built And Door And Vents Sealed Cases

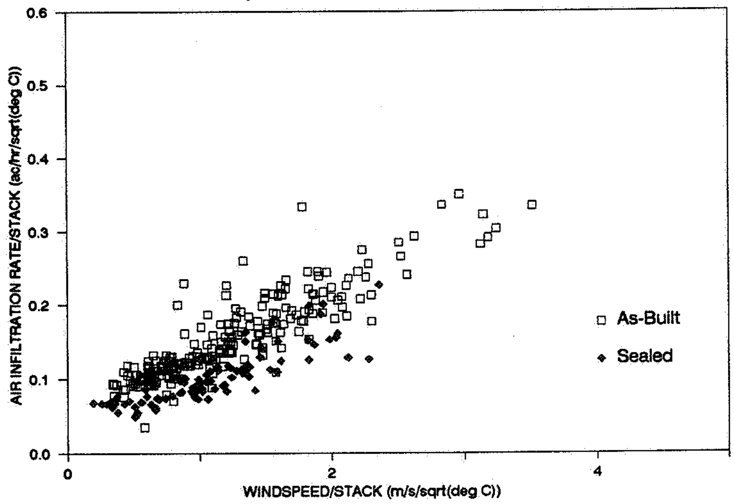


Figure 13: Air Infiltration Rate/Stack Versus Windspeed/Stack (South West)
For Factory As-Built And Door And Vents Sealed Cases

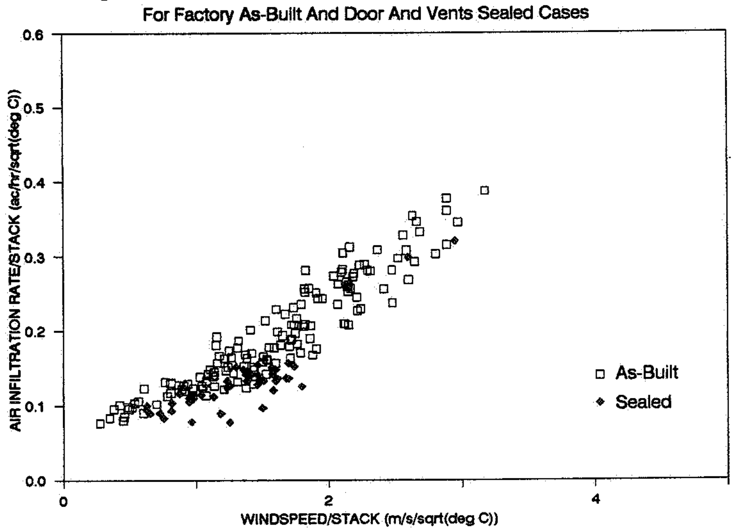


Figure 14: Regression Fit For Air Infiltration Rate As A Function Of Windspeed And Stack. All Wind Directions, Stack = 1

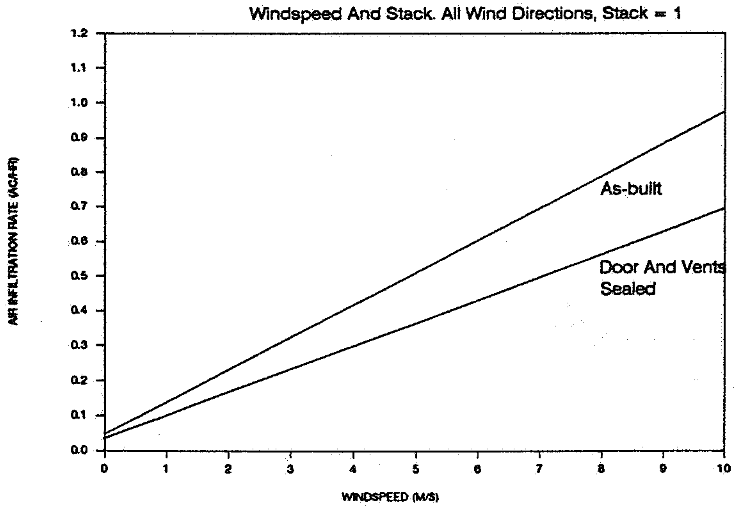


Figure 15: Regression Fit For Air Infiltration Rate As A Function Of Windspeed And Stack. All Wind Directions, Stack = 3

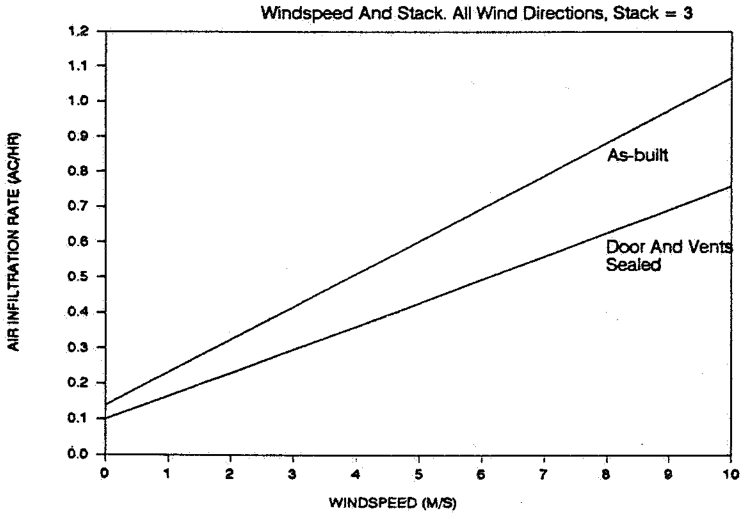
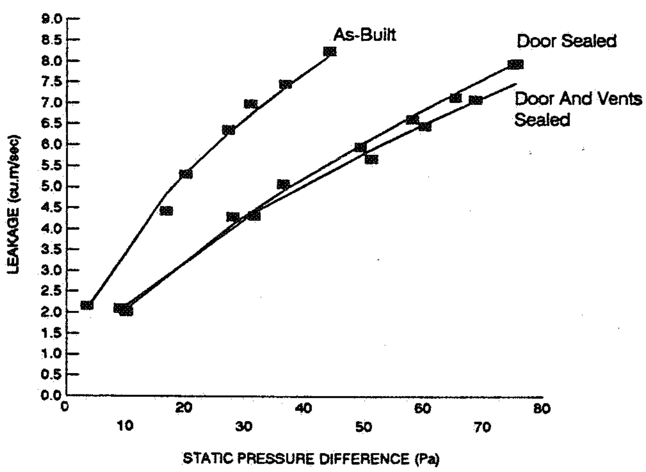


Figure 16: Air Leakage Tests P01, P02 and P03.



PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

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Poster 11

GENERAL FEATURE OF A TWO-DIMENSIONAL ISOTHERMAL MEAN FLOW INSIDE A VENTILATED ROOM WITH A WALL MOUNTED OBSTACLE

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SYNOPSIS

This paper deals with the elaboration and the validation of a user-friendly numerical program (EOL) for the calculation of the ventilation patterns inside industrial premises. After the running-in. In period, "EOL" will be used by the technical staff in charge of ventilation projects.

Here is set out the EOL unit devoted to the calculation of the mean flow inside the rooms. The structure of the software (presently restricted to two-dimensional mean flow configurations) is explained.

The experimental validation of the software is performed through the use of a test configuration set inside an hydraulic bench. The measurements of the mean flow inside the plexiglass scale model are done by a Laser Doppler system.

For two characteristic profiles inside the measuring section, experimental and numerical results are compared. Very fine agreement is obtained. This result is quite encouraging for the development of the EOL programming operation.

1. MAIN PURPOSES

The building of a modern industrial premise requires today an outstanding ventilation system. The optimization of the ventilation patterns is necessary to improve the working conditions inside factories.

The purpose is to develop the preventive concepts for the ventilation technology. The work is to create an engineering tool useful for calculating the main ventilation patterns. This tool will be a programming model (EOL) specially designed to solve air quality problems inside industrial premises. The schedule of conditions includes :
The calculation of the mean flow inside the room (location and size of the recirculating vortices).
The calculation of the pollutant concentration map (when pollutant sources are introduced into the room).
The calculation of the ventilation efficiencies.

The study is divided into two different but complementary parts : the numerical work is to elaborate specialized programming units to calculate the various parameters of the ventilation. Units are based on a turbulent high Reynolds number (k-ε) model. At first, they are elaborated for calculating two-dimensional ("2D") mean flow layouts. The extension for "3D" configurations is under development at INRS. The experimental work consists of the validation of the programming units. It uses an hydraulic water bench inside which scale models of ventilated rooms are tested. The actual air flow through the room is replaced by a water flow. The circulation inside the scale model is carried out by regulated pumps. Of course, it is possible to simulate both "2D" or "3D" flows. The two experimental processings consist of the Laser Doppler Anemometer for the mean velocity measurements, and the video camera system (with specialized image processing) for the pollutant concentration map.

2. THE VENTILATION TEST MODEL

The scale model for the validation of the EOL software is a parallelepipedic plexiglass room [(X = 0 to 520 mm) x (Y = 0 to 280 mm) x (Z = 0 to 360 mm)].

The water circulation through this model is a steady flow with "3D" turbulence. The forced water ventilation flow is extracted from the model through a rectangular outlet window (2 mm x 360 mm) at the bottom of the left side of the model. The rectangular inlet window (10 mm x 360 mm) is located at the top of the right side of the model. The floor mounted obstacle is a square section crossbar (50 x 50 x 360 mm^3) supposed to simulate a desk inside a room or a machine tool inside an industrial premise. The model is built so that the vertical cross section (X x Y) is not dependent on Z.

The experimental and numerical simulations will concern the central vertical cross section of the model. In that section, the mean flow is supposed to be two-dimensional (geometrical symmetry). The measuring section is drawn in figure 1.

Considering the experiment, the mean Reynolds number (built with the inlet mean velocity U and the height H of the model) is 40 000. Considering the actual air flow, the corresponding parameters are :

$$U = 1.5 \text{ m/s and } H = 2.80 \text{ m}$$

3. STRUCTURE AND USE OF EOL

The three major features of EOL is that it is user-friendly, that it contains several tools to analyse the results with an hygienist point of view and that it is in the process of being validated for ventilation situation. Given a particular configuration, you easily enter its geometry and the associated boundary conditions. The program helps to find the appropriate grid, and then computes the flow[1]. Moreover, the possibility of computing local ages, local purging flow rates, time evolution of local concentrations after a sudden contaminant release, local or global concentrations or ventilation efficiencies, and so on is given. EOL is at present restricted to two-dimensional flows in cartesian coordinates but the cylindric and the three dimensional versions are under development[2/3].

The core of EOL is largely inspired by the works of the Imperial College Group[1]. It solves the mean transport equations for U-momentum, V-momentum, mass, turbulent energy and turbulent dissipation. The eddy viscosity is computed using a (k-ε) model[4]. The partial differential equations are transformed into finite differences equations in implicit and conservative form using the hybrid scheme. To satisfy continuity, the SIMPLE[5] algorithm is used.

The mean velocity computation unit of EOL has been used to calculate the mean flow for the considered configuration. The major assumptions for the calculation are : two-dimensional mean flow, steady flow and constant temperature.

The computed numerical mean velocity map is drawn in figure 2. The flow is mainly divided in two big recirculating vortices, separated by the wall obstacle. Let's also underline the importance of the horizontal inlet wall jet.

4. VELOCITY MEASUREMENTS

The velocity measurements inside the scale model are carried out by the DANTEC one-component Laser Doppler Anemometer (LDA). The LDA uses the Doppler shift of light scattered by the moving particles mixed with the flow to calculate their own velocity, and then find the flow velocity itself. The probe volume is the crossing of the two monochromatic coherent laser beams. The DANTEC system consists of the 55X backscatter modular optics and the 57 N 10 frequency analyser. The 55X can rotate to fix the direction of the measured mean velocity component and a Bragg cell is used to calculate the sign along that direction.

The direction of the laser beams going through the model can be either horizontal or vertical. The first position is used to measure the components $[\overline{U}, \overline{V}]$ of the mean velocity vector inside a vertical section. The vertical position is used to measure the two horizontal components $[\overline{U}, \overline{W}]$. The steady flow hypothesis makes possible to measure the three components $[\overline{U}, \overline{V}, \overline{W}]$ in sequence for every point inside the scale model.

The first experiment is the validation of the "2D" mean flow hypothesis inside the measuring section.

The measurements of the transversal velocity component [$\overline{W}$] give that this hypothesis is quite true in the first half-section (next to the inlet window). The results are not so sucessful in the second half-section (next to the outlet window). "3D" circulations do exist inside the experimental mean flow, particulary near the wall just behind the floor obstacle.

The second step is the measurement of the experimental mean velocity map [$\overline{U}$, $\overline{V}$ components] inside the test section.

The measurements, consisting of more than 200 points, are drawn in figure 2.

Special studies about the relative accuracy of the measurements have been performed. They basically consist of comparing several same experiments (repetition) and calculating the convergence of the statistical estimators [$\overline{U}(t)$, $\overline{V}(t)$]. The conclusion is that the relative accuracy for the $\overline{U}$ and $\overline{V}$ mean velocity components is generally better than 20 %.

5. EXPERIMENTAL VERSUS NUMERICAL RESULTS

For a better understanding of the problems, we have decided to select two characteristic velocity profiles for the comparison.

The profiles deal with the horizontal [$\overline{U}$] mean velocity component. Their location is drawn in figure 3.

The horizontal profile 1 describes the inlet wall jet and the base of the recirculating vortex behind the obstacle. It is exactly located 15 mm away from the wall. The comparison for the $\overline{U}$ velocity component is drawn in figure 4. Inside the wall jet, the major difference between experimental and numerical results is a 9 % overestimation for the experiments. Behind the obstacle, the maximum gap is a 25 % relative overestimation : this is quite important, but let's say that the $\overline{U}$ velocity is nearly zero in this area.

The second horizontal profile is located in the upper area of the measuring section (Y = 235 mm). It is used to describe the top of the two big recirculating vortices. The comparison for the $\overline{U}$ mean velocity is drawn in figure 5. We can see again that the experimental results are overestimated (20 à 25 %) compared with the numerical data.

Generally considering the two profiles, we must first admit that there are some differences between the experimental and the numerical results. Those relative differences are not so much important (10 to 20 %) and are often as high as the relative accuracy of the experimental method.

The very important point of the comparison is that the shape of the two profiles is exactly the same for the two methods.

6. CONCLUSION

The EOL software unit for calculating the ventilation flow inside premises has been tested through the use of a characteristic configuration,including a two dimensional mean flow.

The experimental point is to obtain a quasi two-dimensional mean flow in the central section of the scale model. This has been only partially achieved. Nevertheless, the comparisons between experimental and numerical results concerning two characteristic profiles are successful. It must be pointed out that the general shape of the profiles is exactly the same.

Similar results have been obtained concerning other profiles (not drawn in this paper). This example is quite encouraging for the future developments of the program : elaboration of advanced methods for the validation of the numerical software (measurement of the kinetic energy of the turbulence and the turbulent dissipation per unit of mass), study of a "3D" mean flow configuration simulating a living ventilation configuration.

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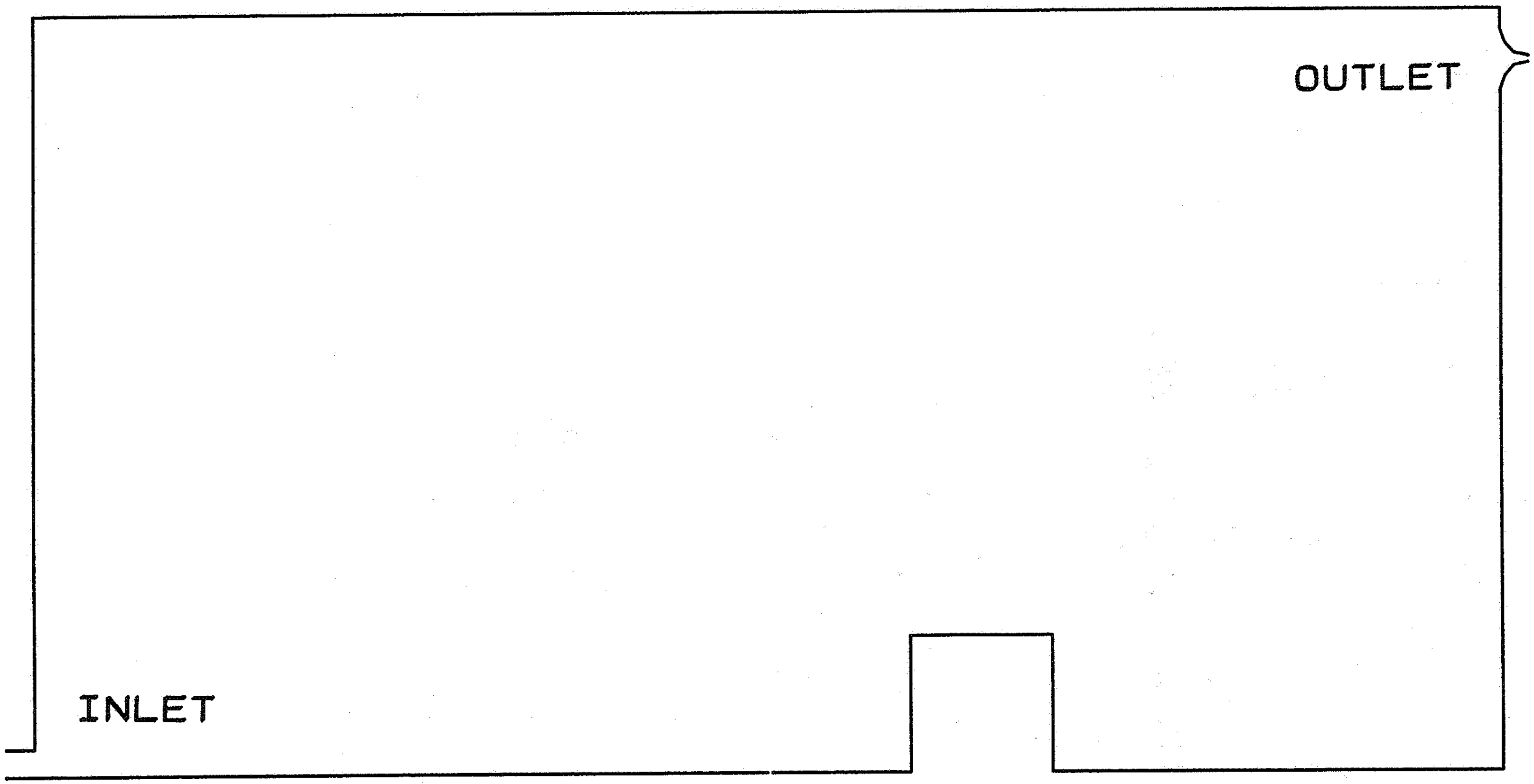
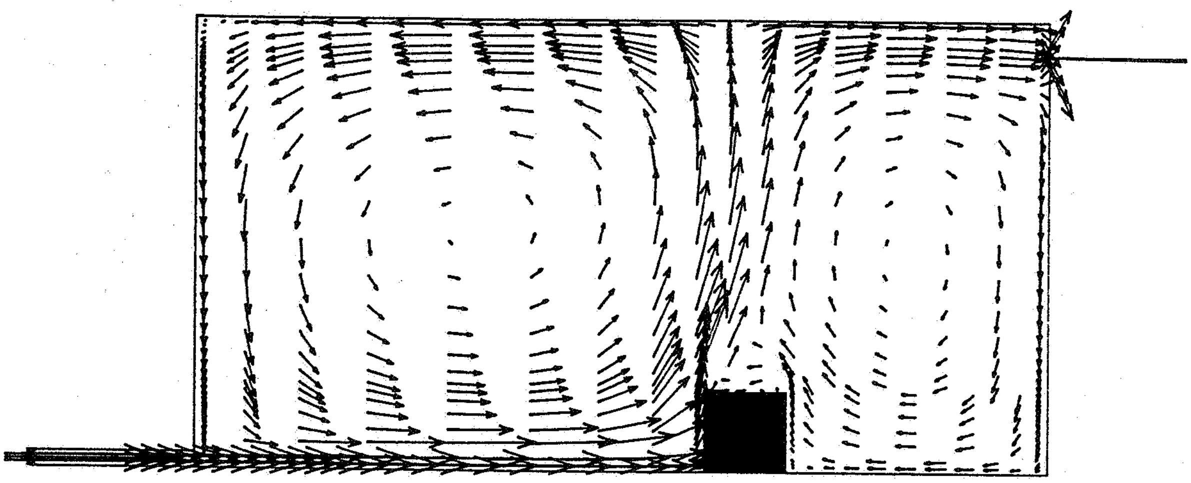


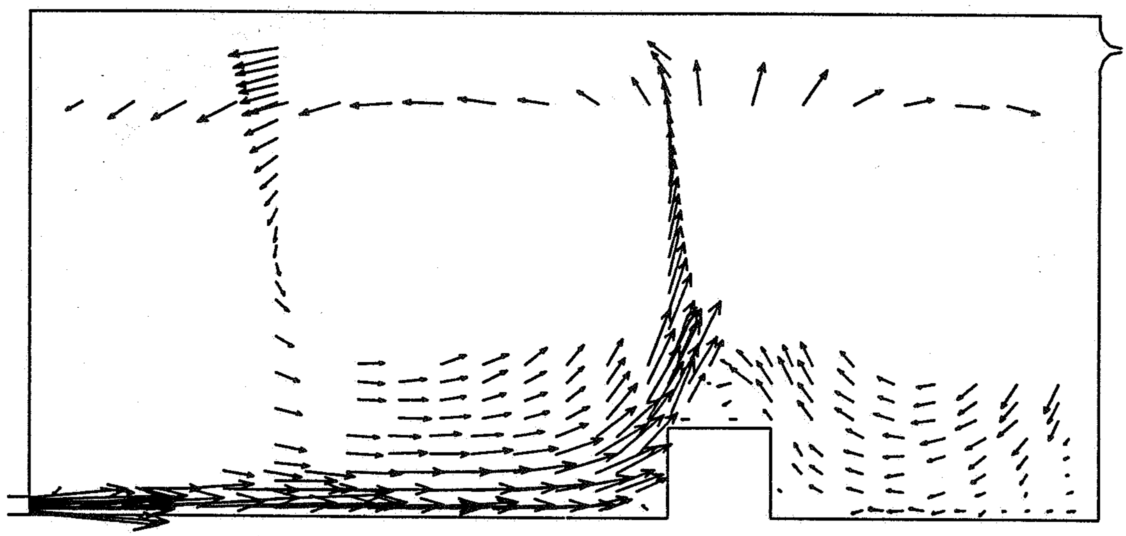
THE MEASURING SECTION INSIDE THE ROOM

('2D' STRUCTURE OF THE MEAN FLOW)

FIGURE 1

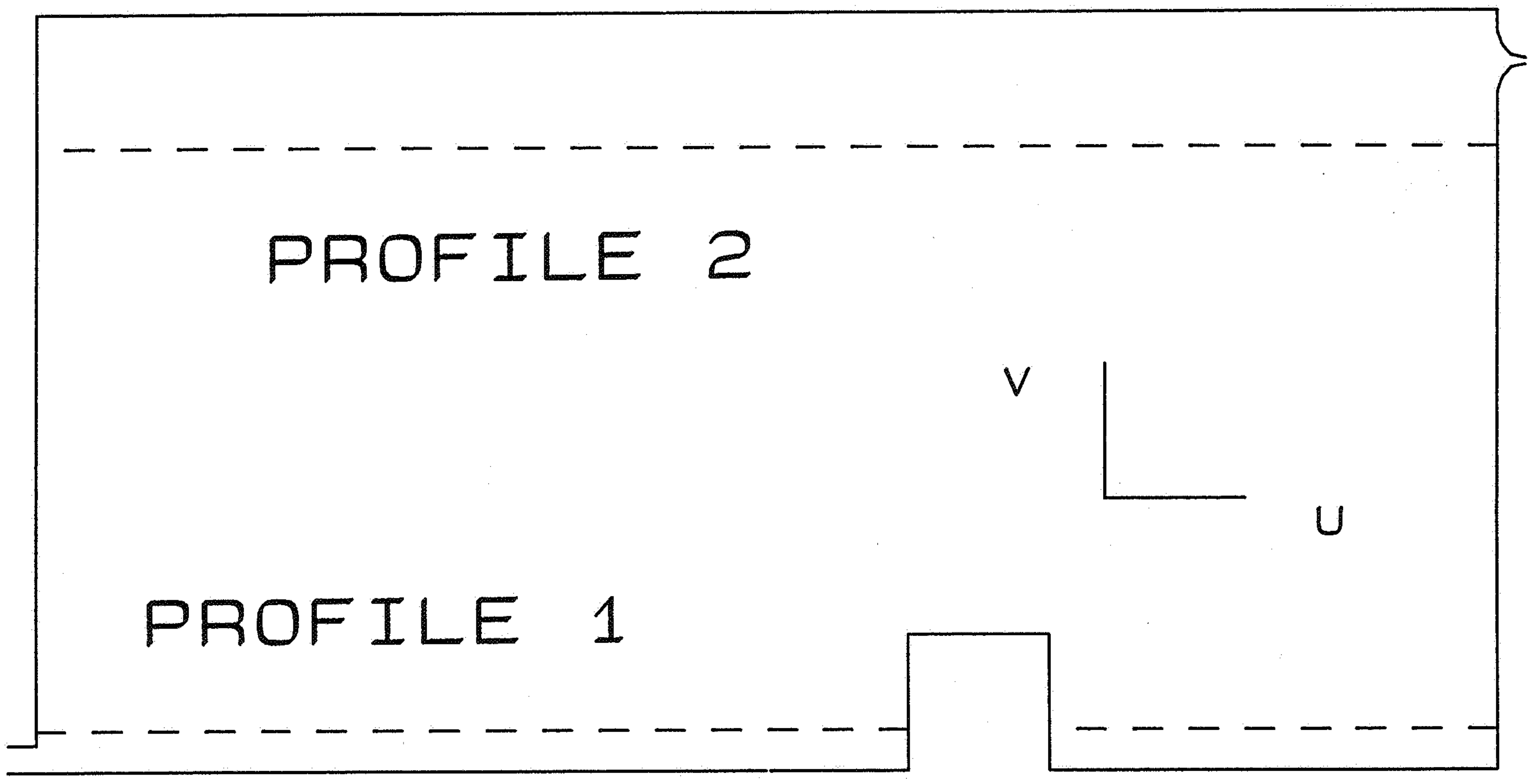


NUMERICAL MEAN VELOCITY MAP



EXPERIMENTAL MEAN VELOCITY MAP

FIGURE 2



LOCATION OF THE PROFILES

FIGURE 3

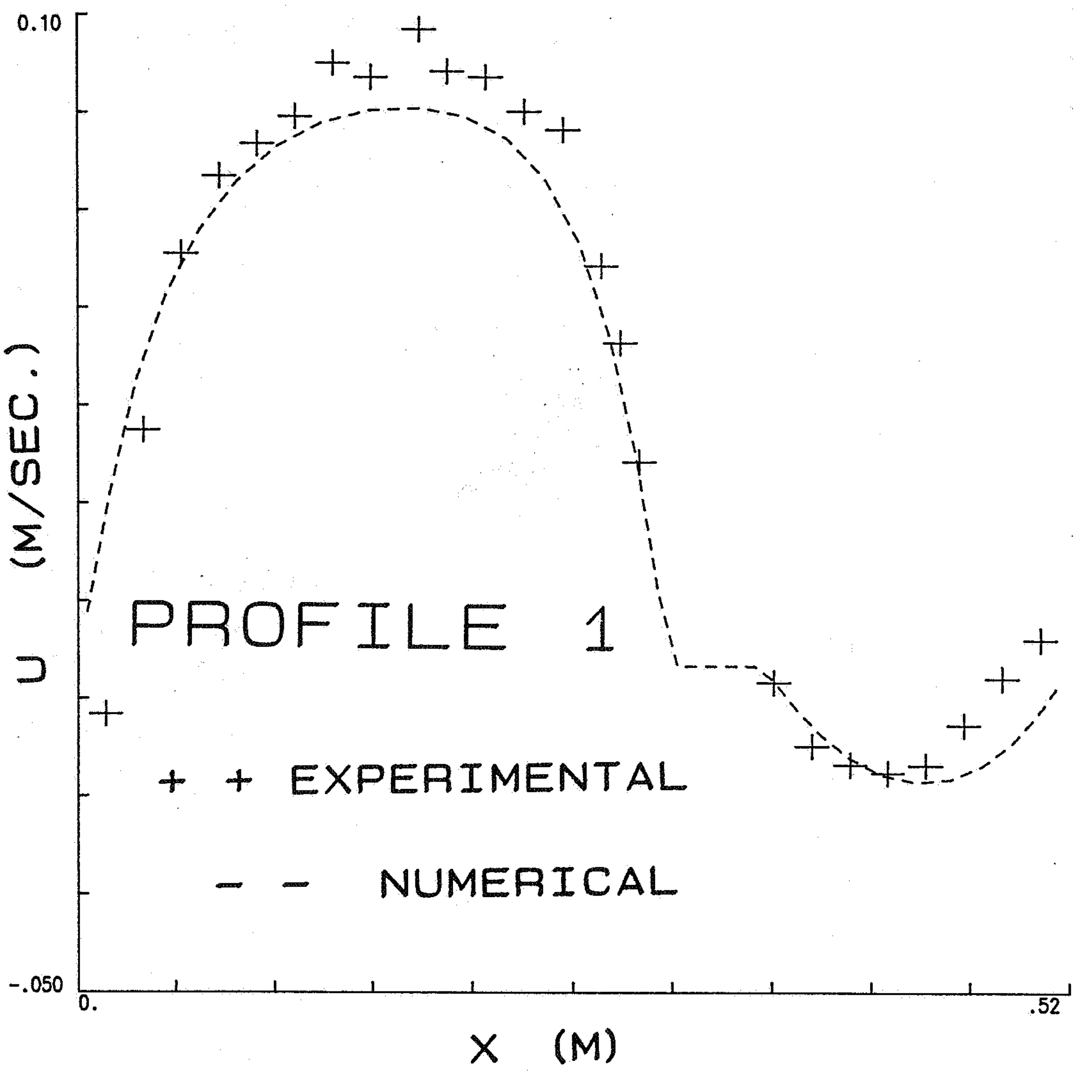


FIGURE 4

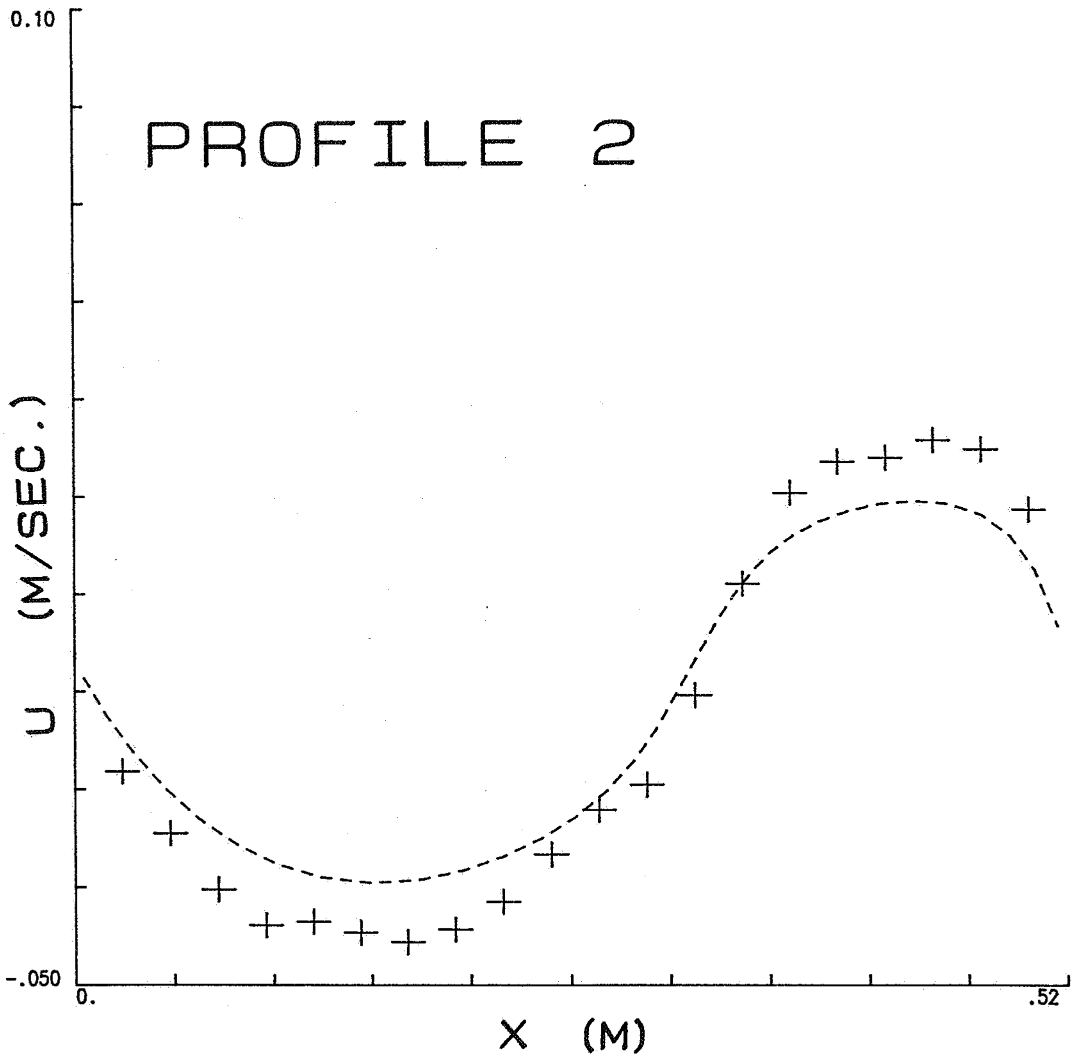


FIGURE 5

PROGRESS AND TRENDS IN AIR INFILTRATION AND VENTILATION RESEARCH

10th AIVC Conference, Dipoli, Finland
25-28 September 1989.

Poster 12

BUOYANCY-DRIVEN AIR FLOW IN A CLOSED HALF-SCALE STAIRWELL MODEL: VELOCITY AND TEMPERATURE MEASUREMENTS

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SUMMARY

This paper describes an experimental study of the buoyancy-driven flow and the associated energy transfer within a closed, half-scale stairwell model. It provides new data on the velocity, temperature, volume and mass flow rates of the air circulating between the upper and lower storeys. The results are presented for various heat input rates from the heater, located in the lower floor. For most of the data presented, heat transfer to the surrounding atmosphere takes place through the side walls. However, the case of insulated side walls is also included and the effects on the parameters of interest are discussed. The velocities were measured using hot-wire anemometers of a temperature compensated type, and the temperatures were measured using platinum resistance thermometers. These measurements were supported by flow visualisation using smoke. The paper also provides data on the rate of leakage through the stairwell joints, measured using a concentration decay method.

LIST OF SYMBOLS

A	Throat area (0.462 m^2)
C, c	Concentration inside and outside stairwell (ppm).
c_p	Specific heat at constant pressure ($J\ kg^{-1}\ K^{-1}$).
DT	Differential temperature ($T_H - T_C$), (C deg.).
g	Gravitational acceleration ($m\ s^{-2}$).
h	One half the height of the stairwell model (m).
k	Heat conductivity ($J\ m^{-1}\ s^{-1}\ K^{-1}$).
$\dot{m}_u$, m_d	Upflow and downflow mass flow rates ($kg\ s^{-1}$).
$\dot{m}_\ell$	Leakage rate ($kg\ s^{-1}$).
$\dot{Q}$	Rate of supply of heat to the stairwell (W).
$\dot{Q}_s$	Volume flow rate through leakage ($m^3\ s^{-1}$).
T_{av}	Average temperature within the stairwell (^oC).
T_H, T_C	Mean temperatures of warm upwards-flowing air and cold downwards-flowing air, respectively (^oC).
U_{maxu}	Maximum velocity of the flow moving up the stairwell ($m\ s^{-1}$).
U_{maxd}	Maximum velocity of the flow moving down the stairwell ($m\ s^{-1}$).
V_s	Stairwell volume (m^3).
$\dot{V}_m$	Arithmetic average of the volume flow rates, of the upflow and downflow ($m^3\ s^{-1}$), $\dot{V}_m = (\dot{V}_u + \dot{V}_d)/2$.
$\dot{V}_u$, $\dot{V}_d$	Upflow and downflow volume flow rates ($m^3\ s^{-1}$).
w	Stairwell width (m).
z	Direction along the throat area (see Figure 1).

GREEK SYMBOLS

β	Coefficient of thermal expansion (K^{-1})
ρ	Fluid density ($kg\ m^{-3}$)
ν	Kinematic viscosity ($m^2\ s^{-1}$)
μ	Dynamic viscosity ($kg\ m^{-1}\ s^{-1}$)

DIMENSIONLESS GROUPS

Fr	Froude number	$= \dfrac{\dot{V}_m}{A\ (gh)^{\frac{1}{2}}}$
Gr	Grashof number	$= \dfrac{g\ \beta\ DT\ Ah}{\nu^2}$
Pr	Prandtl number	$= \dfrac{\mu\ C_p}{k}$
Ra	Rayleigh number	$=$ Gr.Pr
Re	Reynolds number	$= \dfrac{\dot{V}_m}{\nu\ A^{\frac{1}{2}}}$
St	Stanton number	$= \dfrac{\dot{Q}}{\rho\ C_p\ T_{av}\ A(gA)^{\frac{1}{2}}}$

DEFINITIONS

Throat area : The area shown by DD' in Figure 1.

Side wall : Area defined by ACDEFDHIA in Figure 1.

1. INTRODUCTION

Obtaining a better understanding of the mechanism and characteristics of air movement within stairwells, in private dwellings, hotels and public buildings such as hospitals and underground stations, is important in relation to wide ranging problems such as fire safety, energy saving and movement of micro-organisms and contaminants. Improvements in the basic understanding can be obtained by carrying out simple tests on small-scale models, setting up simple analytical models or using high-level computational models.

Fires account annually for thousands of deaths and millions of pounds in loss of property. Moodie et al.(1) have carried out experiments on a one-third scale model of an escalator to investigate the fire which occurred in King's Cross underground station in 1987. The results showed that, following the ignition, the flame was soon established across the full width of the escalator channel. The flame front then remained low in the escalator channel as the fire developed and then progressed in the escalator. Zohrabian et al.(2) have shown that sloping the stairwell ceiling speeds up the migration of smoke from the lower to the upper compartment, hence helps in the spread of fire.

Most rooms in hospitals are connected by corridors and stairwells. The importance of movements of micro-organisms in a hospital stairshaft has been shown by several workers in recent years. The measurements by Münch et al.(3) indicate that transport of micro-organisms depends on the temperature differential within the stairwell. The experimental measurements of Zohrabian et al.(2) on a half-scale stairwell model showed that the volume flow rate of the air flow from the lower floor to the upper was increased by increasing the heating load or changing the stairwell geometry. Hence micro-organisms, as well as toxic agents or contaminants which can cause wound infection, can be transferred to the upper floors by way of the stairwell. Therefore, special arrangements or appropriate stairwell design is needed within hospital buildings, in order to minimize the flow of unwanted germs or contaminants to the higher floors.

A number of other useful experimental studies are reported by Brown and Solvason(4), Shaw(5), Shaw and Whyte(6), Feustel et al.(7) Marshall(8,9), Mahajan(10), Riffat et al.(11) and Riffat and Eid(12). Analytical modelling of relevant flows are reported by Reynolds and Reynolds et al.(13,14), Nevrala and Probert(15), Liddament(16). Applications of Computational Fluid Dynamics to flows relevant to buildings have increased in recent years. But, to the best of the present authors' knowledge, the only applications to stairwell flows have been by Zohrabian et al.(17) and Simcox and Schomberg(18). The latter authors used HARWELL-FLOW3D code to investigate the King's Cross fire.

The present work extends the previous study by Zohrabian et al.(2) by providing new data on the characteristics of buoyancy-driven flows in a half-scale stairwell model.

2. EXPERIMENTAL RIG AND INSTRUMENTATION

A schematic diagram of the half-scale stairwell model is shown in Figure 1. The details of instrumentation can be found in Ref. [2]. Therefore, only a brief reference to the instrumentation and the improvements of the original model is made here.

The velocities were measured using hot wires of temperature compensated type having a time constant of about 1s and accuracy of less than 10 per cent for the range of velocities measured within the stairwell. The temperatures were measured using platinum resistance thermometers having a time constant of 3 min and accuracy of $\pm$ 0.25^{o}C. The probes were fixed to the walls by specially designed clamps. The signals from these probes were transferred to an Apple computer for processing, via signal conditioning electronics and Analogue-to-Digital converters.

The modifications to the rig described in reference [2] were:

(i) The light bulbs used as heat source were replaced by an electric radiator with surface area of 0.57 m x 0.659 m and with a loading power of 1 kW.

(ii) Both side walls were made of Perspex of 10 mm thickness (originally one of the walls was made of Perspex of 12.5 mm thickness and the other made of wood of 18 mm thickness). This ensured better thermal symmetry in the stairwell.

3. RESULTS

3.1. Flow characteristics

A two-dimensional view of the flow pattern is shown in Figure 2. The main recirculating flow, recirculation zones, and upward and downward flows in the stairway can be seen. Comparison with previous work [2] showed that, although some changes had been introduced into the design of the rig and completely different type of heat source was used (see section 2), the overall characteristics of the flow did not change significantly. For further detail see reference [2].

3.2. Velocity and temperature profiles in the throat area

Figures 3 and 4 show, respectively, the velocity and temperature distributions at various distances from the side wall and for various heat input rates. The results indicate two distinct regions: one associated with the warm upflow, in the upper part of the throat area, and one with the cold downflow, in the lower part. The results indicate an increase in velocity to a maximum very close to the ceiling, after which it drops to zero at the ceiling. The maximum velocity varies from about 0.24 m/s at 100 W to about 0.54 m/s at 900 W heat input rate. Figure 4 shows that the temperature varies approximately linearly from its lowest value near the stairs to a maximum very near to the ceiling of the throat area. The same behaviour can be seen for other heat input rates. As expected, the increase in heat input rate has resulted in an increase in velocity and temperature of the recirculating flow. The temperature data also show that the maximum temperature varies from approximately 31^{o}C at 100 W to about 56^{o} at 900 W heat input rate.

In order to obtain some idea of the three-dimensional behaviour of the flow, the above results are also presented in an alternative way. Figures 5 and 6 show, respectively, the velocity and temperature profiles at various distances from the side wall. Figure 5 shows that the velocity profiles can be considered uniform over about two-thirds of the width of the stairwell. Therefore, the overall fluid flow may be considered as two-dimensional. Figure 6 shows higher temperatures for higher heat input rates. Also, the temperature profiles show higher degree of uniformity, across the stairwell, for higher input rates. This uniformity is even more pronounced in the upper region of the throat area. However, it is interesting to note that, although, as expected, the lowest temperatures were measured at position w/6, the highest temperatures were not recorded at the mid-width of the stairwell (except for the 100 W heat input rate), but at w/3. This indicates complex flow behaviour near the stairs.

3.3. Leakage Measurement

Figure 7 shows the results of the tracer-decay test for a 100 W heat input rate. It shows variations of CO_2 tracer gas concentration against time. The negative slope of the line is equal to the air change rate, given by

$$\dot{Q}_s/V_s \quad = \quad \text{Air change rate}$$

where V_s is the stairwell volume, and $\dot{Q}_s$ is the volume flow rate through leakage.

However, for the actual volumetric air flow rate, or air leakage rate in this case, the stairwell volume was multiplied by the air change rate. Table 1 gives the air leakage rates through the stairwell joints, for various heat input rates. The results indicate that the leakage rate increases with heat input rate. It should be noted that the calculation of leakage mass flow rate was based on the arithmetic average T_{av} and room temperature. The calculation of leakage heat flow rate was based on the difference between T_{av} and the room temperature.

3.4. Maximum velocities, mean temperatures and flow rates

Table 2 shows the maximum velocities and mean temperatures in the upflow and downflow streams at various distances from the side wall and for various heat input rates. The results indicate that the maximum air velocity varies slightly across the width of the throat area. It should be noted that calculations of the mean temperatures were based on the arithmetic mean of the temperatures measured in the warm upflow and cold downflow.

Table 3 shows the volume and mass flow rates for various heat ihput rates. The results show that the volume and mass flow rates increase as the heat input rate increases. The variation of the average volume flow rate $\dot{V}_m$, with the heat input rate $\dot{Q}$, can be written in the following form, as suggested by Reynolds[13].

$$\dot{V}_m \propto \dot{Q}^n$$

It was found that n=0.22. This is in agreement with the previous results [2] and also is generally consistent with the model described by Reynolds(13), in which the value of 0.25 was suggested.

Assuming that the inflow and the outflow through cracks take place in the lower and upper compartment, respectively, one expects that $\dot{m}_u = \dot{m}_d + \dot{m}_\ell$. The results of table 3 show that the two sides of above relationship differ by less than 5 per cent. The possible reasons for this discrepancy are given in section 4.3.

3.5. Rate of heat loss through the stairwell

Table 4 gives the rate of heat loss from the stairwell for various heat input rates. The results show that the heat losses through the several stairwell boundaries vary approximately linearly with the heat input rate. The results also indicate that over 50 per cent of the heat is lost through the stairwell side walls (for further discussion see section 4.4).

3.6. Effect of insulated side walls on the results

Table 5 shows the rate of heat loss from the stairwell with insulated side walls. The results show a substantial increase in the rate of heat transfer through the other walls. For example, the rate of heat transfer through the upper compartment ceiling has tripled. The results (not given here) also showed that higher velocities and temperatures resulted, both in the upflow and in the downflow streams.

4. DISCUSSION

4.1. Characteristic dimensionless numbers

The characteristic dimensionless numbers relevant to the natural convection in stairwell flows are Froude, Stanton, Reynolds and Grashof numbers. Table 6 gives values of the dimensionless numbers for the half-scale stairwell geometry. Full discussion of their range for the half-scale model is given by Reynolds et al.(14). The corresponding values for full-scale stairwells can be found using the scaling principles set out by Reynolds(13). These are based effectively on Froude scaling. For example, for a Froude number about 1.4 times the prototype value, requires that the ratio of the energy input of the model to the prototype be 0.24. With this Froude scaling, the magnitudes of the temperatures and velocities become identical in full and model scales. Using this scaling for various heat input rates, the dimensionless numbers at the throat area were in the ranges :

$$0.01385 < Fr < 0.0313 \ , \quad 0.1765 \times 10^{-3} < St < 1.5845 \times 10^{-3}$$

$$2060 < Re < 4400 \ , \quad 9.7330 \times 10^{7} < Gr < 3.8190 \times 10^{8}$$

on the half scale and

$0.0089 < Fr < 0.0224$, $0.130 \times 10^{-3} < St < 1.1668 \times 10^{-3}$

$4160 < Re < 8900$, $8.024 \times 10^{8} < Gr < 3.1480 \times 10^{9}$

on the equivalent full scale.

It should be noted that :

(i) The fluid properties were evaluated at the arithmetic mean of the 45 temperature readings distributed within the upper and lower compartments.

(ii) The volume flow rates used for the calculation of Reynolds and Froude numbers were based on the average of the upflow and downflow volume flow rates.

The flow characteristics in the lower compartment may be understood by reference to the Rayleigh number. One of the most common, and simplest natural convection problems occurs when a vertical heated flat plate transfers heat to a still, colder surrounding fluid. This is approximately the case close to the heater, where, depending upon fluid properties and the thermal gradient, transition to turbulent flow occurs when Rayleigh number is about 10^{9}. The Rayleigh number, based on the heater height, the difference between the surface temperature of the heater and that of the surrounding fluid, and the fluid properties evaluated at the arithmetic mean of the temperatures measured in the lower compartment, was found to be within the range 4.361×10^{8} to 1.763×10^{9}, for 100 W to 900 W heat input rates, respectively.

4.2. The velocity and temperature distributions in the throat area

The flow was symmetrical with respect to the mid-plane of the stairwell. This was examined by measuring temperatures and velocities at the throat area at w/2, w/3 and w/6 distances from both side walls of the stairwell. There was a discrepancy of less than 2 per cent between the temperatures and less than 3 per cent between the velocities measured from the two sides. Therefore, a symmetrical condition was assumed with respect to the mid-plane of the throat area, and the measurements were carried out in only one half of the stairwell.

4.3. Volume and mass flow rates and factors affecting their accuracy

Table 3 gives the flow rates in the upflow and downflow at the throat area for various heat input rates. As mentioned in section 3.4, the results indicate a discrepancy of less than 5 per cent between the upflow and downflow (including leakage) rates. This discrepancy can be attributed mainly to experimental errors and the calculation procedure using Simpson's rule. Moreover, the zero velocity did not usually coincide with the mid-height of the throat area. However, this was assumed for the calculation of volume flow rates, as determination of the actual position, which varied with time, was difficult. Finally, the nearest point at which the velocity could be measured was 10 mm from the wall. Therefore, for locations closer to the wall the velocities were measured using a less accurate air velocity meter.

4.4. Heat transfer rates through the stairwell walls and joints

The heat transfer rates through the stairwell are given in Table 4. The results indicate that more than half of the heat input to to the stairwell is lost through the side walls. The results also indicate that there is a difference of less than 3 per cent between the measured heat input rate to the stairwell and the rate of measured heat loss from the stairwell. This error in the heat balance can be attributed to the following facts :

(i) The calculation of the heat loss was based on the surface temperatures measured at the mid-plane of the stairwell, and were assumed uniform across its width. However, limited measurements showed that for a heat input of 600 W, the surface temperature varied by 0.4C deg.

(ii) The heat input rate to the radiator was determined from recordings of voltage and current. The accuracy of the heat input rate was about 2 per cent.

In order to investigate the effect of the heat transfer rate through the side walls on the velocity and temperature profiles at the throat area, the side walls of the stairwell were insulated. Table 5 gives the heat transfer rates for this case. The results (not shown here) indicated that insulation of the side walls resulted in an increase in temperatures in the throat area of approximately 3C deg. for 300 W and between 5C deg. to 8C deg. for 600 W heat input rates, respectively. Also, the velocities at the throat area increased by up to 13 per cent in the upflow and up to 15 per cent in the downflow, for 300 W and 600 W heat input rates, respectively. The results also indicated that insulation of the side walls resulted in a significant increase in the rate of heat transfer through the other walls. For example, the rate of heat transfer through the upper-compartment ceiling increased by about 60 per cent for both 300 W and 600 W heat input rates. Furthermore, the Reynolds number increased by 20 per cent, and the Grashof number by about 6 per cent.

5. CONCLUDING REMARKS

This paper has addressed a number of questions left open in the authors' earlier studies of stairwell flows. The symmetry of the flow, and influences which give rise to departures from it, have been investigated. Tracer-gas determinations of the leakage rate from the nominally sealed test rig have been carried out, and the results have been used to analyse discrepancies in other measurements. The influence of thermal boundary conditions has been studied, and the distribution of heat outflows through the boundaries of the stairwell system has been clarified.

Taken together, these additional measurements provide a more comprehensive and convincing picture of the processes involved in this family of buoyancy-driven flows.

ACKNOWLEDGEMENTS

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HEAT INPUT RATE $\dot{Q}$ (W)	AIR CHANGE RATE $\dot{Q}_s/V_s$ per hour	LEAKAGE RATE $\dot{Q}_s$ (m^3/s)	LEAKAGE RATE $\dot{m}_\ell$ (kg/s)	EXTERNAL TEMP. T_R (^oC)	EXTERNAL CONCENTRATION (PPM)
100	0.0659	5.42×10^{-5}	6.45×10^{-5}	19.5	150-160
300	0.0998	8.21×10^{-5}	9.69×10^{-5}	21.0	150-160
600	0.1085	8.92×10^{-5}	1.04×10^{-4}	21.5	180
900	0.1575	1.29×10^{-4}	1.49×10^{-4}	21.5	280

Table 1: Rate of leakage through the stairwell joints, for various heat input rates.

DISTANCE FROM SIDE WALL (m)	$\dot{Q}$ (W)	U_{maxu} (m/s)	U_{maxd} (m/s)	T_H (^oC)	T_C (^oC)
w/2	100	0.17	0.24	29.7	28.3
	300	0.28	0.31	35.4	32.1
	600	0.36	0.36	43.6	36.8
	900	0.42	0.54	50.1	41.8
5w/12	100	0.16	0.23	29.8	28.0
	300	0.27	0.27	35.5	31.6
	600	0.34	0.36	43.6	37.3
	900	0.41	0.54	50.4	41.5
w/3	100	0.17	0.22	29.7	27.8
	300	0.27	0.27	35.5	31.6
	600	0.34	0.36	43.4	37.2
	900	0.46	0.50	50.7	41.1
w/4	100	0.17	0.21	29.8	27.3
	300	0.26	0.23	35.5	31.1
	600	0.32	0.32	43.2	37.0
	900	0.38	0.47	50.6	42.5
w/6	100	0.17	0.18	28.0	26.8
	300	0.24	0.21	35.0	30.2
	600	0.31	0.31	42.5	36.2
	900	0.35	0.47	49.6	41.4
w/12	100	0.17	0.16	27.6	26.0
	300	0.24	0.18	32.8	28.2
	600	0.31	0.31	39.9	34.7
	900	0.31	0.47	46.3	37.7

Table 2: Maximum velocities and mean temperatures in the upflow and downflow streams (at the throat area) at w/2, 5w/12, w/3, w/4, w/6, w/12 for various heat input rates.

$\dot{Q}$ (W)	$\dot{V}_u$ (m^3/s)	$\dot{V}_d$ (m^3/s)	$\dot{V}_m$ (dm^3/s)	$\dot{m}_u$ (kg/s)	$(\dot{m}_d + \dot{m}_\ell)$ (kg/s)
100	0.0223	0.0220	22.15	0.0251	0.0245
300	0.0297	0.0288	29.25	0.0327	0.0322
600	0.0378	0.0382	38.00	0.0410	0.0415
900	0.0490	0.0512	50.10	0.0515	0.0543

Table 3: Volume and mass flow rates in the upflow and downflow (including leakage), at throat area for various heat input rates.

CORRESPONDING WALL	100 (W)	300 (W)	600 (W)	900 (W)
AC	15.6	39.9	67.4	95.6
CD	4.7	17.2	41.5	45.5
DE	3.0	7.3	18.2	23.1
EF	6.3	29.4	58.7	83.0
FG	3.7	9.6	18.1	26.7
GH	2.5	6.1	12.1	18.7
HI+IA	5.5	16.8	33.5	42.8
SIDE WALLS	58.6	164.0	350.4	583.9
LEAKAGE	0.6	1.1	1.8	3.1
TOTAL LOSS	100.5	291.4	601.7	922.4

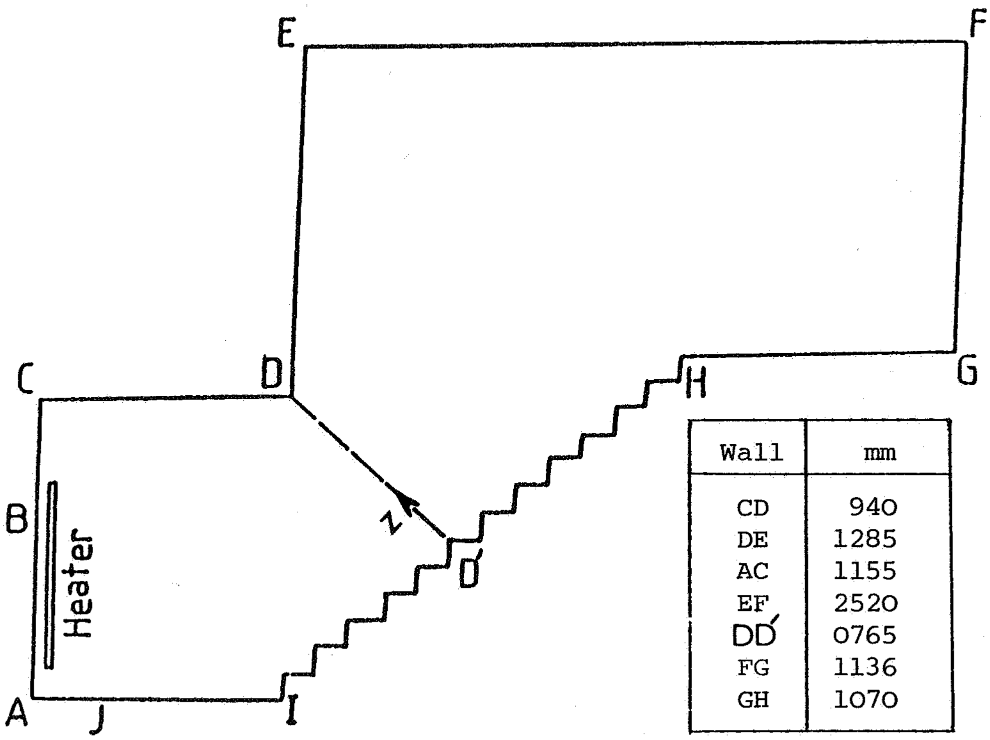
Table 4: Rate of heat loss from the stairwell at various heat input rates (see Figure 1).

CORRESPONDING WALL	300 W	600 W
AC	46.2	85.0
CD	33.6	66.2
DE	15.2	38.9
EF	76.6	178.7
FG	21.3	42.2
GH	14.1	38.4
HI	42.8	67.1
IA	37.0	67.5
LEAKAGE	1.6	2.5
TOTAL LOSS	288.4	586.5

Table 5: Rate of heat loss from the stairwell, with insulated side walls (see Figure 1).

$\dot{Q}$ (W)	T_{av} (^{o}C)	$\dot{V}_m$ (dm^3/s)	1000xFr	1000xSt	Re	Gr	DT (C deg)
100	28.1	22.15	13.85	0.1765	2060	9.733×10^{7}	1.7
300	31.8	29.25	18.33	0.5287	2650	2.179×10^{8}	4.1
600	38.9	38.00	23.81	1.0567	3300	2.945×10^{8}	6.1
900	41.9	50.10	31.31	1.5845	4400	3.819×10^{8}	8.6

Table 6: Basic performance characteristics of the closed-stairwell geometry.



Wall	mm
CD	940
DE	1285
AC	1155
EF	2520
DD'	0765
FG	1136
GH	1070

Figure 1. Schematic diagram of the half-scale stairwell model.

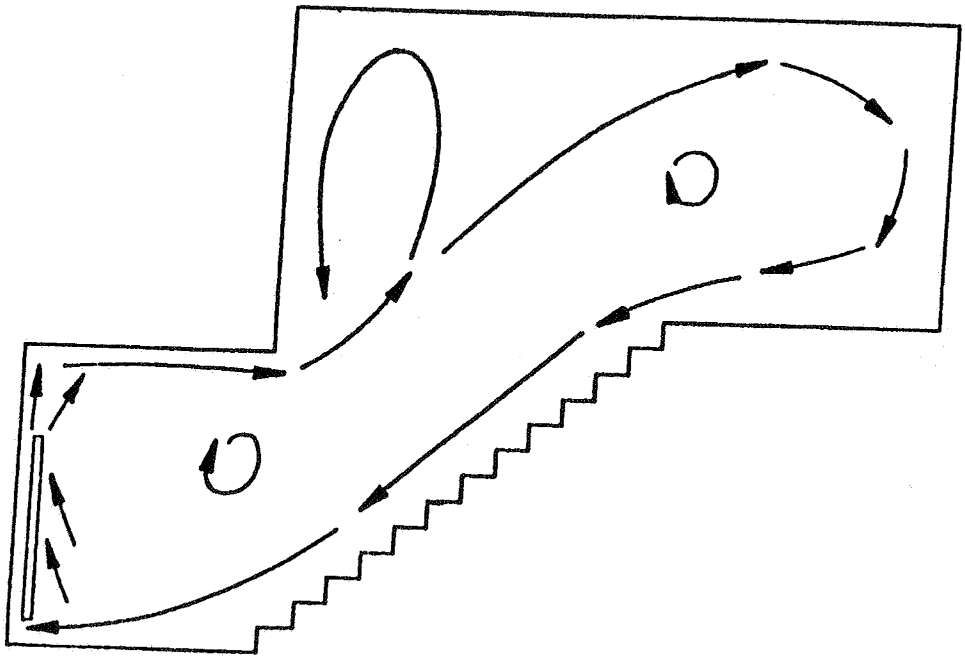


Figure 2. A two-dimensional view of the flow pattern in the stairwell.

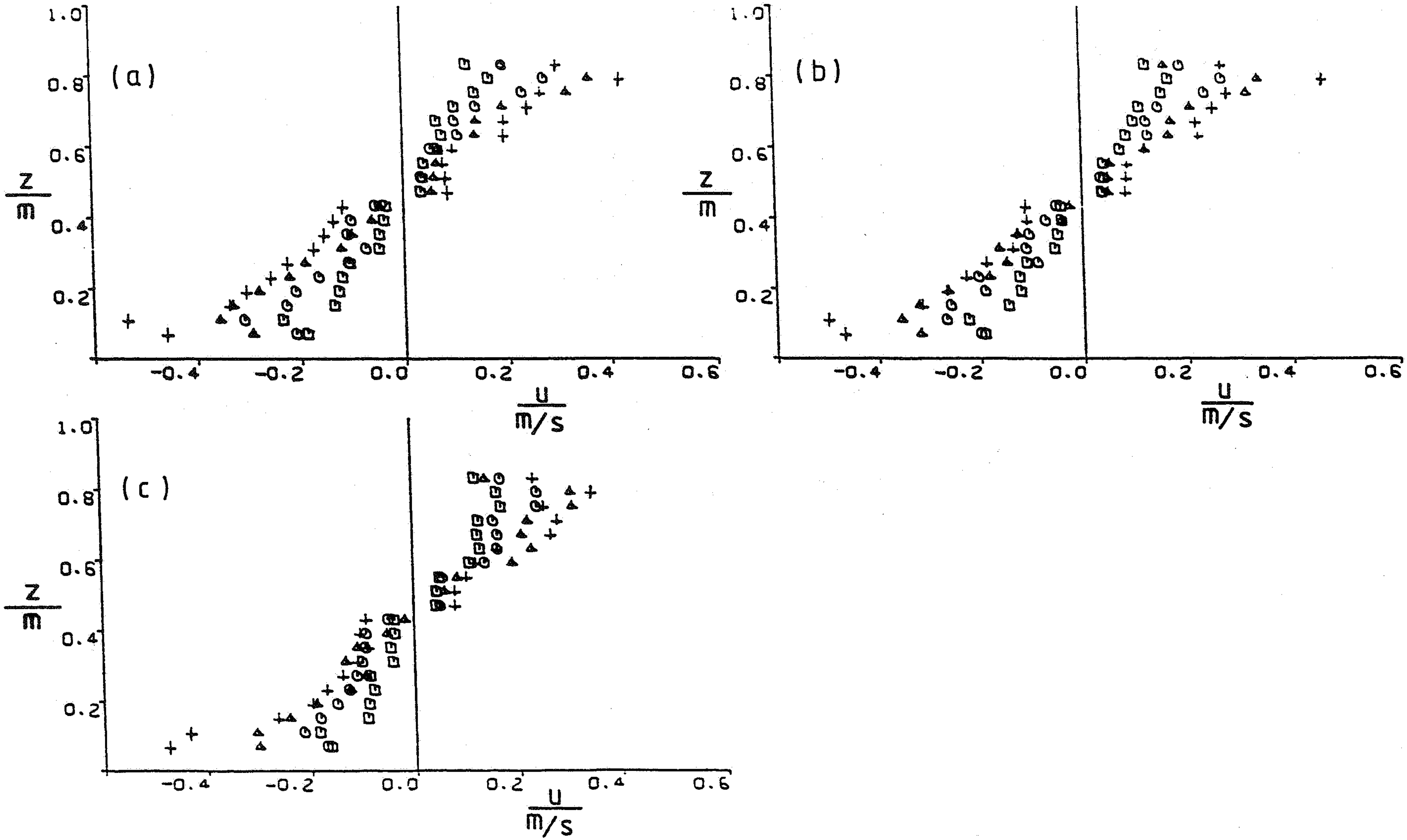


Figure 3 Velocity profiles at various distances from the side wall for various heat input rates.
(a) w/2, (b) w/s, (c) w/6
□ 100 W, ⊙ 300 W, Δ 600 W, + 900 W.

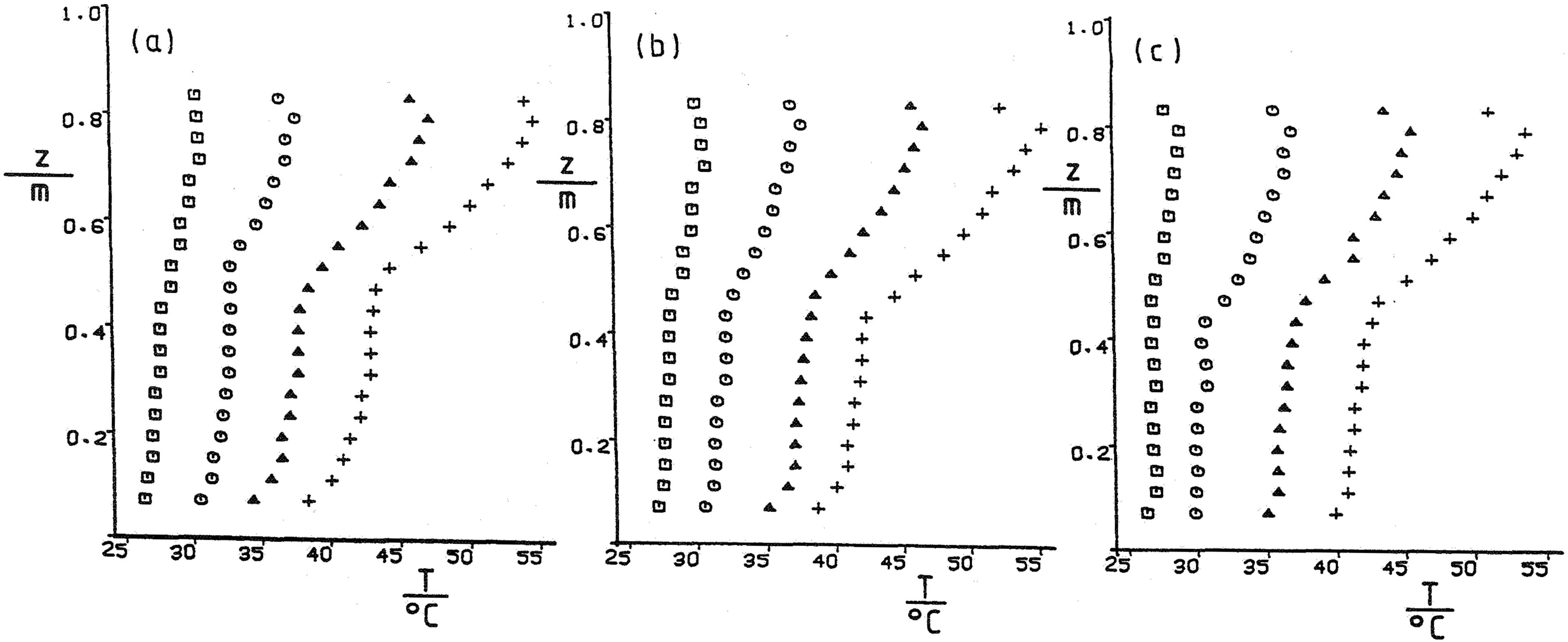


Figure 4 Temperature distributions at various distances from the side wall, for various heat input rates.
(a) w/2, (b) w/3, (c) w/6
□ 100 W, O 300 W, Δ 600 W, + 900 W.

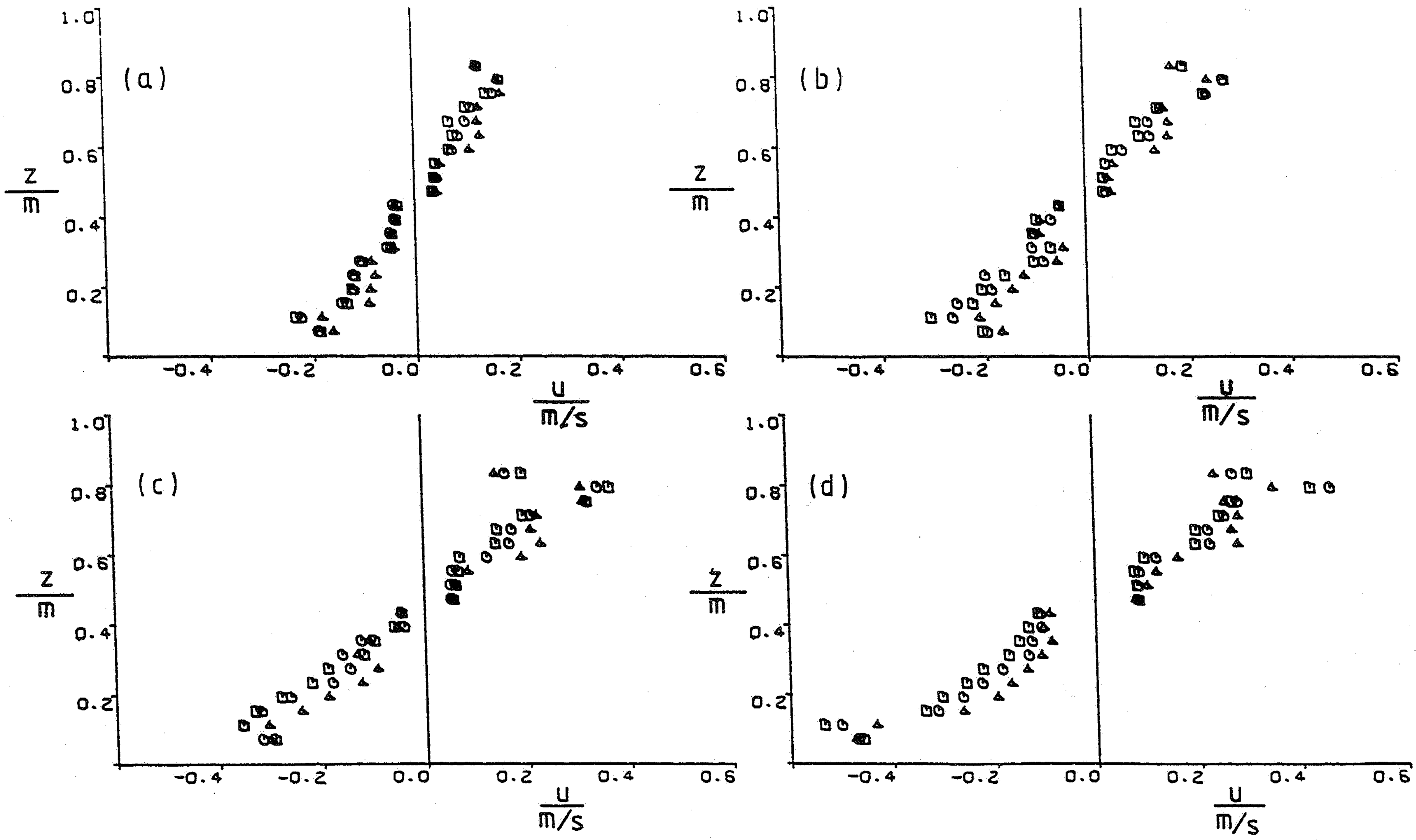


Figure 5. Velocity profiles for various heat input rates, at various distances from the side wall.
(a) 100 W, (b) 300 W, (c) 600 W, (d) 900 W.
○ w/2, □ w/3, Δ w/6.

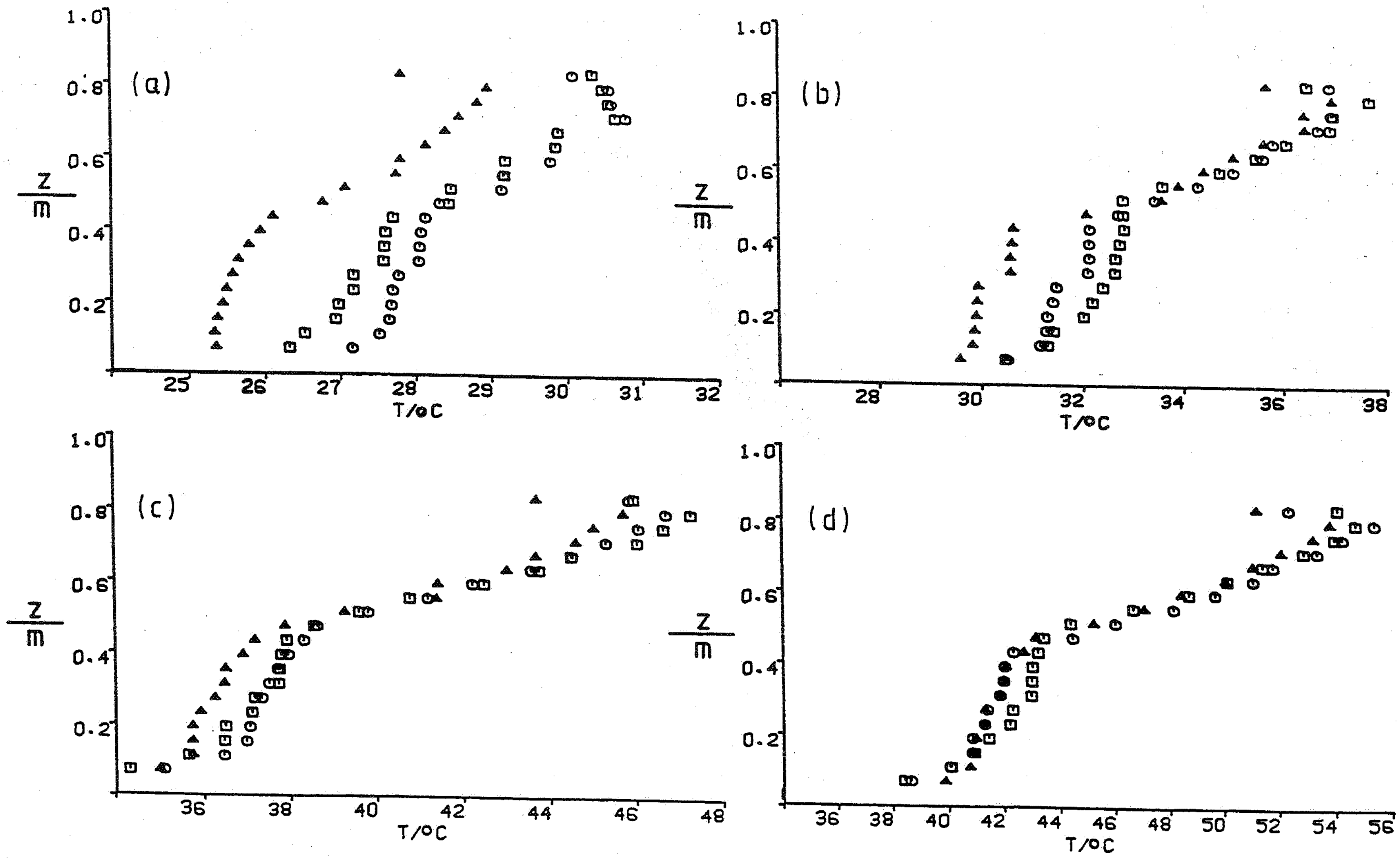


Figure 6 Temperature distributions for various heat input rates, at various distances from the side wall.
(a) 100 W, (b) 300 W, (c) 600 W, (d) 900 W.
O w/2, □ w/3, Δ w/6.

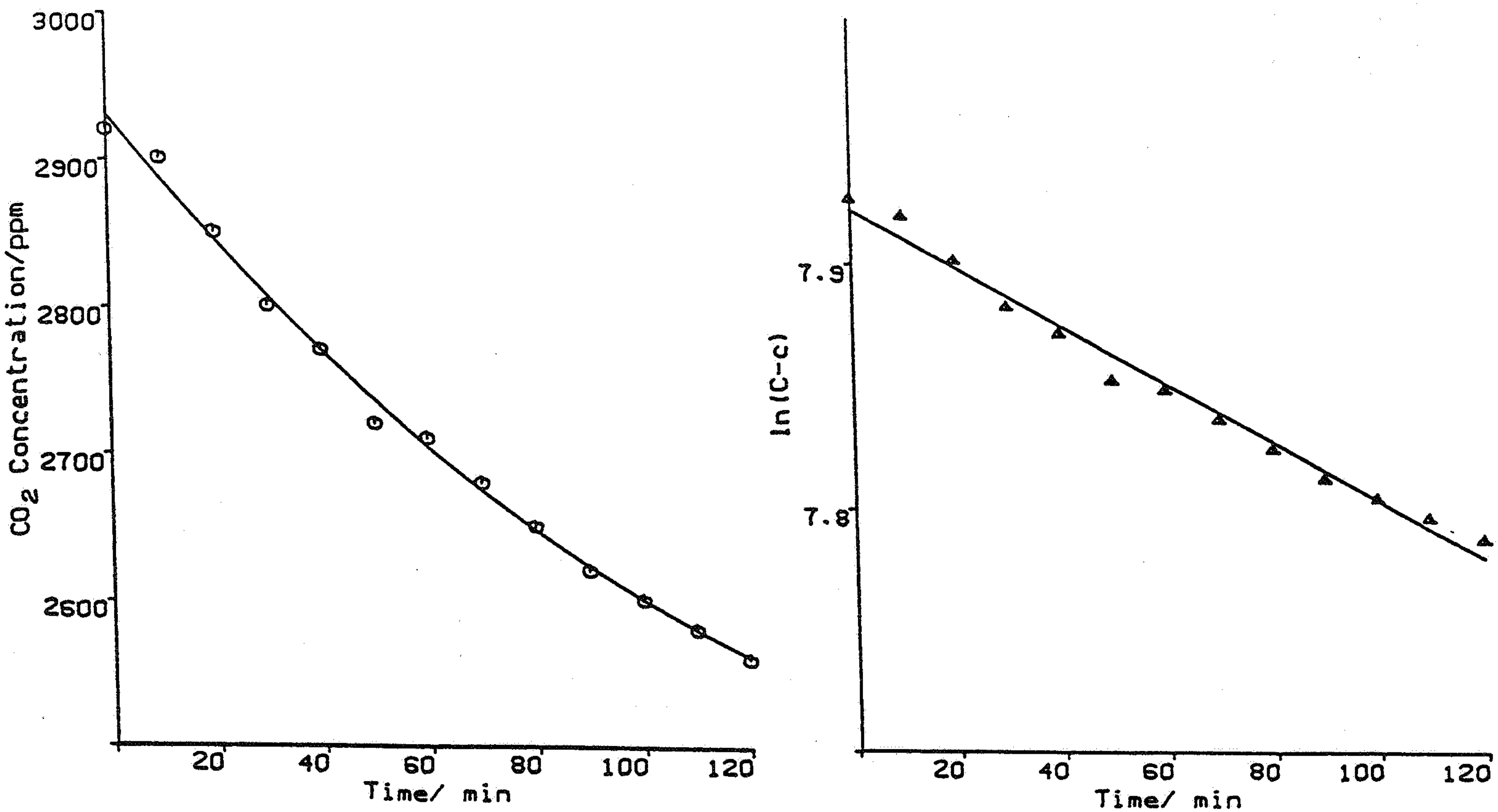


Figure 7. Typical curve of the CO_2 tracer gas concentration against time. (Heat input rate = 100 W).

PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

10th AIVC Conference, Dipoli, Finland
25-28 September, 1989

Poster 13

THE SIMULATION MODEL OF INDUSTRIAL CONDITIONING SYSTEMS

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The Simulation Model of Industrial Air Conditioning Systems

SYNOPSIS

This paper describes a simulation program which was developed for the modelling of air-conditioning systems and conditioned spaces in industrial buildings. The program can be used for a design of systems for new buildings and for analysis of existing ones. By viewing the building as a dynamic entity, it is possible to investigate how thermal capacity of the building elements acts on both the conditioned space and the performance of the air-conditioning system.

The program simulates three important aspects of a building. Simulation of the central air-conditioning system determines enthalpy, temperature and humidity of the air at different points of the system. System performance, capacity requirements and energy consumption under various circumstances are also determined. Simulation of the conditioned space shows its temperature and humidity, when different central air-conditioning systems, airflows and supply-air temperatures are being used. Simulation of the control system makes it possible to investigate the effect of different control methods and devices on the performance of the components of the central air-conditioning system and on the conditions within the conditioned space.

Conditions in the simulated space, capacity requirements of the central air-conditioning system and the air mixture conditions regarding fresh and recirculated air have been compared with measured values. The measurements have shown that simulation gives a realistic indication of how the simulated aspects behave under the given circumstances.

The temperatures of the simulated space have also been compared with the values calculated by the Ventac-program. The average difference was 0,2 ^{o}C.

List of symbols

A_i = area of the windows [m^2]
A_s = area of the walls [m^2]
c_w = heat capacity of the wall [kJ/kgK]
DH = output
e_n = error signal
h_s = heat transfer coefficient of the walls [W/m^2K]
i_s = enthalpy of the room air [kJ/kg]
i_{sp} = enthalpy of the supply air [kg/kg]
i_u = enthalpy of the outdoor air [kg/kgK]
K_p = propotional gain factor
K_i = integral gain factor
$\dot{m}_h$ = vapour from the open containers [kg/s]
$\dot{m}_{hk}$ = vapour from the machines [kg/s]
$\dot{m}_{ih}$ = moisture from the human bodies [kg/s]
$\dot{m}_{iv}$ = mass flow of the supply air [kg/s]
m_s = mass of the room air [kg]
$\dot{m}_{sp}$ = mass flow of the supply air [kg/s]
$\dot{m}_v$ = mass flow of the infiltration air [kg/s]
m_w = mass of the wall [kg]
t = time
T_s = indoor temperature [^oC]
T_u = outdoor temperature [^oC]
T_w = temperature of the walls [^oC]
U_i = thermal transmittance coeff. of the windows [W/m^2K]
U^* = thermal transmittance coefficient of the walls without the effect of the inner heat transfer coefficient [W/m^2K]
x_s = humidity of the indoor air [kg/kg]
x_{sp} = humidity of the supply air [kg/kg]
x_u = humidity of the outdoor air [kg/kg]
ϕ_{rad} = radiating heating load [W]
ϕ_s = convective heating (or cooling) load [W]

The Simulation Model of Industrial Air Conditioning Systems

1 Scope of the Model

In the research and design work of industrial air conditioning systems a need for a simulation program became evident. Because suitable and commercially available programs were not found, it was decided to develop such a program. Using now the newly developed IVSIMUL-model it is possible to simulate

- indoor temperature and humidity
- temperature, humidities and enthalpy at different points of the air conditioning system
- heating and cooling effects and energy consumption
- effects of different control systems
- effect of the thermal capacity of the building envelope

The program consists of three different parts which are linked together. One part which is linked to the dynamic room model simulates the behaviour of the air conditioning unit. The control section makes it possible to simulate the operation of the control system.

2 Room model

In many fields of industry it is important that the air-conditioning system maintains the required indoor humidity and temperature. The simulation of the air conditioning system is also based on the knowledge of humidity, temperature and enthalpy at different points of the system. It's obvious, therefore, that a room model must be created in such a way, that humidity is also taken into account.

Enthalpy equation of the room air is written:

$$\frac{di_s}{dt} m_s = \phi_s + \dot{m}_{sp} (i_{sp} - i_s) + h_s A_s (T_w - T_s) + U_i A_i (T_u - T_s) + \dot{m}_v (i_u - i_s) \qquad (1)$$

The heat balance of the inner surface of the exterior wall is given by the equation (2). The equation is simplified, because it is assumed that the wall consists of only one layer.

$$h_s A_s (T_s - T_w) + \phi_{rad} = U^* A_s (T_w - T_u) + \frac{d T_w}{dt} m_w c_w \qquad (2)$$

The humidity balance equation (3) must be solved together with the equations (1) and (2).

$$\frac{dx_s}{dt} = \dot{m}_{ih} + \dot{m}_{hk} + \dot{m}_h + \dot{m}_v \, (x_v - x_s) + \dot{m}_{iv} \, (x_{sp} - x_s) \quad (3)$$

3 Control simulation

Figures 1 and 2 illustrate an example of a control loop simulation. In both cases the required room temperature is 20 ^{o}C. When a proportional control is used, a difference exists between the actual and required temperature. When a proportional and integral (PI) control is being used, the required room temperature can be achieved. A step by step calculation of the output from the PI-controller is carried out using the formula (4).

$$DH = K_p(e_n - e_{n-1}) + K_i e_n t \qquad (4)$$

It is also possible to simulate the proportional control with hysteresis, which appears in real systems when self acting thermostatic valves are being used.

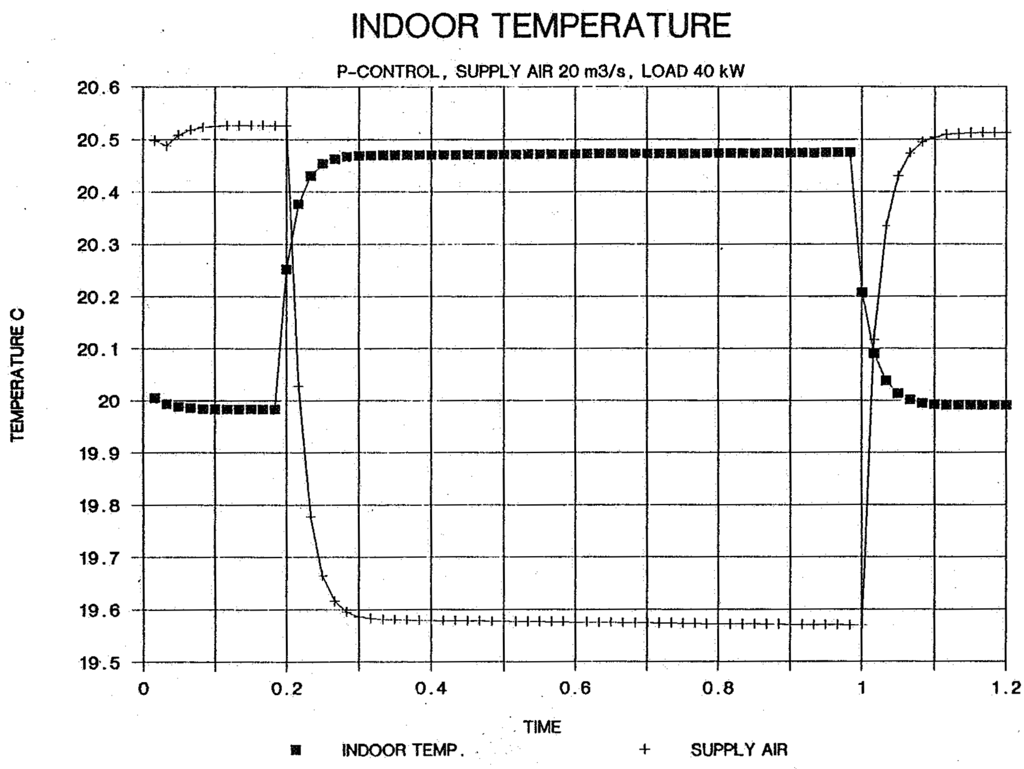

Figure 1. Simulated indoor temperature, P-Control.

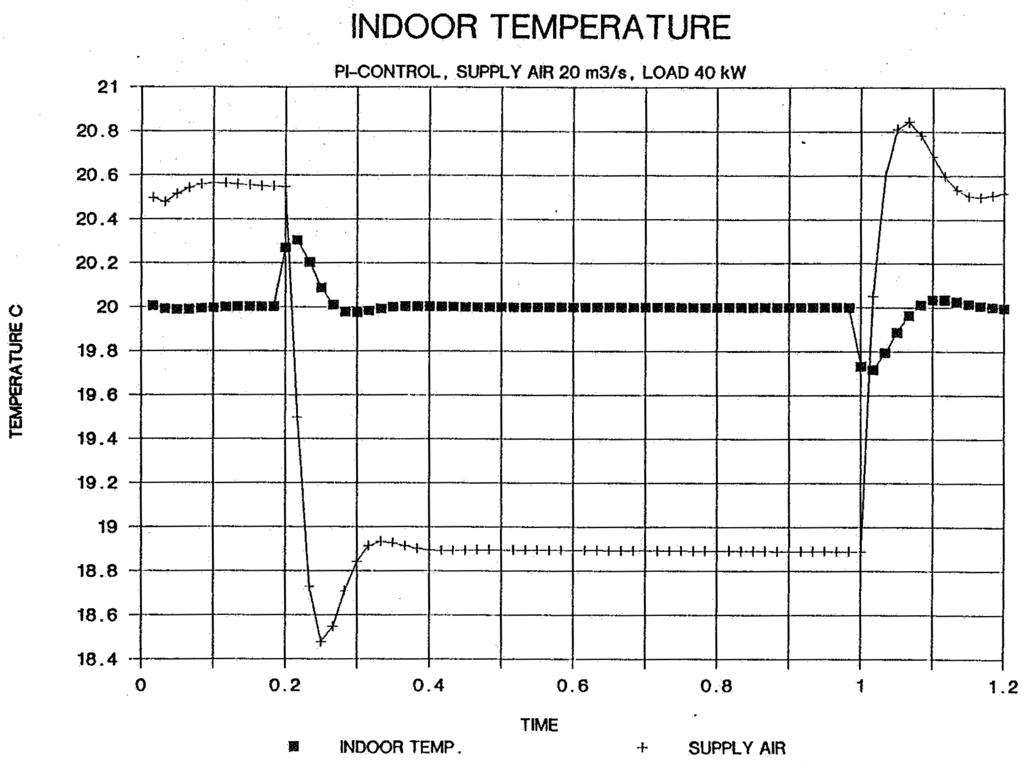

Figure 2. Simulated indoor temperature, PI-Control.

4 Comparison of the simulated and measured values

Simulated values of a printing shop were compared with its measured values. The measured values were

- indoor temperature and humidity in four points
- temperature and humidity of the air in different places of the air conditioning units
- water temperature before and after the heating coils
- energy consumption of the air conditioning
- energy consumption of the radiators
- power demand of the printing machines

The difference between the simulated values and the measured mean daily temperatures was 0,4 °C on the average. The biggest difference was 3,5 °C. An example of one day is illustrated in the figure 3.

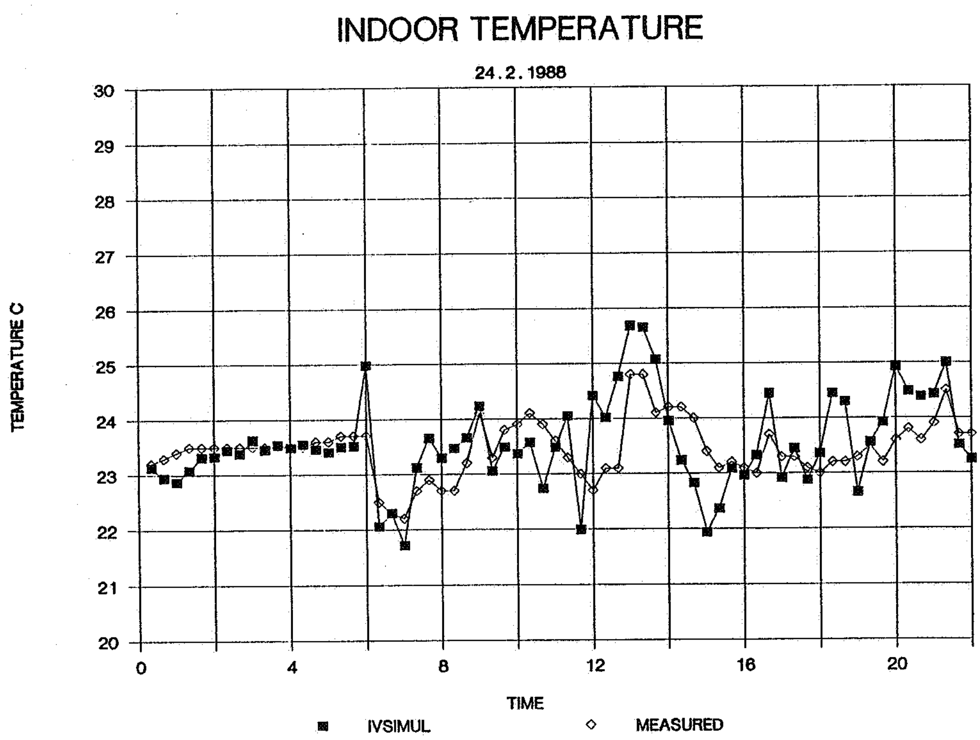

Figure 3. Measured and simulated indoor temperature.

The simulated indoor humidity was somewhat higher than the measured value. The average difference was 0.0018 kg/kg. Reliability of the moisture measurements is not, however, as good as that of the temperature measurements.

The simulated temperature and humidity of the mixed indoor and outdoor air were close to the measured values. The average difference between the simulated and measured effect of the heating coil was 4,9 %. An example of the measured and calculated effect can be seen in the figure 4.

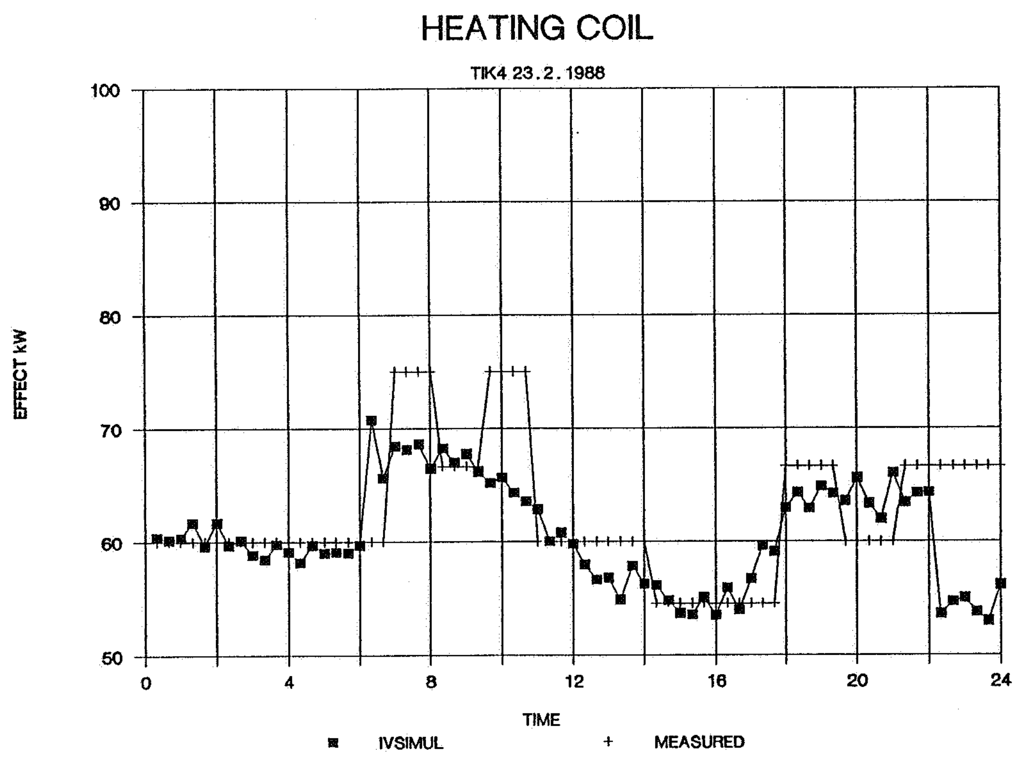

Figure 4. Measured and simulated effect of the heating coil.

5 Comparison of the IVSIMUL and Ventac programs

The IVSIMUL-program was also compared with the Ventac-program. A small hypothetical industrial building was used as a test building . Two cases were used in the comparison. Another was a massive building with minor internal load and the other was a light building with heavy internal load. The floor area of the building was 200 m^2. Window area to the south was 10 m^2 as it was to the north as well. Lighting load was 20 W/m^2. Internal load from machinery was 1 kW in the heavy building and in the light building 5 kW. Supply air flows were 1 l/s m^2 and 10 l/s m^2 accordingly. Working hours were from 7.00 to 16.00 and a presence of 10 workers was assumed. Supply air temperature was 18 ^oC and the air conditioning was operating continuously. There was no moisture or latent heating load.

Calculations showed that the results from the tested programs were near to each other. Differences of the mean daily temperatures of typical days from January to June were from -0,1 to 0,6 ^oC in the heavy industrial building and from -0,3 to 0,4 ^oC in the light industrial building. The standard deviations of the daily differences were from 0,2 to 0,4 ^oC in the light building and from 0,2 to 0,7 ^oC in the heavy building (figures 5 and 6).

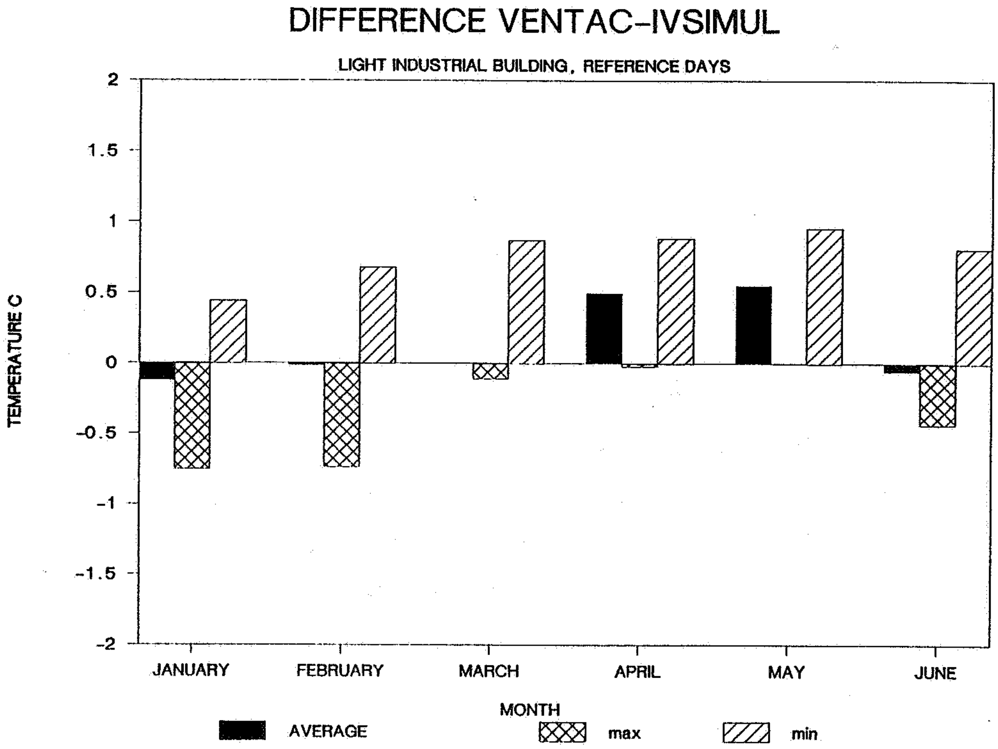

Figure 5. Comparison of the Ventac and IVSIMUL-programs.

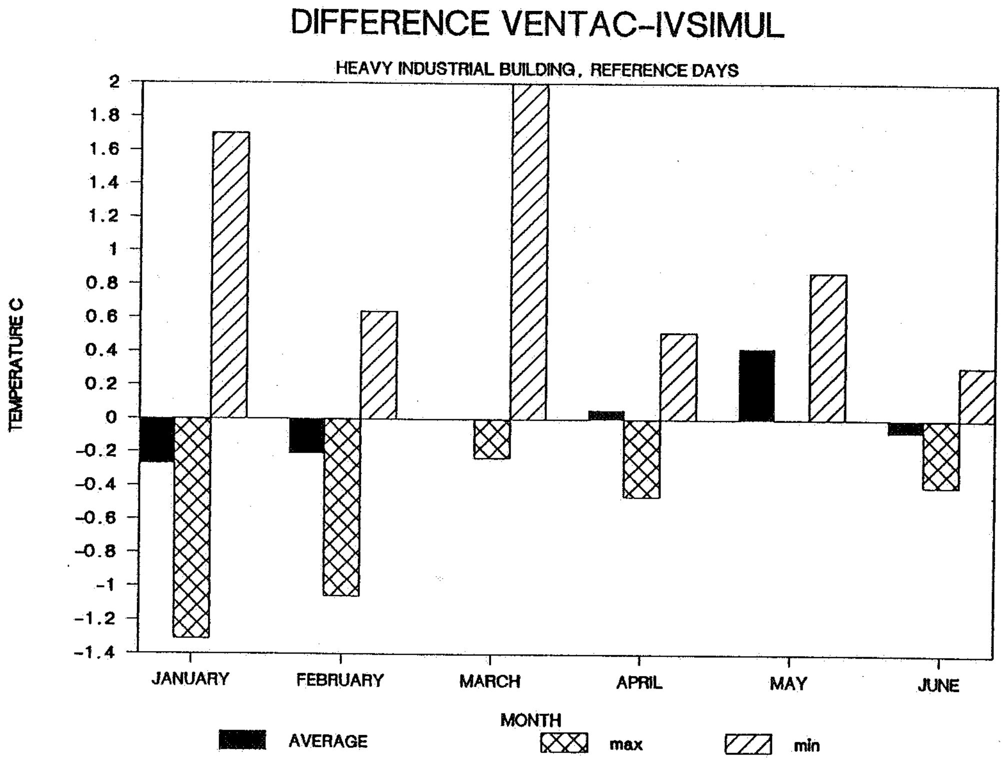

Figure 6. Comparison of the Ventac and IVSIMUL-programs.

6 Conclusions and further development

The IVSIMUL-program seems to give reasonably good results for practical purposes. Several simplifications, however, have been made and they should be born in mind when using the program. The program will be used in the future for analysis of new and existing industrial buildings. More accurate description of the cooling coils will be one of the future enhancements. The control system part of the program could also be extended. The possibility of linking the IVSIMUL thermal simulation model to a concentration model developed by PI-Consulting Ltd is under consideration.

Acnowledgements

The author wishes to thank Finnish Academy and PI-Consulting Ltd which funded the work.

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PROGRESS AND TRENDS IN AIR INFILTRATION AND VENTILATION RESEARCH

10th AIVC Conference, Dipoli, Finland
25-28 September, 1989

Poster 14

VAV - DUCT SYSTEMS - SIMULATING

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1. INTRODUCTION

VAV - air conditioning system makes it possible to control indoor conditions even when the heat loads are changing. But this is possible only when each part of the system works as it is intended to work.
When the air flows varies in a large range, it can cause situations, where pressure loss of some flow dampers are out of their operate range. This is possible especially when the system is large and the velocities are high. This means that the air flow is not correct. Also increasing noise levels may appear.
Because of increasing lack of space, it's impossible to use large ducts to ensure systems function under all conditions. So how we can be sure that the system works ?

2. SIMULATING PROGRAM WITH EASY USER INTERFACE

To solve this problem, we have developed a computer program with which we can easily dimension the ducts, simulate different air flow situations in VAV-systems and easily make pressure- and sound-calculations for each situation. We can also simulate different placings of the pressostat, which controls the fan unit.

2.1. Draw a line scheme of the duct system

The user interface of the program is based on graphics. This means that it is quite easy to input a duct system in to the program. You only need to draw a line scheme of the system. The program can deduce from the scheme most of the needed information (connections, duct lengths, bends etc).

2.2. Select dampers and diffusers from pull-down menues

The dampers, attenuators, diffusers and other units can be selected from product menues of the program. The menu system is build up like a tree and on the bottom there are menues consisting single-loss dampers, constant-flow dampers, VAV-dampers, sound attenuators and various types of diffusers.

2.3. Copy branch

If you want to copy some branch in to another place, you only need to show with the cursor which branch and where to copy.

2.4. Low mistake possibilities

So it doesn't take much time to input even large duct systems. Possibility to make input mistakes is quite low because all mistakes are immediately shown in the picture, which is impossible if the user interface is based on table.
When you want to change informations for some part of the duct system, you only need to point the part with the cursor and then select new values from the menu or give the new values in a small window.

2.5. Making calculations

Various different calculations can be made with the program. First balance the duct-system when the program calculates the total pressure needed, positions of the single-loss dampers and the pressure loss of the other parts.
Then simulate different air flow situations. Simply show the dampers which have different air flows and then give the valid air flow. Also show the place for the pressostat.
In simulated air flow situations you can make the sound calculations. If you want to look at some route, simply show the beginning and the end of the route.
You can also get a partlist of procucts needed to build up the duct system.

2.6. Outputs

In pressure drop calculations (balancing or simulating), there are four possibilities for output. You can i.e look at a route, when the program draws a curve from which you can see how the pressure drops in each part of the route. You can also check the operating points of the dampers.
In sound calculations you can take out a route and check how the sound is generated in each part. You can also make automatic sound calculations, which means that the program calculates the sound levels from the fan to each diffuser and outputs the sound levels for the diffusers.

3. CONCLUSION

A duct system simulating program makes it easy to simulate all possible air flow situations of the current VAV-system. With the program consultants can easily find out possibilly occuring problems in the system and correct them immediately. It is much cheaper the correct the system on a screen than to do it in an existing building.

PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

10th AIVC Conference, Dipoli, Finland
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Poster 15

IDENTIFICATION METHODS FOR MULTIPLE CELL SYSTEMS

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SYNOPSIS

A tracer gas technique for determining volumes and airflow rates in multi-cell systems with a single tracer gas is considered. Tracer gas is injected in all cells simultaneously according to a cetain pattern and the resulting tracer gas concentrations are recorded. We show how the volumes and flow rates can be identified from the measurements using the quadratic programming method. A characteristic of this method is that the unknown model parameters, i.e. the volumes and flows, can be determined subjected to given constraints. Accordingly we can utilize all known information about the flow system and make sure that a found solution is physically compatible with a flow system.

Different formulations of the mass balance relations are discussed and a new method to overcome the difficulties to calculate the derivatives of the tracer gas concentrations is proposed. The properties of the different formulations such as the sensitivity to noise and the significance of the sample period are exemplified by identifications on a simulated three cell system.

LIST OF SYMBOLS

A	the matrix $V^{-1}Q$	n by n
A	matrix of measured data, see (3.4)	mn by n_p
b	vector of measured data, see (3.5)	mn by 1
$c(t)$	tracer gas concentration vector	n by 1
$\dot{c}(t)$	derivatives of c(t)	n by 1
D	the matrix $A(\exp(AT_s)-I)^{-1}$	n by n
$E(x)$	expectation value of x	
G	matrix of constraints	n_g by n_p
g	vector of lower limits in the constraints	n_g by 1
I	identity matrix	
m	number of samples	
m_{int}	number of subintervals of the interval of integration	
n	number of cells in the flow system	
n_c	number of parameters in the LCP problem, see (3.22)	
n_g	number of constraints	
n_p	number of parameters	
$p(t)$	tracer gas injection vector	n by 1
Q	flow matrix	n by n
Q	the matrix A^TA	
q_{ij}	flow to cell i from cell j	
T_{int}	interval of integration	
T_s	sample period	
TC	building time constant (inverse of air exchange rate)	
t	time	
t_k	discrete time	
u	vector of Lagrange multipliers	n_g by 1
V	volume matrix	n by n
v	vector of Lagrange multipliers	n_p by 1
x	model parameter vector, see (3.1)	

1 INTRODUCTION

Following the ideas suggested by Jensen (1986,1987b) we consider an experiment where tracer gas is injected in all cells simultaneously using a certain pattern in time, so that the influence of different tracer gas inputs can be separated. Ideal mixing is assumed in each cell and the flows are assumed to be constant and without time delays.

A large system of equations can be stated based on the mass balance equations, measured tracer gas concentrations and tracer gas injections. As the model is stated in continuous time the system of eqautions is linear in the model parameters, i.e. the unknown flows and volumes which are to be identified. There are also linear constraints on the model parameters which must be fulfilled if the solution should be compatible with a flow system. Important constraints are that all flows and volumes must be non-negative. Because of these constraints the commonly used least square identification method is not so useful in this case.

So far the linear programming (LP) method has been used, see Jensen (1987b). This method permits constraints but, however, as the number of measurements (m) increases the demand for computer time and storage grows very fast (approx. as m^3 and m^2, respectively) and soon this may be a serious limitation. This is certainly true on a PC based system.

In this paper some alternative methods to state and solve the identification problem are proposed. Especially we show that quadratic programming (QP) has some attractive properties. The QP method has been used on a number of simulated test experiments with a three cell system and with different levels of measurement noise. It has proved to work well in these tests.

Like the LP method, the QP method makes use of the constraints of the model parameters to guarantee that the solution is a flow system. But, and this is the main advantage, the dimension of the QP problem does not depend on the number of measurements, but only on the number of model parameters and constraints. In spite of a slightly more complicated solution, the savings in computer time and storage may become most considerable when the number of measurements increases.

2 MODELS FOR FLOW SYSTEMS

In this chapter we shall derive some different variations of the mass balance relations which can be stated from measurement of a multi-cell flow system. These models will be used in the next chapter when we show how to compute the flows and volumes from given measurements.

Section 2.1 summarizes some basic facts about the mass balance equations for a flow system. This is a well-known model, see e.g. Sinden (1978) for a more detailed treatment. A difficulty with the mass balance is that it contains the time derivatives of the tracer gas concentrations. The following sections deal with different ways to overcome this problem. Since it is not suitable to use an ordinary discrete time model, we show how to calculate the derivatives from measurements both approximately and exactly with an iterative model, see Section 2.3. Another way to avoid the derivatives is to integrate the mass balance equations. In this case, however, a new problem is how to compute the integral of the tracer gas concentration. Again, this can be done approximately as well as exactly by a model based method. This is described in Section 2.4. Finally, in Section 2.5 we make a summing up of the models.

2.1 The mass balance of a flow system

Consider a flow system with n cells. Each cell can be connected to the other n-1 cells and to the exterior or the outside by flows through one-way passages, see Figure 2.1

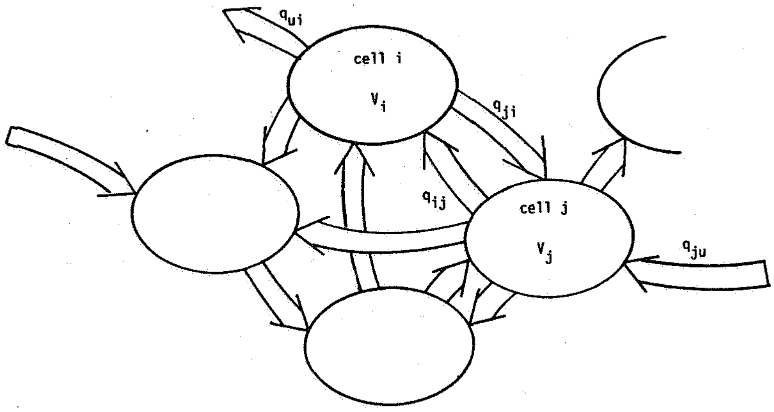

Figure 2.1 An example of a flow system

The parameters of the system are the volumes, denoted v_i, the flows to and from the outside, denoted q_{ui} and q_{iu} respectively and the interflows between the cells, where the flow to cell i from cell j is denoted q_{ij}. The number of parameters is n volumes, n inflows, n outflows and n(n-1) interflows or altogether n(n+2) parameters. Some of the flows may, of course, be equal to zero.

As the sum of flows into each cell must be equal to the sum of flows out from the cell, it follows that n flows are dependent of the other flows. Thus a general flow system with n cells has n(n+1) independent parameters.

Provided that all flows are constant and without time delays, and that the mixing in each cell is perfect the following mass balance equation holds for each cell i, i=1,n

$$v_i\dot{c}_i(t) = \sum_{j\neq i}^{n} q_{ij}c_j(t)-q_{ti}c_i(t)+p_i(t), \quad i=1,n \tag{2.1}$$

where $c_i(t)$ denotes the tracer gas concentration in cell i at time t, $\dot{c}_i(t)$ denotes the time derivative of $c_i(t)$ and $p_i(t)$ denotes the tracer gas injection in cell i. q_{ti} denotes the total outflow from cell i.

The total outflow for each cell i can be written as

$$q_{ti}= \sum_{j\neq i}^{n} q_{ji}+q_{ui}, \quad i=1,n \tag{2.2}$$

and the equally large total inflow can be written as

$$q_{it}= \sum_{j\neq i}^{n} q_{ij}+q_{iu}, \quad i=1,n \tag{2.3}$$

By using matrix notation, all the n mass balances in (2.1) can also be expressed in a compact form

$$V \dot{c}(t) = Q\, c(t) + p(t) \tag{2.4}$$

Now, $\dot{c}(t)$, $c(t)$ and $p(t)$ are n by 1 vectors with the elements $\dot{c}_i(t)$, $c_i(t)$ and $p_i(t)$, $i=1,n$. V is an n by n diagonal volume matrix with entries equal to the volumes v_i. Q is an n by n flow matrix. The diagonal elements Q_{ii} are equal to $-q_{ti}$ and the off-diagonal elements Q_{ij}, $i \neq j$ are the interflows q_{ij}.

However, if a given model in the form (2.4) should be compatible with a flow system the matrices V and Q must have the following properties, (2.5)-(2.9). All the volumes and flows must be non-negative, that is

$$V_i \geq 0, \quad i=1,n \tag{2.5}$$

$$Q_{ij} \geq 0, \quad i \neq j \tag{2.6}$$

The total flows to/from each cell are positive. Consequently, the diagonal elements in the flow matrix must be negative

$$Q_{ij} < 0, \quad i=j \tag{2.7}$$

From the equations (2.2) it follows that the sums of the rows of the flow matrix equal the negative value of the inflows from the outside. As all inflows must be non-negative it follows that

$$\sum_j Q_{ij} = -q_{iu} \leq 0, \quad i=1,n \tag{2.8}$$

In the same way the equations (2.3) state that the sums of the columns of the flow matrix equal the negative value of the outflows to the outside. Hence, also the sums of columns must be non-positive

$$\sum_i Q_{ij} = -q_{uj} \leq 0, \quad j=1,n \tag{2.9}$$

These five constraints on the volume and flow matrices will play an important role when we shall choose identification method.

2.2 The mass balance equations in discrete time

The most convenient way to work with a system of linear differential equations, as our mass balances (2.4), is to transform it to a system of difference equations. The difference equations only describe the variables at discrete sample instants. But provided that the inputs (i.e. the tracer gas injections) are constant during the sample periods, this is an exact description of the continuous time system at the sample instants.

It is very easy to make simulations with the difference equations, and the methods to identify discrete time models are very well established, see Jung (1987).

To obtain the mass balance in discrete time we shall first rewrite (2.4) in a more common form

$$\dot{c}(t) = A\, c(t) + B\, p(t) \tag{2.10}$$

where A and B are the n by n matrices

$$A = V^{-1}Q \tag{2.11}$$

$$B = V^{-1} \tag{2.12}$$

The corresponding equation in discrete time is given by (see e.g. Åström (1985) for complete details or just integrate (2.10))

$$c(t_k+T_s) = Fc(t_k) + Gp(t_k) \tag{2.13}$$

where $c(t_k)$ och $p(t_k)$ are the sampled values of $c(t)$ and $p(t)$ at the discrete sample instants $t_k=t_o+kT_s$, $k=\ldots,-1,0,1,\ldots$. T_s is the sample period and F and G are n by n matrices given by

$$F=\exp(AT_s) \tag{2.14}$$

$$G=(\exp(AT_s)-I)A^{-1}B=(\exp(AT_s)-I)Q^{-1} \tag{2.15}$$

In our case, however, the discrete time model is not an appropriate model to identify. The reason is that the constraints on the flow matrix Q (2.8) and (2.9) will not be linear after transformation to discrete time, this is shown in detail in Jensen (1987a). As we cannot make use of all the constraints we can neither guarantee that an identified model will be compatible with a flow system. Of course, it may be compatible this is easy to check afterwards but generally, we will not find a flow system. Moreover, even if the identified model is a flow system, it is hard to compute the diagonal matrix V from (2.14) and (2.15) as F and G are filled matrices.

Since we cannot use the discrete time model we have to work with the original time continuous model (2.4). The next two sections will treat how to deal with the time derivative in this model.

Nevertheless, we will take advantage of the discrete time model in the following sections when computing the derivative and integral of the tracer gas concentration vector. It will also be used to simulate the test example in chapter 4.

2.3 Computing the time derivatives of the tracer gas concentration vector

The simplest but certainly not the most accurate method to estimate the time derivative is to use a difference approximation. There are several possibilities: forward, backward, forward/backward difference and so on. The most useful method, see Jensen (1986), turns out to be the ordinary forward difference

$$\dot{c}(t) \approx (c(t_k+T_s) - c(t_k))/T_s \tag{2.16}$$

If the sampled signal $c(t_k)$ is measured without measurement noise then the approximation becomes better the smaller the sample period is chosen. But (2.16) is quite sensitive to noise and the error caused by noise increases as the sample period is reduced. The final choice of sampling period becomes a compromise between error from the approximation and error from the noise.

As already mentioned, the derivative can also be computed exactly from the discrete time model. To derive a formula for this, start from equation (2.13) and replace the matrices F and G with the expressions (2.14) and (2.15), respectively. After subtracting both sides by $c(t_k)$ we get

$$c(t_k+T_s)-c(t_k) = (\exp(AT_s)-I)c(t_k)+(\exp(AT_s)-I))Q^{-1}p(t_k) \qquad (2.17)$$

Multiply this equation from the left by D, where D denotes the n by n matrix

$$D = A(\exp(AT_s)-I)^{-1} \qquad (2.18)$$

After reduction we obtain

$$D(c(t_k+T_s)-c(t_k)) = A\ c(t_k) + B\ p(t_k) \qquad (2.19)$$

Comparing this equation with (2.10) we recognize that the left side in (2.19) equals $\dot{c}(t)$, i.e.

$$\dot{c}(t_k) = D(c(t_k+T_s) - c(t_k)) \qquad (2.20)$$

Notice that this formula is not an approximation but holds exactly the same assumptions as provided for the discrete time model. A comment is that theoretically the tracer gas injection p(t) thus must be constant between the sampling instants. In practice, however, a more convenient way is to inject the tracer gas by pulses. But these pulses must be sufficiently small and equally spread over the sampling period. The meaning of 'sufficiently small' in this context is possible to calculate, given a discrete time model.

A difficulty with (2.20) is that D depends on A, see (2.18). Since we want to identify the flow and volume matrices, $A(=V^{-1}Q)$ is unknown and so we cannot calculate the derivatives.

A solution of this problem is to start with $D=IT_s^{-1}$ as in the forward difference approximation and identify a preliminary model. With this model we can calculate a new matrix D and a new better approximation of $\dot{c}(t_k)$. Then we can identify a new model and the process proceeds iteratively. The method will converge in a few steps if the sampling period is not too long compared with the time constants of the system, see examples in chapter 4.

The best numerical method to calculate D from (2.18) is to use the following Taylor series expansion

$$T_sD=(AT_s)(\exp(AT_s)-I)^{-1}=I-(AT_s)/2+bn_1(AT_s)^2/2!-bn_2(AT_s)^4/4!$$
$$+bn_3(AT_s)^6/6!-\dots \qquad (2.21)$$

where bn_i, i=1,2,... denotes the Bernoulli numbers.

From (2.21) it is clear that the matrix D is approaching IT_s^{-1} as the sample period T_s decreases. As a limit, the formula (2.20) will turn into the forward difference (2.16) when T_s goes to zero.

The matrix D has the properties that the diagonal elements are a bit greater than 1 while the off-diagonal elements are rather small and negative. This results in a somewhat higher sensitivity to noise in the model based formula than in the forward difference formula. In both formulas the noise sensitivity increases as T_s decreases.

2.4 The integrated mass balance

A common way to suppress noise is to integrate. Integrating the mass balance (2.4) from the time t_k to t_k+T_{int} yields

$$V(c(t_k+T_{int}) - c(t_k))=Q\ C(t_k)+P(t_k) \tag{2.22}$$

where $C(t_k)$ and $P(t_k)$ denote the integrals of $c(t_k)$ and $p(t_k)$ respectively

$$C(t_k)=\int_0^{T_{int}} c(t_k+s)ds \tag{2.23}$$

$$P(t_k)=\int_0^{T_{int}} p(t_k+s)ds=T_{int}p(t_k) \tag{2.24}$$

The last equality in (2.24) makes use of the assumption that the tracer gas injection is constant during the interval of integration, T_{int}.

The integral $C(t_k)$ can be calculated approximately with the trapezoidal rule. Suppose that the interval of integration is subdivided in $m_{int}=T_{int}/T_s \geq 1$ equal subintervals, then the integral is approximated by

$$C(t_k)=T_s \sum_{i=0}^{m_{int}} f_i c(t_k+iT_s) \tag{2.25}$$

where the coefficients $f_0=f_{m_{int}}=1/2$ and the others $f_i=1$, $i=1,n-1$.

In this formula it is always advantageous to choose a short sampling period in order to use many measurements and reduce the noise corruption.

The integral $C(t_k)$ can also be determined exactly and this can be done in several different ways. Since the discrete time model is valid for every choice of the sample period as long as the input is constant, we can exchange T_s in (2.13) for the time argument s, where $0\leq s\leq T_{int}$. This gives

$$c(t_k+s) = \exp(As)c(t_k) + (\exp(As)-I)A^{-1}B\ p(t_k) \tag{2.26}$$

and $C(t_k)$ can be computed by integration of this expression as stated in (2.23). Let us call this method A.

In this approach we use the values of the constant tracer gas injections and the values of the tracer gas concentrations in the beginning of the interval of integration in order to determine the tracer gas concentrations during the whole interval.

An alternate approach is to use the concentrations at the end of the interval instead of the values at the beginning of the interval. An equation expressing $c(t_k+s)$ as a function of $p(t_k)$ and $c(t_k+T_{int})$ can be derived in the following way. Multiply (2.13) from the left by $\exp(-AT_{int})$ and use the equation to express $c(t_k)$

$$c(t_k)=\exp(-AT_{int})c(t\ +T_{int})-\exp(-AT_{int})(\exp(AT_{int})-I)A^{-1}\ Bp(t_k) \tag{2.27}$$

Substitute $c(t_k)$ in (2.26) with (2.27). After reduction we obtain

$$c(t_k+s)=\exp(A(s-T_{int}))c(t\ +T_{int})+(\exp(A(s-T_{int}))-I)A^{-1}Bp(t_k) \tag{2.28}$$

Now the integral $C(t_k)$ can be computed by integration of (2.28) as well. This will be called method B.

With respect to noise reduction, however, it seems sensible to use the concentration at both the beginning and the end of the interval. This can be done in two ways. One is to integrate the mean value of $c(t_k+s)$ expresed by (2.26) and (2.28). Let us call this method C. The other way is to eliminate $p(t_k)$ from (2.26) by (2.28) and express $c(t_k+s)$ from the resulting equation, i.e., method D.

Now we have four possibilities A-D to calculate the integral $C(t_k)$ and all of them hold exactly, provided that $p(t_k)$ is constant during the interval of integration and, of course, that there are no measurement errors. In the presence of noise however, the four methods will behave quite different. (For example we can notice that (2.28) is an extrapolation backwards of a stable system, and this is an unstable process with respect to disturbances in the initial value. On the other hand, (2.26) is an extrapolation forward of a stable system and does not cause any problem.

In Hedin (1989) the noise sensibility of the four methods are calculated in a simple case. It is shown that method C, i.e., integrating the mean of $c(t_k+s)$ from (2.26) and (2.28), gives the best result provided the interval of integration is reasonably small compared to the time constants of the system.

Leaving methods A, B and D, we can perform the integration with method C. After simple calculations this gives

$$C(t_k)=\int_0^{T_{int}} c(t_k+s)ds=R\ c(t_k)+S\ c(t_k+T_{int})+T\ B\ p(t_k) \tag{2.29}$$

where R, S and T are n by n matrices given by

$$R=(\exp(AT_{int})-I)A^{-1}/2 \tag{2.30}$$

$$S=(I-\exp(-AT_{int}))A^{-1}/2 \tag{2.31}$$

$$T=((\exp(AT_{int})-\exp(-AT_{int})A^{-1}-2IT_{int})A^{-1}/2 \tag{2.32}$$

For the numerical calculations of R, S and T we can use the following Taylor series expansions

$$R=(IT_{int}+AT_{int}{}^2/2!+A^2T_{int}{}^3/3!+\ldots)/2 \tag{2.33}$$

$$S=(IT_{int}-AT_{int}{}^2/2!+A^2T_{int}{}^3/3!-\ldots)/2 \tag{2.34}$$

$$T=AT_{int}{}^3/3!+A^3T_{int}{}^5/5!+\ldots \tag{2.35}$$

The extension of (2.29) to the case with several subdivisions ($m_{int}>1$) of the interval of integration is easy to derive. Just exchange T_{int} for T_s in (2.29)-(2.35) and calculate the integral as the sum of integrals for each subdivision

$$C(t_k)=R \sum_{k=0}^{m_{int}-1} c(t_k)+S \sum_{k=0}^{m_{int}-1} c(t_{k+1})+T\ B\ p(t_k) \qquad (2.36)$$

Notice from the equations (2.33)-(2.35) that when the number of samples in each interval of integration is increasing, then the matrices R and S will approach $IT_s/2$ while T will approach $AT_s^3/3!$ (or almost zero, as T_s is a small number). Accordingly, the equations (2.29) and (2.36) will approach the simple trapezoidal rule as T_s becomes small.

Like the model based estimation of the derivative this model based estimation of the integral will be an iterative method as we do not know the matrix A in advance. As a first estimation of the integral it is natural to use the trapezoidal rule.

2.5 Summing up the models

In this chapter we have been engagaed with the problem of finding the relations between tracer gas injections and the resulting tracer gas concentrations in a multicell flow system.

The fundamental model is the mass balance equations (2.4)

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

To express the time derivative we have used two formulations of the mass balance equations

Formulation 1 forward difference of $\dot{c}(t_k)$

$$V(c(t_k+T_s)-c(t_k))/T_s=Q\ c(t_k)+p(t_k) \qquad (2.37)$$

Formulation 2 model based estimation of $\dot{c}(t_k)$

$$V\ D(c(t_k+T_s)-c(t_k))=Q\ c(t_k)+p(t_k) \qquad (2.38)$$

An alternative, where we avoid the time derivative, is to integrate the mass balance equation. To obtain the integral $C(t_k)$ of $c(t_k)$ we have also used two formulations

Formulation 3 integrated mass balance with the trapezoidal rule

$$V(c(t_k+T_{int})-c(t_k))=Q(c(t_k)+c(t_k+T_{int}))/2+T_{int}p(t_k) \qquad (2.39)$$

Formulation 4 integrated mass balance with model based integration of $c(t_k)$

$$V(c(t_k+T_{int})-c(t_k))=Q(R\ c(t_k)+S\ c(t_k+T_{int})+TBp(t_k))+T_{int}p(t_k) \qquad (2.40)$$

If we have m samples with measurements of $p(t_k)$ and $c(t_k)$, k=1,m, then the different formulations can be stated m-1 times. The measurements are normally recorded at a constant sampling rate, but this is no demand.

3 IDENTIFICATION OF MULTI-CELL FLOW SYSTEMS

In this chapter we address the identification problem: Given measured values of the tracer gas injections, $p(t_k)$, and the tracer gas concentrations $c(t_k)$ in a multi-cell flow system, how could the volumes and flow rates be determined?

The chapter is organized as follows. In Section 3.1 we show how to state the measured data as an overdetermined system of equations.

Next, in Section 3.2, we show how this system of equations can be formulated as an quadratic programming problem.

Finally, this QP problem is formulated as a linear complementary problem (LCP) with Kuhn-Tucker conditions. The LCP problem is then solved with Lemke's algorithm, see Section 3.3.

3.1 Formulation of the QP problem

First of all we notice that all four formulations of the mass balance (2.37)-(2.40) have the same structure as the original mass balance (2.4). Therefore, it is sufficient to show how to perform the identification by using model (2.4) as if the measured data consist of $\dot{c}(t_k)$, $c(t_k)$ and $p(t_k)$, k=1,m. When we actually use one of the other four formulations, we only have to modify the input data set to the identification routine. For example, when using formulation 1 $\dot{c}(t_k)$ should be exchanged for $(c(t_k+T_s)-c(t_k))/T_s$ and so on.

Secondly, we observe that the mass balances are not quite adapted to identification since the known tracer gas concentrations and injections are mixed with the unknown flows and volumes. Consequently, our first task will be to rearrange the mass balance equations into the common form Ax=b, where A is a matrix and b a vector which contains all known measurements while x is a model parameter vector which contains all the unknown flows and volumes that we want to identify.

Since all flows are not independent of one another, the model parameter vector can be chosen in many ways, but comparing with (2.4) the following may be a natural choice

$$
\begin{aligned}
x = (&V_1, -Q_{11}, Q_{12}, \ldots, Q_{in}\\
&V_2, Q_{21}, -Q_{22}, \ldots, Q_{2n},\\
&\ldots\\
&V_n, Q_{n1}, Q_{n2}, \ldots, -Q_{nn})^T
\end{aligned}
\tag{3.1}
$$

The vector x has a dimension of n_p*1, where n_p=n(n+1) equals the number of unknown parameters.

Remember that Q_{ii}<0 (equation (2.6)). By using a minus sign on these elements in (3.1) all model parameters are made non-negative. Do also note that the in and outflows to/from the outside are not included in the model parameter vector, but they can easily be calculated with (2.8) and (2.9), respectively.

Now, order the measured data in the following way: First form an m by (n+1) matrix A_i for each cell, i=1,n. Let the rows k=1,m in A_i contain the following measured data set from the k:th sample

$$
(A_i)_{row\ k} = (\dot{c}_i(t_k), -c_1(t_k), \ldots, c_i(t_k), \ldots, -c_n(t_k)), \quad k=1,m \tag{3.2}
$$

Similarly, set up an m by 1 vector b_i for each cell, i=1,n. Let its elements be given by

$$(b_i)_{row\ k} = p_i(t_k)\ ,\ k=1,m \tag{3.3}$$

Now we can form an mn by n_p block matrix A (do not confuse with the n by n matrix $A=V^{-1}Q$), and an mn by 1 block vector b with

$$A=\begin{bmatrix} A_1 & 0 & \dots & 0 \\ 0 & A_2 & \dots & 0 \\ 0 & 0 & \dots & A_n \end{bmatrix} \tag{3.4}$$

$$b=\begin{bmatrix} b_1 \\ \vdots \\ b_n \end{bmatrix} \tag{3.5}$$

With this notation all mn mass balances can easily be written as

$$A\ x = b \tag{3.6}$$

As there are $n_p=n(n+1)$ unknown model parameters we need at least $m=n+1$ samples in order to identify the model parameters. If $m>n+1$, and this is the common case, the system of equations (3.6) will be overdetermined and cannot be exactly satisfied. Instead, our object becomes to find a solution x that minimizes the norm $\|r\|$ where $r=Ax-b$. Depending on which norm we use, the problem can be solved by different methods.

Up to now the linear or l_1 norm has been used. This leads to a linear programming (LP) problem, see Jensen (1987b). But the computations are greatly simplified if we instead use the Euclidean or l_2 norm

$$\|r\|_{l_2} = \sqrt{\sum_i r_i^2} \tag{3.7}$$

The problem can now be stated as a quadratic programming problem. Disregarding the constraints (2.5)-(2.9) the problem can be stated

$$\text{Minimize}\quad \|r\|_{l_2} \qquad \text{where } r = A\ x - b \tag{3.8}$$

The least square solution to (3.7) is given by the well-known normal equations

$$A^T A\ x = A^T b \tag{3.9}$$

and x can be solved as

$$x = (A^T A)^{-1}\ A^T b \tag{3.10}$$

Notice that matrix A has the dimension $mn*n_p$, while matrix A^TA has the dimension n_p*n_p and thus it is not dependent of the number of samples, m. As it is desired to use many samples in order to reduce the influence of measurement errors, this is an important quality. This is also a reason to prefer the QP solution to the LP solution, which deals with the system of equations in its original form (3.6). However, as we do not use the constraints it is not sure that the solution will correspond to a flow system. For example, it may happen that a row sum of the identified flow matrix is positive and thus one inflow would be negative, which is physically impossible. Furthermore, it is unwise not to use all the information that is known about the solution as this can help us to find a better solution. The appropriate solution is therefore to minimize $||r||_{12}$ as above, but to do it subjected to the constraints which we know must be fulfilled. This will be done with the QP approach in the next two sections.

Before we discuss how to formulate the QP problem we shall do some remarks about the normal equations and the choice of pattern for the tracer gas injection.

The least square solution with the normal equations which is the basis for the QP approach does not have the best of reputations from a numerical point of view. This is certainly true when the tracer gas concentrations are highly correlated to each other. The matrix A^TA will then be near singular and the problem will be ill-conditioned. But even when A^TA is calculated, some numerical precision will be lost which cannot be recovered.

To obtain correct result in single precision, it is therefore necessary to form and solve the normal equations in double precision. Because of these reasons, other least square solutions as e.g. QR decomposition, is often preferred to the simpler normal equations. But their solutions will be more complicated when constraints are involved.

Fortunately, when identifying a flow system the demands for accuracy are low. Since the measurement error of the tracer gas injections and concentrations hardly can be lower than some per cent, i.e., uncertainty in the second or third digit, then the normal equations seem to be accurate enough.

However, it should be stressed that it is very important that the tracer gas concentrations are not too similar to each other. Accordingly, the choice of input sequence of the tracer gas injection is very important. The Pseudo Random Binary Signal (PRBS) sequence is frequently used in process identification, see Jung (1987) and may be a good choice. Characteistic for this is that the same sequence, but with different time delays, can be used for all inputs. The mutually correlation between the inputs will be as low (i.e., negative) as possible. To be able to choose suitable parameters to the PRBS sequence, it is necessary to have a feeling of the values of the searched parameters. Preferably, the amplitude should be bigger, the bigger the volumes are, and the period should be of the same magnitude as the time constants according to some rules.

3.2 The QP problem with constraints

We shall now write the overdetermined system of equations (3.6) and its constraints in the standard formulation for a quadratic problem. The constraints (2.5)-(2.7) can simply be written as

$$x \geq x_o \tag{3.11}$$

where x_o is an n_p by 1 vector which elements all are zero. The 2n inequalities (2.8) and (2.9) are linear in the model parameter vector and in matrix notation they are written (after multiplying with -1)

$$G\,x \geq g \tag{3.12}$$

where G is a 2n by n_p matrix and g is a 2n by 1 vector. Each row in G describes one sum of rows or columns of the flow matrix and it consists of n elements with the value +1 or -1 while all other elements are zero. The vector g is the minimal permitted value, or just the lower limit, of the sum. In (2.8)-(2.9) we suppose that the lower limit is zero but, of course, if we e.g. know the value of an exhaust flow then the corresponding lower limit should be set to this value.

It is also easy to extend the number of linear constraints. Every new constraint is written as an extra row of G and g. The row elements in G define a linear combination of model parameters and the corresponding row in g is its lower limit. For example, if we know that

$$v_1 + v_2 \geq v_{min}$$

this is written as

$$(G)_{new\ row} = (\ 1 \quad 0 \quad 0 \ldots \quad 1 \quad 0 \quad 0 \ldots 0\ )$$

$$(g)_{new\ row} = v_{min}$$

where the two '1' are placed in the same locations as v_1 and v_2 have in x^T.

Inequalities which contain an upper limit must be converted by multiplying with -1. For example

$$v_1 + v_2 \leq v_{max}$$

is written as

$$-v_1 - v_2 \geq -v_{max}$$

and can then be treated in the same way as above.

Equality constraints are expressed by two inequalities. For example

$$v_1 + v_2 = v_{sum}$$

is written as

$$v_1 + v_2 \geq v_{sum}$$

$$-v_1 - v_2 \geq -v_{sum}$$

To fix a model parameter x_i to zero, it is sufficient to use the inequality $-x_i \geq 0$ since the inequality $x_i \geq 0$ already is given by (3.11).

An alternative to fix parameters is, of course, to replace the parameter by its value in the mass balances. This will decrease the size of the problem instead of increasing it but, on the other hand, it requires more programming since it is not so easily described.

As seen, the rows of G have no restrictions such as they must be linear independent. But, naturally, there must not be any contradictory demands. In that case a solution does not exist.

Assuming that besides (2.8)-(2.9) we use n_e extra inequality constraints, then G and g have the dimensions of n_g by n_p and n_g by 1 respectively, where $n_g=2n+n_e$.

Finally, we shall rewrite (3.8) so it fits to the standard form of quadratic programming

$$r^Tr = (b-Ax)^T(b-Ax) = b^Tb - b^TAx - x^TA^Tb + x^TA^TAx \qquad (3.13)$$

The first term is constant and can be disregarded in the minimizing. The two following terms are equal and can be expressed by a single term.

Introduce the n_p by 1 vector $c=-2A^Tb$ and the n_p by n_p matrix $Q=A^TA$ (not to be confused with the n by n flow matrix Q).

Now the QP problem can be stated in its standard form

$$\text{minimize:}\quad f(x)=c^Tx+x^TQx \tag{3.14}$$

subject to

$$G\,x \geq g$$
$$x \geq 0$$

The main difficulty when solving this QP problem is to know which constraints are binding and which are non-binding in the optimal solution. If we had known all the binding constraints, these constraints could be replaced by equality constraints and the other could be rejected. In this case the solution had been given by the well-known method of Lagrange multipliers. From the computational point of view this had only led to a minor expansion of the normal equations.

Kuhn and Tucker have developed an extension to the method of Lagrange multipliers to deal with inequality constraints (see Appendix for a short description). Using this the QP problem above can be transformed into an algebraical problem which is easier to deal with.

3.3 The solution of the QP problem

In the QP problem (3.14) all constraints are linear and as $Q=A^TA$ is positive definite it follows that the object function is convex. Thus there are no local minima and the Kuhn-Tucker conditions point out the global one.

As we have two constraints in (3.14) there will be two sets, u and v, of Lagrange multipliers. According to the Kuhn-Tucker condition (A.2)-(A.5), the solution (x,u,v) must satisfy the following system of equations

$$c^T + x^T(Q+Q^T) - u^TG - v^T = 0 \tag{3.15}$$

$$Gx-g \geq 0$$

$$x \geq 0$$

$$u^T(Gx-g) = 0$$

$$v^Tx = 0$$

$$u \geq 0$$

$$v \geq 0$$

where u is an n_g by 1 vector of Lagrange multipliers for the n_g inequality constraints and v is an n_p by 1 vector of Lagrange multipliers for the n_p constraints that the model parameter vector is non-negative.

If we introduce an n_g by 1 slack vector $s \geq 0$ then we can write the first inequalities as an equality

$$G\,x - g = s \tag{3.16}$$

The system of equations (3.15) can now be written

$$\begin{bmatrix} v \\ s \end{bmatrix} = \begin{bmatrix} Q+Q^T & -G^T \\ G & 0 \end{bmatrix} \begin{bmatrix} x \\ u \end{bmatrix} + \begin{bmatrix} c \\ -g \end{bmatrix} \tag{3.17}$$

subject to

$$v^T x + s^T u = 0$$

$$v \geq 0 \ , \ s \geq 0 \ , \ x \geq 0 \ , \ u \geq 0$$

This can be written more compact by introducing the following symbols

$$M = \begin{bmatrix} Q+Q^T & -G^T \\ G & 0 \end{bmatrix} \tag{3.18}$$

$$w = \begin{bmatrix} v \\ s \end{bmatrix} \tag{3.19}$$

$$z = \begin{bmatrix} x \\ u \end{bmatrix} \tag{3.20}$$

$$q = \begin{bmatrix} c \\ -g \end{bmatrix} \tag{3.21}$$

where M is an n_c by n_c matrix with $n_c = n_p + n_g$ and w, z and q are three n_c by 1 vectors. The solution of the QP problem will now be given by the solution of the following problem, called

<u>The linear complementary pivot problem</u>:

Find the vectors w and z so that

$$w = M\,z + q \tag{3.22}$$

subject to

$$w^T z = 0$$

$$w \geq 0 \ , \ z \geq 0$$

This problem has been fully studied in the econometric literature during the last 25 years. The basic method to solve (3.22) is called Lemke's complementary pivot algorithm. This is an iterative method and provided certain assumptions it will always find a solution, if there exists one. Such an assumption that is applicable to our problem is that the matris M is positive semi-definite.

From considerations of space we will not describe Lemke's complementary pivot algorithm in this paper but we refer to the main report Hedin (1989) or rather, to the literature of mathematical programming, e.g. Balinsky and Cottle (1978) which also includes a number of further references. To mention something about the solution of (3.22), we can observe that from the constraints it follows that at least one of the two elements w_i or z_i is equal to zero for i=1,n. The solution thus consists of at most n_g non-zero elements in z and w. As there as well are n_g equations in (3.22), the system of equations may be possible to solve.

There are computer routines published which perform Lemke's algorithm. We have used one of them written by Ravindran (1972). This is a quite uncomplicated program. The complete program listing is covered in about a single sheet of paper. It has newly been revised to be up to date and has proved to work well.

4 NUMERICAL EXPERIMENTS

In this chapter we shall test the identification method and its different formulations. Only a few of the simulated identifications will be discussed in this paper. To make it easier to compare the results with prior works, we shall use the same flow system, pattern of tracer gas injection and initial conditions as in Jensen (1987b).

4.1 A simple testsystem

The simulated flow system consists of three cells with volumes and flows as shown in Figure 4.1.

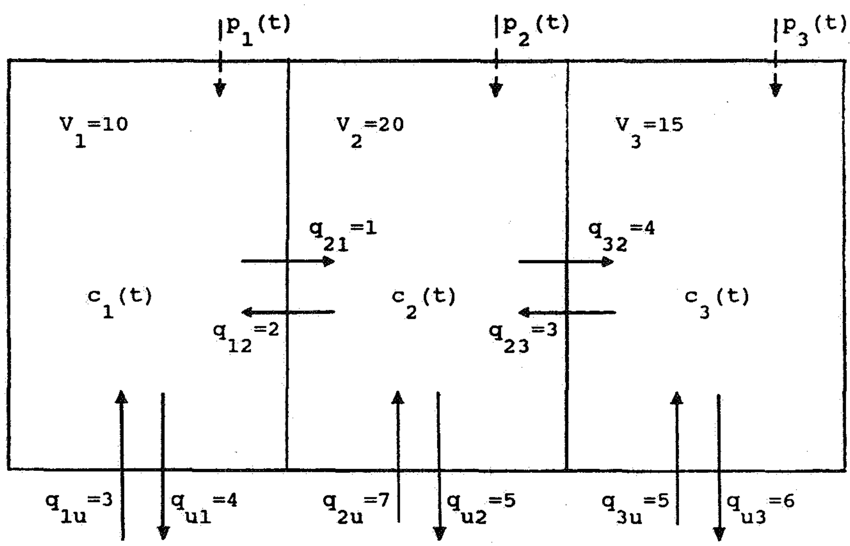

Figure 4.1 The simulated flow system. Units are arbitrary.

The volume and flow matrices are given by

$$V = \begin{bmatrix} 10 & 0 & 0 \\ 0 & 20 & 0 \\ 0 & 0 & 15 \end{bmatrix} \quad \text{and} \quad Q = \begin{bmatrix} -5 & 2 & 0 \\ 1 & -11 & 3 \\ 0 & 4 & -9 \end{bmatrix}$$

The three time constants $TC_i=v_i/q_{it}$ are 2.00, 1.82 and 1.67 time units, respectively.

The flow system has been simulated with the discrete time model (2.13) The initial values were $c(0)=(0.8, 1.1, 1.2)^T$ and the tracer gas injection in each cell has been varied as a very simple pseudo random binary sequence. Figure 4.2 shows the tracer gas injections and the resulting tracer gas concentrations during the first 30 time units. (In most cases we will only use the first 15 time units).

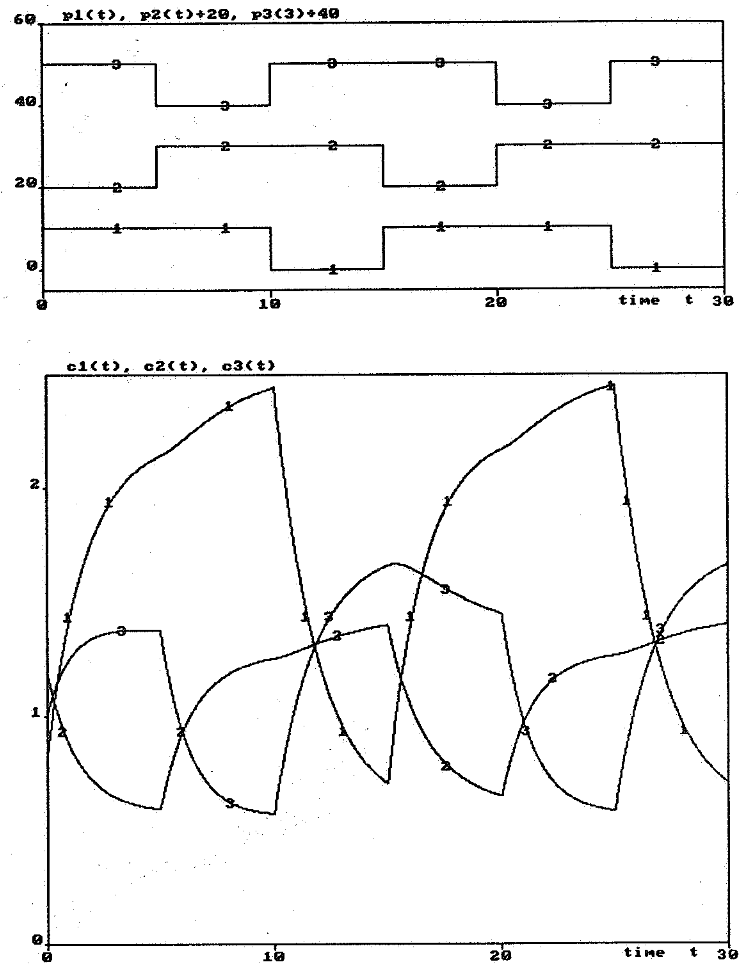

Figure 4.2 Tracer gas injections $p_1(t)$-$p_3(t)$ and the resulting tracer gas concentrations $c_1(t)$-$c_3(t)$ in the simulated three cell system. No measurement noise.

The measurements of the tracer gas concentrations can also be simulated with a random noise. The standard deviation, std, of the noise will be chosen to 0.01, 0.02, 0.05 or 0.10. Setting for example 1 unit equal to 100 ppm NO_2 means that std=0.01 corresponds to a measurement error of 1 ppm. As the standard deviations of $c_1(t)$, $c_2(t)$ and $c_3(t)$ are about 0.59, 0.26 and 0.40 respectively, we can expect the parameters to be best estimated in cell 1 according to the different signal to noise ratios.

4.2 The criterion for the best model

There is no obvious criterion to compare different parameter estimations. The loss functions or norms $\|r\|$, which are minimized by the QP routine do not help us as they cannot be compared. Especially we cannot compare a LP with a QP solution by the norms. But, since the true parameters are known it will be natural to compare any norm of the error Δx of the model parameter vector

$$\|\Delta x\| = \|\hat{x} - x\| \tag{4.1}$$

where $\hat{x}$ denotes the estimated parameters and x the true parameters. A minor modification from this has been done as the n parameters q_{ii} have been exchanged by the 2n calculated in and outflows (q_{iu} and q_{ui}). The underlying idea was to calculate the estimation error on the primary physically parameters.

Depending on the norm chosen we will calculate the mean absolute error (linear norm), maximum absolute error (maximum norm) and mean error and standard deviation. (The two last mentioned come from the Eucledian norm after the splitting $E(\Delta x^2) = E^2(\Delta x) + Var(\Delta x)$).

A weakness of the norm (4.1) is that it only takes the absolute errors into account. This will lead to a quite too large influence on the absolute norms from the big volumes. An alternative is to work with the norm of the relative error

$$\|(\hat{x} - x)/x\| \tag{4.2}$$

Also the norms of the relative errors will be calculated. Two of the model parameters, q_{13} and q_{31}, are equal to zero. These parameters will of course be excluded when the norms of (4.2) are calculated.

4.3 Simulated experiments

If there is no noise, then the model parameters can be determined exactly with the model based formulations 2 and 4. The number of iterations that is required will depend on how good the start solution is. The lesser the sample period is, the better the start solution will be.

In the presence of noise, however, we can no longer suppose to find the correct model parameters. How large the estimation error will be depends not only on the noise level but also on the formulation used and the choice of sample period. For the two integrated formulations it will also depend on the choice of interval of integration. Tables 4.1 and 4.2 show the estimated parameters of the test system determined from the simulated experiment in Figure 4.2, but at different noise levels. The formulations 2 and 4, respectively, are used to express the mass balance equations. At the lower noise levels, the advantages over the non-iterative formulations 1 and 3 (not shown in this paper) are considerably but at the higher noise levels the differences are small and for std=0.1, the non-iterative formulations even give a somewhat better result. The latter means that, at this noise level, the estimation of the matrices D and R, S, T respectively, cannot be improved from the simple initial values.

Usually, a straightforward way to obtain better parameter estimations is to use more measurements. Table 4.3 shows the estimated parameters at the lower noise level, std=0.01, when the sample period is reduced from T_s=1 to 0.5, 0.2 and 0.1. However, the estimations get worse. Neither is it possible to improve the estimations significantly by lengthening the total experiment time. These unexpected results are due to the bias of the estimation. In the main report, see Hedin (1989), the expected bias is calculated in some simplified cases. It is shown that the bias is increasing when the sample period is decreasing. It is also shown that the bias is considerably higher in the estimation of the volumes than in the estimation of the flows. Another finding is that the bias in the estimation of the flows will not be lower, but higher, if the volumes are known and fixed to their correct values. This surprising result is also confirmed by simulated experiments.

A way to obtain better estimations that work, is to use formulation 3 or 4 with a sample period that is shorter than the interval of integration. Table 4.4 shows the estimations for formulation 4 when the interval

Table 4.1

Experiment		A0	A0	A0	A0
length		15.0	15.0	15.0	15.0
sampling period		1.0	1.0	1.0	1.0
noise st.dev.		0.01	0.02	0.05	0.10
Parameters	true	estimated	estimated	estimated	estimated
q11	-5.00	-5.02	-5.03	-5.08	-5.18
q12	2.00	1.99	1.99	1.98	1.97
q13	0.00	0.00	0.00	0.00	0.00
q21	1.00	1.02	1.02	0.91	0.61
q22	-11.00	-11.05	-11.06	-10.96	-10.52
q23	3.00	3.01	3.01	3.01	2.94
q31	0.00	0.00	0.00	0.00	0.10
q32	4.00	4.02	4.03	3.95	3.55
q33	-9.00	-9.02	-9.02	-8.99	-8.87
q1u	3.00	3.02	3.05	3.11	3.21
q2u	7.00	7.02	7.03	7.04	6.98
q3u	5.00	4.99	5.00	5.04	5.22
qu1	4.00	4.00	4.01	4.17	4.48
qu2	5.00	5.03	5.05	5.03	5.00
qu3	6.00	6.01	6.01	5.98	5.93
V1	10.00	9.93	9.86	9.60	9.08
V2	20.00	20.05	19.76	17.17	10.90
V3	15.00	14.81	14.49	12.91	9.32

Estimation errors	abs	rel%	abs	rel%	abs	rel%	abs	rel%
mean error	-0.01	0	-0.05	0	-0.34	-3	-1.05	-10
standard dev	0.06	1	0.15	1	0.88	6	2.67	19
max abs err	0.19	2	0.51	3	2.83	14	9.10	45
mean abs err	0.03	1	0.08	1	0.39	4	1.18	13

Table 4.2

Experiment		A0	A0	A0	A0
length		15.0	15.0	15.0	15.0
sampling period		1.0	1.0	1.0	1.0
noise st.dev.		0.01	0.02	0.05	0.10
Parameters	true	estimated	estimated	estimated	estimated
q11	-5.00	-5.01	-5.03	-5.07	-5.14
q12	2.00	1.99	1.98	1.95	1.89
q13	0.00	0.00	0.00	0.00	0.00
q21	1.00	1.03	1.05	0.97	0.63
q22	-11.00	-11.05	-11.10	-11.14	-10.56
q23	3.00	2.99	3.01	3.10	3.01
q31	0.00	0.00	0.00	0.00	0.04
q32	4.00	4.02	4.04	4.01	3.71
q33	-9.00	-9.02	-9.03	-9.04	-8.88
q1u	3.00	3.02	3.05	3.12	3.25
q2u	7.00	7.03	7.05	7.06	6.91
q3u	5.00	4.99	4.99	5.02	5.13
qu1	4.00	3.98	3.98	4.10	4.47
qu2	5.00	5.04	5.08	5.17	4.96
qu3	6.00	6.02	6.03	5.93	5.86
V1	10.00	9.94	9.87	9.65	9.29
V2	20.00	20.07	19.74	16.83	10.07
V3	15.00	14.81	14.48	12.91	9.59

Estimation errors	abs	rel%	abs	rel%	abs	rel%	abs	rel%
mean error	0.00	0	-0.04	0	-0.34	-2	-1.08	-9
standard dev	0.06	1	0.16	2	0.96	6	2.82	19
max abs err	0.19	3	0.52	5	3.17	16	9.93	50
mean abs err	0.04	1	0.08	1	0.42	4	1.20	13

Table 4.3

Experiment		A0	A0	A0
length		15.0	15.0	15.0
sampling period		0.5	0.2	0.1
noise st.dev.		0.01	0.01	0.01
Parameters	**true**	**estimated**	**estimated**	**estimated**
q11	-5.00	-5.04	-5.05	-5.16
q12	2.00	2.05	2.09	2.31
q13	0.00	0.00	0.00	0.00
q21	1.00	1.02	0.85	0.58
q22	-11.00	-11.00	-11.06	-11.07
q23	3.00	2.97	3.21	3.57
q31	0.00	0.01	0.20	0.51
q32	4.00	3.98	3.55	2.83
q33	-9.00	-9.00	-8.96	-8.90
q1u	3.00	2.99	2.96	2.85
q2u	7.00	7.01	7.00	6.93
q3u	5.00	5.01	5.21	5.56
qu1	4.00	4.01	4.00	4.07
qu2	5.00	4.97	5.41	5.93
qu3	6.00	6.03	5.75	5.33
V1	10.00	9.82	9.46	8.39
V2	20.00	19.64	15.84	10.13
V3	15.00	14.84	13.03	10.03

Estimation errors	abs	rel%	abs	rel%	abs	rel%
mean error	-0.04	0	-0.43	-4	-1.07	-9
standard dev	0.11	1	1.17	9	2.82	23
max abs err	0.36	3	4.16	21	9.87	49
mean abs err	0.06	1	0.58	7	1.46	20

Table 4.4

Experiment		A0	A0	A0
length		15.0	15.0	15.0
sampling period		0.5/1.0	0.2/1.0	0.1/1.0
noise st.dev.		0.01	0.01	0.01
Parameters	**true**	**estimated**	**estimated**	**estimated**
q11	-5.00	-5.03	-5.02	-5.02
q12	2.00	2.03	2.02	2.03
q13	0.00	0.00	0.00	0.00
q21	1.00	1.03	1.01	1.02
q22	-11.00	-11.04	-11.00	-11.02
q23	3.00	3.00	2.98	2.97
q31	0.00	0.01	0.00	0.00
q32	4.00	4.04	4.02	4.01
q33	-9.00	-9.05	-9.02	-9.02
q1u	3.00	3.00	3.00	2.99
q2u	7.00	7.01	7.02	7.03
q3u	5.00	5.00	5.00	5.00
qu1	4.00	3.99	4.01	4.00
qu2	5.00	4.97	4.96	4.97
qu3	6.00	6.05	6.05	6.05
V1	10.00	9.89	9.93	9.93
V2	20.00	20.12	19.94	20.03
V3	15.00	14.92	14.90	14.92

Estimation errors	abs	rel%	abs	rel%	abs	rel%
mean error	0.00	0	-0.01	0	0.00	0
standard dev	0.05	1	0.04	1	0.04	1
max abs err	0.12	3	0.10	1	0.08	2
mean abs err	0.03	1	0.03	1	0.03	1

of integration is 1.0 and the sample period is 0.5, 0.2 and 0.1. In this case the bias is not affected while the random error is reduced when more measurements are used.

4.4 Comparison with linear programming

Most of the shown identifications have also been solved with the linear programming method. This has been easy to do as Lemke's algorithm can also solve the LP problem, (see Hedin (1989)). The conclusions of these identifications are that the differences between the QP and LP solutions are surprisingly small. This is true, no matter what criterion is used, linear, maximum or Eucledian. Even the bias seems to be approximately the same. The great difference is instead the execution time required to solve the QP and LP problems. As mentioned earlier the dimension of the LP problem will depend on the number of measurements. Stated as a complementary problem the dimension of the LP problem will be $n_c = n_g + n_p + 2mn = 2n + n(n+1) + 2mn$ while the dimension of the corresponding QP problem will be only $n_c = n_g + n_p$. The following table shows the cpu time when one of the problems is solved with the QP and LP methods, respectively.

m	QP method n_c	cpu time (s)	LP method n_c	cpu time (s)
15	18	0.05	108	59
30	18	0.05	198	377
75	18	0.05	378	4383

When the number of measurements is increasing, then the cpu time for the LP solution is increasing exponentially. The exponent has been calculated to 2.676. Taking the first case m=15 as a reference, this means

$$\text{cpu time }(m) = \text{cpu time }(15)\ (m/15)^{2.676} \tag{4.3}$$

In other cases the exponent has been calculated to 2.6 - 2.9. This is somewhat lower than 3 which is the expected value from a theoretical point of view.

4.5 Concluding remarks

One of the objects of this paper was to see if the QP solution is competitive with the LP solution. The conclusion is that it certainly is competitive.

In contrast to the LP method, the QP method is fast enough to be used in real-time applications even on a PC based system. Thus it is possible, and advisable, to incorporate the identification procedure into the computer programs which control the tracer gas injections and measurements. Now the identification can be done on-line and the estimated parameters can be followed. The advantages of knowing the latest model is obvious. It will give an immediate check that an experiment in progress is sufficiently informative so the parameters can be determined. It will also indicate when an experiment can be completed. A possibility may also be to let the computer adapt the tracer gas injections to the actual flow system.

Other advantages of the QP method that may be mentioned is that it is possible to estimate the error of the determined parameters. This is done by means of the covariance matrix. Further, it is easy to include different weights of the measurements. This is done by computing Q as A^TWA, where W is a weight matrix. It is also easy to track slowly time varying parameters. This is done by computing $Q=A^TA$ as a filtered sum of inputs

$$Q(t_k+T_s) = f\,Q(t_k) + (1-f)(A_{t_k}{}^T A_{t_k})$$

where f, the 'forgetting factor', is a measure of how fast old data are forgotten. It usually has a value close below one. A lower value gives a faster tracking but a higher sensitivity to noise.

APPENDIX

Kuhn and Tucker have made an extension to the method of Lagrange multipliers to deal with minimizing problems with inequality constraints. Consider the following problem

Minimize f(x) (A.1)
subject to

$$g_j(x) \geq 0\ , \quad j=1,n_g$$

According to the Kuhn-Tucker conditions, every optimal solution (x,u) must fulfill the following necessary conditions

$$\nabla f(x) - \sum_j u_j\, \nabla g_j(x) = 0 \qquad \text{(A.2)}$$

$$g_j(x) \geq 0\ , \quad j=1,n_g \qquad \text{(A.3)}$$

$$u_j g_j(x) = 0\ , \quad j=1,n_g \qquad \text{(A.4)}$$

$$u_j \geq 0\ , \quad j=1,n_g \qquad \text{(A.5)}$$

If all constraints are linear and the object function f(x) is convex, then these conditions will also be sufficient for (x,u) to be the optimal solution. In this case, there are no local minima and the conditions (A.2)-(A.5) point out the global one.

For a detailed treatment of the Kuhn-Tucker conditions, see any textbook in quadratic programming as e.g. Van de Panne (1975).

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PROGRESS AND TRENDS IN AIR INFILTRATION
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Poster 16

THE COMIS INFILTRATION MODEL

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1. SYNOPSIS

The COMIS workshop (Conjunction Of Multizone Infiltration Specialists), using a multi-national team, is planning to develop a reliable, smooth running multizone infiltration model on a modular base. This model not only takes crack flow into account but also covers flow through large openings, single-sided ventilation, cross ventilation and HVAC-systems. The model contains a large number of modules which are peripheral to a steering program. COMIS can also be used as a basis for future expansion in order to increase the ability to simulate buildings. Small task groups were formed to work on particular problems in developing the modules. Each COMIS team member works on several task groups.

2. INTRODUCTION

The first COMIS *Newsletter* was send to colleagues almost three years ago to inform them about the joint research project being planned to develop a multizone infiltration model at Lawrence Berkeley Laboratory (LBL). Even though this kind of co-operation is well established in other fields of research, e.g., high energy physics, in the field of building physics it is new to engage in a research project which one individual or country would not be able to do alone. From the beginning the COMIS idea was well received. Owing to the diverse background of the group several national and international research programmes are co-ordinated with the COMIS workshop.

Special emphasis has been given to the input/output routines so that the final program should not only be "user-tolerant" but "user-friendly." It is being developed in such a way that it can be used either as a "stand-alone infiltration model" or as an "infiltration module" of a building simulation program. The input/output procedure is therefore being developed in such a way that either the COMIS input/output modules can be used or only the input/output interface. This makes it possible for the user to connect the program with other software (e.g., CAD-systems).

The input program section is subdivided into several separate modules with each of them requiring specific data. The modules will provide backup from data bases for default value input and for checking terminal input. However, these databases will not be filled extensively within the COMIS workshop.

The building description section will allow for a 3D repetitious input of wall elements to build up rooms, floors and complete wings of the building. Physical properties are then assigned to each wall. The program also allows for direct generation of the flow network.

One of the major tasks has been to find a method of determining the wind pressure distribution for a building according to measured data from available literature. This will allow building designers to work with the COMIS model even if wind tunnel results are not available for the building under consideration.

Crack flow, large openings and mechanical ventilation systems can be modelled by COMIS. Furthermore, additional flows which do not influence the pressure distribution in the network in a major way, i.e., simultaneous two way flow at large openings and wind turbulence effect at single-sided windows etc., are studied. The correction of coefficients of power law for crack flow, taking into account the effect of the temperature distribution of air in the crack is also studied.

Calculating the infiltration and ventilation flow rates requires the solution of a non-linear system of equations. The main task has been to find an efficient solving method. The starting point is the Newton-Raphson method which, in most cases, allows the system of equations to converge rapidly. The method is modified to avoid occasional convergence problems when working on power functions but in principle, it is a question of finding an appropriate relaxation coefficient. Most work done up to now concerns efficient linear methods. The special characteristics of the linear system of equations have been studied, and based on this, a direct method has been developed, timed, and documented. An iterative method is under investigation.

The aim of the experiments being developed in the COMIS project is to provide real data for multizone model validation. Two full scale experiments are planned. One will be performed by the EPFL group in Lausanne using the LESO facility and the second is under way in the Bay area using a multizone house. This house has been used by the Radon group at LBL and is available for COMIS throughout the workshop.

A handbook will be written to describe ways of using the model and its physical background. It will describe the program structure and each of the modules in the COMIS library as well as the input and output of the program. Special emphasis will be given to the use of the model starting with the installation procedure and going on to describe how to connect COMIS with other programs (interfaces) as well as explaining the input. The input data set for a simple example will be included in the handbook.

The features of expert systems have been studied in terms of function, data structure, and development procedures. We found that most of the expert systems were developed for specific problems and are still prototypes in the research stage. They need to be further developed for practical use.

Applying the expert system to multizone infiltration would have many benefits to both users and researchers. A CAD system as a user interface to describe a building would be especially helpful in developing a user-friendly system.

3. STRUCTURE

3.1 General

Although plans for COMIS began in 1985 the need to think about the structure of the model in detail came up at the first COMIS meeting in Lausanne in the fall of 1987. To start a one year workshop with 8 participants offered the chance of creating the so-called "worlds best ventilation model". But selecting the best methods for input processing, data handling and calling routines and even defining the precise goals for the model easily could take months. Therefore, a proper common census program structure and a definition of the goals of the program became necessary.

3.2 Related processes

Ventilation, heat flow and pollutants are strongly combined in a building. In studies of ventilation processes, heat flows and pollutants are generally not allowed to be ignored. Most of the problems in buildings nowadays deal more with indoor air quality, energy and ventilation rather than with ventilation only and a connection is even made with sound, lighting and comfort. As occupants of the building can have a major influence on all these processes the effect of occupant behaviour also cannot be ignored. This pointed to the need to simulate all the physical processes in a building as precisely as possible in order not to exclude the possibility of combining the different phenomena.

3.3 Problem solving

The number of different problems to be solved using such a model requires flexible routing procedures. For example, a possible way to solve ventilation problems is well demonstrated in the "Air Infiltration Calculation Techniques" guide [1]. Depending on the desired degree of accuracy a multizone building can be simulated using a smaller number of zones in the COMIS program or even with a regression formula. The results of this can be re-diverted for instance, into the different flows required by the user. To make the correct decisions for routing the program and to check input and results, a "problem definition" and a data storage for "administrative control data" are necessary. An expert system is the obvious way to have flexible control of these processes but although Berkeley is world-famous for such systems the possibility of including one in the COMIS program appeared to be low. This is partly because of our need for a system that would be capable of handling graphics. As some of the COMIS participants will develop expert systems in the near future it was decided to develop the COMIS program without an expert system and later replace the "IF - THEN - ELSE" decision section with a rule base for the expert system.

3.4 Data

Data storage should be flexible in containing information about a building and its physical properties at almost any level of detail. As this includes full geometrical description of the building a CAD system would be preferable. As the code should be free of proprietary software the possibilities for including a CAD were referred to "later add-on's by the user". The input data for the program will include:

- Problem Definition
- Building
- Operation Schemes
- Environment
- Meteo
- Wind Pressure Coefficients
- Air Leakage Values

In order to be self-explanatory most of the data files will include text headers.

3.5 Features

One of the thresholds for an international program is the use of non-SI units. The program will operate internally with SI units but as the input allows for text headers these text headers will be interpreted to look for other units. The conversion factors will be looked up in a data store. Units mentioned for the results will be used to translate the output from SI into other units. A language text resource will be used for most input/output at the user interface. This allows for an easier translation into different languages.

3.6 Structure

The COMIS structure we now use includes schematic drawings of 87 modules and shows the relations between these and the major data flows. This is helpful in preventing flow chart errors in the program and in locating missing functions or routines. Besides this structure the list of parameters used in the main programs and the list of the prepared and finished modules is frequently updated.

4. INTERACTIVE INPUT PROGRAM

The input is given using seven different blocks, namely:

1) Problem Description
2) 3D-Building Description
3) Direct Network Description

4) Operating Schedules Input
5) Cp - Value Input
6) Environment Description
7) Meteo Description

The Problem Description Section asks for run control parameters and for the input type and then guides the user through the specific input modules.

The Building Description Section allows for repetitive 3D input of wall elements to build up rooms, floors and complete wings of the building. Physical properties are then assigned to each wall. The program also allows for direct generation of the flow network. A well defined interface for the geometric data will not only allow for the generation of simple graphic echo of the building geometry but also for a possible future linking to a CAD system.

Type as well as subtype definition are used to characterize airflow components. The description of these components relies on the airflow component database which can also be used for default properties data input providing, e.g., standard leakage values.

The format of the airflow component properties database is set up for demonstration purposes only and must be defined in more detail --considering already established databases (e.g., ASHRAE [2] or AIVC handbook [1]) as well as the results of ongoing research projects (eg. IEA-ECB Annex XX). It is therefore planned to add a database management module to the input program later on. This module also will include a routine for the polynomial description of fan components.

The Operation Schemes module allows for the input to, or for the assignment of, all kinds of schedule files or of parameters for program internal schedule-generators.

The Cp-Values module sets up the cp-value data for the specified facade elements including backup input from the database. As a special option a routine for the calculation of cp-values is available which is based on data given in the 3D building description and the description of the environment.

The Environment description asks for the non-time dependent surrounding parameters and the obstacles geometries. The Meteo Description module allows for input of weather data from the terminal as well as from separately generated weather data files.

A prototype version of the interactive input programme module has been established as described in the last COMIS Newsletter [3].

5. LARGE OPENINGS

5.1 General

Airflow rates through doorways, windows and other common large openings are significant ways in which air, pollutants and thermal energy are transferred from one zone of a building to another [4].

However in a previous review [5] of multizone infiltration models, made in 1985, none of the described codes were able to solve this problem in any other way than to divide the large opening into a series of small ones described by crack flow equations.

COMIS's contribution to this fundamental problem will be to describe the physical problem, review the various solutions developed in the literature and compare these solutions using both a numerical and a physical point of view.

5.2 Integration of Large Openings in a Multizone Infiltration Model

Basically, a multizone infiltration model like COMIS is defined by a network description of the pressure field in a building. The pressure nodes or zones are linked by non-linear resistances and the law of mass conservation in each zone leads to a non-linear system of pressure equations.

It is obviously possible to describe the behavior of large openings in two ways. Either one can describe the air flow rate through a large opening using a non-linear equation of the pressure drop or one must solve this singular problem separately and include the results as an unbalanced flow in the mass conservation equation of the described zone.

Both solutions are currently being investigated by COMIS.

5.3 Unbalanced Flow Approach.

The basic problem of instantaneous air transport through a large opening linking two zones with different air densities can be solved analytically in the case of an incompressible and inviscid flow in steady conditions by using Bernouilli's equation. In the case of an existing supply of air in one zone, or of different thermal gradients from both sides of the opening, a solution can also be reached by using numerical tools.

These solutions, based on a fluid mechanics approach, have been already developed by COMIS. They lead to the definition of an air mass flowing in both directions through the opening.

As natural convection is usually the main driving phenomenon another solution consists of using empirical correlations which give the total heat transfer rate through the opening as a function of its geometry and of the thermal state of each zone. The heat transfer can then be converted to a mass flow rate using calorimetry equations.

In the case of both solutions the large opening is disconnected from the general pressure network and is solved separately. It is then represented by an unbalanced flow in the mass conservation equation of each zone.

5.4 Introducing Large Openings in the General Network.

To connect large openings to the general network, one must define their behavior in terms of nonlinear equations of the pressure drop.

The first idea is to describe the large opening as a conjunction of parallel small openings. Each small opening is then described by a crack flow equation taking into account the local pressure drop, and the whole system of nonlinear equations can be introduced directly in the pressure network [6,7].

Another possibility consists of interpreting the flow equations given by the fluid mechanics approach in terms of pressure. This method leads to the definition of new flow equations in pressure characterizing the behavior of large openings.

All these elementary solutions are going to be tested and compared in order to define clearly the limits and the advantages of each one.

5.5 Unsteady Wind Effect and Turbulence

The general laws demonstrated by thermal or fluid mechanics approaches are also valid for large exterior openings in steady state conditions. But none of these methods enables us to quantify the effect of an unsteady wind or large scale turbulences.

Experimental results have shown that these effects can be particularly significant in the case of one-sided ventilation. Nevertheless very few correlations have been proposed and most of those that have concern particular configurations. It seems difficult, therefore, to introduce these effects in a general way in our first model. However, we will hope to do so later on as an improvement to COMIS.

6. CRACK LEAKAGE PERFORMANCE

6.1 General

The temperature of air flowing through a crack depends on the following factors:

- air flow rate
- air temperatures of the zones on both sides of the crack
- dimensions and form of crack

In most cases the temperature of the air in a crack is quite different from the temperatures of the the zones on either side of the crack. Furthermore, air leakage performance measurements are usually performed in a certain temperature condition but used at different temperatures. The temperature variation, however, has a big influence on the air leakage flow due to changes in the air viscosity and air density. Unfortunately, almost all the models dealing with air leakage characteristics ignore this phenomenon.

6.2 Crack Flow Equation

Data obtained from measurements on crack models show that, for turbulent crack flow, the mathematical description of the friction factor is identical with the one found for conduit flow with smooth walls. Therefore crack flow can be seen as duct flow with a more complicated flow path.

6.3 Correction Factors for Temperature Influence

We found from the crack flow equation research, that the flow performance is strongly temperature dependent. In order to arrange the results in the usual form we have introduced correction factors which account for the temperature influence. The correction factor depends on the type of leakage. We have developed three different equations for the different correction factors.

6.4 Crack Form and Air Flow Temperature

We can easily build an air leakage temperature module according to the crack forms. Fortunately, we found that the crack form mainly depends on the structure of the building or on the type of building component and that its size depends on the workmanship. We therefore classify crack forms into three groups: double frame windows, single frame windows and doors, and walls.

6.5 Crack Types

Double Frame Windows

The air passing through the window unit is well mixed with the air contained in the space between the two frames. The air temperature depends on the flow direction, flow rate and window structure etc. The air leakage temperature is assumed to be the air temperature in the air space of the window.

Single Frame Windows and Doors

The air has only a very short distance to pass through the crack and therefore, the influence of the crack surface temperature is quite small. Most of the air flowing through the crack does not come from the well-mixed part of the zone, but from the boundary of natural convective heat transfer. The air leakage temperature is therefore assumed to be equal to the boundary air temperature at the high pressure side of the flow path.

Walls

The form of cracks in a wall can be divided into three groups. Straight cracks, whose flow length is approximately equal to the thickness of the wall, belong to the first group. The second group covers labyrinth cracks in which air travels a long distance before it leaks out. A typical example of this type of crack is the air space wall. The third group covers walls in which air leaks homogenously through the wall. We have established the physical and mathematical models for all three crack forms but up to now use only the equation formulated for the third group.

7. HVAC SYSTEMS

7.1 General

HVAC-Systems (Heating, Ventilating and Air-Conditioning Systems) are composed of ducts, duct fittings, junctions, fans, air filters, heating and cooling coils, air-to-air heat exchangers, flow controllers, etc. Several of the program modules concerning ventilating systems have already been developed allowing us to calculate the coefficients of the flow equation for duct works with fittings, the static pressure losses for T-junctions and the volume flow rate of a fan as well as for a flow controller as a function of the pressure difference.

7.2 Flow Coefficients for Ducts

This module calculates the flow coefficients of the flow equation for ducts including duct fittings. The flow coefficients C and n are calculated using the following procedure:

1) An estimated volume flow rate through a duct is given as input data. If the flow is in the turbulent range the duct friction factor is calculated by an approximate explicit equation from Moody (instead of the Colebrook equation, which is an implicit expression). If the flow is in the laminar region the duct friction factor is calculated dividing a constant by the Reynolds number. In the transitional range the factor is taken as an interpolated value between the two transitional points. The pressure loss along a duct, including dynamic pressure losses through duct fittings, is calculated.
2) In order to obtain the flow coefficient and the pressure exponent the pressure losses for an alternative volume flow rate are automatically calculated in the module. That value of the flow is 10 % higher in the turbulent range and 10 % lower in the laminar range than the initial value.
3) The flow coefficient of C and n are obtained by the straight line connecting the two points plotted on a log-log chart. The exponent is given by the slope of the curve. The exponent n and the initial values of the flow rate and the pressure difference determine the coefficient C.

7.3 Static Pressure Loss at T-junctions

Since the duct systems are described by a network in the air flow model the junction is treated as a pressure node. Input data are the three volume flow rates through the three ducts which are connected at the junction as well as the static pressure at the point in a duct just before the junction. The output are the static pressures at the two points in the two ducts just after the junction.

There are some data available in the literature for the pressure loss coefficients at the T-junction. We obtained data from the ASHRAE Handbook of Fundamentals [2], German handbooks [8,9], the final report of IEA-Annex X "System Simulation" [10] and the Dutch building standard [11], as well as a Japanese research paper [12]. To our surprise we found that the values of the pressure loss coefficients were significantly different according to the sources. For example, in the case of converging flow, the pressure loss coefficient through the main duct of the T-junction obtained from one source is double the value of the loss coefficient given in another.

As a first step we use the data prepared by Ito and Imai [12], since all six pressure loss coefficients for the four possible cases of flow patterns at a T-junction are expressed by empirical equations. Furthermore, his values are close to the values presented in the ASHRAE Handbook.

7.4 Fan Performance

The fan performance curve is expressed on the basis of more than three data sets of the volume flow rate and the pressure difference, by the polynomial approximate formula using the least square method. If the pressure difference is outside the normal operation range the fan performance is expressed by the straight line connecting the two points which show the maximum pressure difference and the minimum pressure difference in the normal operating range. Input data is a pressure difference; output is the volume flow rate calculated from the fan curve.

A module has also been prepared for correcting the fan performance due to the air density.

7.5 Flow controller

The pressure loss curve is expressed by equations based on data sets of the pressure loss and the volume flow rate. The input data is the driving pressure difference of the flow controller. The output data is the volume flow rate.

There are other components connected to the HVAC-Systems which cause dynamic pressure loss, e.g., air filters, heating or cooling coils, different types of junctions, etc. The calculation procedures for these components will be added as soon as possible.

8. WIND PRESSURE COEFFICIENTS

8.1 Evaluation of The Surface Pressure Coefficient

The pressure distribution around a building is usually described by a dimensionless pressure coefficient (Cp), which is the ratio of the surface pressure and the dynamic pressure in the undisturbed flow pattern, measured at a reference height [13].

Wall averaged values of Cp usually do not match the accuracy required for air flow calculation models. More detailed evaluations, taking the Cp distribution on the envelope of buildings into account, can be made in different ways:

- performing full scale measurements when an existing building is being studied
- carrying out wind tunnel tests on models of existing buildings or buildings in the design stage
- generating Cp values by numerical models based on parametrical analysis of wind tunnel test results

The first way is pratically impossible to follow unless done within expensive and time-consuming experimental plans. The second way depends too much on the availability of test equipment and relevant assistance. The third method seems to be the only one assuring easy and wide data access.

Recently, some research has been carried out in this direction (Swami & Chandra [14]) but a lot of work remains to be done, mainly in improving the experimental knowledge of the phenomenom.

8.2 Parametrical Analysis on Pressure Coefficients

In order to calculate the Cp-distribution for buildings we are working on a method based on a parametrical study, to determine the Cp-values. The available methods have been checked by comparing calculated results with findings from wind tunnel tests found in the literature. Since the results did not match the data well, a parametrical analysis of wind tunnel test data, aimed at developing a calculation model for Cp-data, has been carried out.

To find a set of data large enough to cover a wide range of parameters affecting the variation of Cp, several wind tunnel test reports have been considered. Two tests have been chosen as references: Hussein & Lee [15] and Akins & Cermak [16].

We have taken as reference the center line Cp vertical profile of the Hussein & Lee cube-shaped model for wind direction normal to the wall with no surrounding obstacles and at a height equal to the height of the surrounding roughness elements in a low density urban area. Several CP data sets from the tests have been analyzed. The relevant variations at different relative model heights were normalized with respect to the reference profile.

Each data set has been related to a specific parameter among the following:

- Velocity Profile Exponent, characteristic of the roughness
- Plan Area Density, representing the density of surrounding buildings
- Relative Height, ratio of model height to height of surroundings
- Aspect Ratios, model length or width to model height

- Wind Direction Angle, measured from the line perpendicular to each wall
- Relative horizontal position of the point being looked at

Data curves have been fitted in relation to each parameter in order to obtain polinomial equations. Correction coefficients for reference Cp were found. A routine to calculate surface pressure coefficient values at any point on the wall has been developed.

A test of the routine has been carried out by comparing calculated results to Cp values from the reference data sets. The comparison shows the method to be reasonable accurate within the limits of the application and the consistency of the data sets themselves. A comparison with other authors has not been carried out as yet because of lack of data.

In addition to the detailed calculation module a simpler module dealing with wall-averaged Cp values from AIVC data sets [1] has yet to be developed.

9. SOLVER

A building is basically modelled by pressure nodes that are interconnected with air flow links. For one time step, the outside of the building is represented by a fixed boundary condition. The pressures of the internal nodes in the air flow network have to be solved so as to determine the different air flow rates. Solving these infiltration and ventilation flow rates requires the use of a non-linear system of flow equations. The main task was to find an efficient and stable method.

The starting point is the Newton-Raphson method, with derivatives, operating on a node-oriented network which, in most cases, quickly brings about the convergence of the system of equations. The method has been modified to avoid occasional convergence problems when working with power functions. Fortunately, the origin of the convergence problems is well understood. The solving method is working on the flow balance equations and not on the flow equations. If one or several of these balance equations have an exponent close to one-half the Newton-Raphson method will not work well due to the nature of the procedure, in finding the next approximation. One instance when this happens is when a leakage opening with a flow exponent of one-half is predominant in one zone. In this case the flow balance equation will also have an exponent close to one-half. An under-relaxation will increase the convergence velocity and bring us to the solution. In principle it is a question of finding an appropriate relaxation coefficient.

Three methods were tested. First, a constant under-relaxation coefficient can be used for all flow balance equations if the iterations converge only slowly. This is the simplest approach. This method does not require any calculations in finding the coefficient and a coefficient close to one-half is expected to solve the system acceptably. The disadvantage of this method is that the convergence is slow and probably unstable. Because of this no further work has been done using this approach. As a comparison a conventional Newton-Raphson method is used.

Second, a separate under-relaxation coefficient can be used for each flow balance equation. The coefficients are determined as an extrapolation from the two preceding iterations. The main disadvantage seems to be that each coefficient is determined without dependence of the other coefficients. The convergence velocity can therefore not be expected to be optimal. However, the amount of work in determining the coefficients is relatively limited which may speed up the method.

The third approach is more systematic. The problem of finding a relaxation coefficient is considered to be an optimization problem. The coefficient that causes the system to converge fastest is chosen. Therefore, only one coefficient for all zones is used. The disadvantage is that relatively considerable work is required to find the coefficient in each iteration. This method is expected to be very stable.

Furthermore, the derivatives are not defined if the pressure difference across an opening is zero. The closer the exponent of the flow balance equation is to one-half the higher is the risk for an overflow in the computer when the pressure difference is close to zero across the predominant opening. If an efficient relaxation method is used these problems may eventually arise but they can be avoided in several ways. The function can be linearized at this point. Another possibility is simply to disregard links with a very small pressure difference. Neither of these modifications is expected to change the result significantly. The former method has been implemented into the COMIS-code.

In each step of the iterative method a linear system of equations has to be solved. The special characteristics of the linear system of equations have been studied and the most important of these, as long as the matrix is not singular, are the symmetry and positive definiteness of the Jacobian matrix.

In the light of these studies a direct method, based on the Cholesky's method modified for band matrices, has been developed, timed, and documented. The Cholesky's method consists, in essence, of two parts. The first is the decomposition of the matrix into two triangular matrices. The second is the backward and forward substitutions of these matrices. The method does not require pivoting and only the lower triangular matrix has to be calculated during the decomposition. The method has been modified in such a way that band matrices can be handled efficiently and the band feature avoids unnecessary calculations with zero-elements. In a similar way the method can be modified using the skyline approach. The latter modification limits the work even more. If no unique solution exists, due to the modelling or round-off errors, the routine determines where the singularity is located in the matrix.

A direct method may run into calculation time problems for a poorly-structured matrix, i.e., large band width. It may be interesting to study methods which renumber the nodes in such a way that the band width decreases. If the matrix is large and has a poor structure an iterative method is probably the best choice. An iterative method was therefore originally planned to be integrated into the solver but due to convergence problems for a tested algorithm, this work has been cancelled.

The solver consists of about 10 subroutines and two steering programs. The modular approach makes it easy to exchange modules. The linear solver has its own steering program which can be exchanged entirely. The modular approach makes it also very easy to put several modules in parallel. The most efficient routine can therefore, be selected for every situation, e.g., different linear solvers.

Four basic networks have been selected for making a comparison between the different solvers. The smallest is a 2-node network and the largest is a 45-node network. All solvers have been tested regarding number of iterations and time required to reach the solution. The main result is that the optimized method requires about 40% less iteration steps than a method with extrapolated relaxation coefficients. However, finding an optimal under-relaxation factor requires twice as much work as a conventional Newton-Raphson iteration. Therefore, the total time for the two solvers is, in average, about the same. It has been decided to implement the optimized method in the COMIS program because of the fewer number of iterations required to reach the solution. This characteristic implies a more stable routine.

The linear band-solver is, in average, 30 times faster than an ordinary Gaussian solver for the most complex network used in this study. The sky-line solver is about 60% faster than the band solver. Although the time spent in the linear solver, in general, represents just a small fraction of the total CPU-time for a program, the sky-line approach has several advantages and was therefore chosen for inclusion in the COMIS-program.

10. FOLLOW UP OF COMIS

Work on the COMIS program will certainly not be finished by October 1989. A computer code will be available but such a program is ever perfectible. The validation procedure itself is a huge work and may not be completed during the COMIS year. Moreover, new knowledge will be available after 1989 which will be integrated into this program.

A roundtable discussion between the COMIS participants and the COMIS review panel about future perspectives revealed a strong feeling that COMIS (or its successor) ought to operate as an international institution with participants committing themselves to a definite work load.

It was questioned, whether COMIS should be attached to an existing IEA-Annex or be an IEA-Annex on its own. The attachment to an existing annex is critical because of the time frame. Annex XX, which would be the most likely candidate for a merge, is already in the second year of its 3 1/2 year existance. Annex V is not a task shared annex; work is performed by AIVC's staff in Britain.

Whether the Executive Committee of the IEA would support the installation of a new air flow related annex could not be answered during the discussion. AIVC's operating agent was asked at the steering group meeting in April 1989 to outline the possible interactions with COMIS at the June ExCo meeting. Following the September steering group meeting the operating agent will bring a firm proposal to the December ExCo meeting.

In order to keep the momentum alive Lawrence Berkeley Laboratory plans to continue to work on multizone infiltration modelling for the upcoming fiscal year. If there is a positive response from the ExCo, this could include taking the steps necessary to get an international working group together.

The following list outlines the tasks to be performed by a COMIS successor:

- additional work on pressure coefficients
- additional experimental studies (flow data from a structure in a controlled environment as well as parametric studies of wind pressure distribution around buildings)
- generation of occupant's behaviour
- implementation of new knowledge
- HVAC-Performance
- data bases for material, leakage characteristics, absorption, desorption
- sensitivity study to reduce input requirements
- expert system
- zone to zone ventilation effectiveness
- simulation in time
- back draft problems
- **INPUT-features:**

- language text resource for user interface
- unit translator
- subsidary modules that act on the input
- adjust building leakage to measured data
- input check on data and problem definition

- **OUTPUT-features:**

- subsidary programs at output
- dosis of occupants
- origin of air
- typical pressure distribution in time across building parts
- comparison of different runs for one building
- comparison of ventilation with demanded values in standards
- check for comfort, safety and sound of pressure differences across cracks, windows and doors

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Poster 17

AIR INFILTRATION IN CANADIAN HOMES — A DECADE OF CHANGE

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SYNOPSIS

This paper explores the results of air infiltration and ventilation research carried out in Canada over the last decade and specifically examines its application to low-rise residential buildings.

With Canada's cold climate, the reduction of space heating costs by dealing with air infiltration and ventilation issues in residential buildings is particularly important and has been the subject of government and industry initiatives. The results over the last decade have been gratifying. Not only has there been a large number of "innovative" houses built with exceptionally good control of air infiltration through the building envelope and well controlled space heating costs, but this information has been transferred to the general building industry. Current Canadian new house construction is showing significant improvement from the construction of ten years ago.

This paper outlines how this change has been brought about and how it is expected to continue.

1.0 INTRODUCTION

An examination of the evolution of house construction practices in Canada over the last decade is a study of the changes wrought by a concern over air infiltration.

The search to find methods of controlling air infiltration was driven by steep energy price increases in the 1970's. Canada's concerns were heightened by the fact that, due to our cold climate, a significant percentage of the energy used in our country is used to heat our homes. In 1973 it was estimated that:

- Approximately 20 per cent of the nation's energy was used for residential purposes[1]. and
- 77 per cent of this energy was used for space heating[2]. A large portion of this was heating infiltration air.

The control of air leakage in buildings is also particularly important in the cold Canadian climate due to a number of technical factors:

- In cold climates, the stack effect is the dominant driving force for air leakage. As a result, the highest air change rates occur during winter when the costs of heating that air are the greatest.
- In a cold climate, the exfiltration of indoor air can lead to condensation in the building envelope. Condensation can have a particularly negative impact on wood frame construction -- the typical type of construction in Canada.
- Virtually all Canadian houses have central heating with thermostatic temperature control. The most common "comfort problems" are due to drafts from air leakage.

The building industry and government agencies concerned with housing and energy reacted by stimulating research and development into reducing energy consumption for space heating. Researchers and experimental builders tackled the problems and initiated a variety of changes -- some unsuccessful, some very successful -- that have lead to significant modifications in the way houses are built in Canada. In recent years, many of the experimental techniques developed to control air leakage have been incorporated into the construction practices of tract and merchant home builders.

This has had an effect on residential energy use in Canada. In 1987, approximately 18 per cent of the total energy used was for residential purposes and, of that, space heating accounted for 68 per cent[2].

On average, energy use per household was estimated to have decreased by 32.5 per cent between 1973 and 1987. This 32.5 per cent decline in residential energy use intensity equates to a value of 50.8 GJ/household. Roughly half of the decline, 26 GJ/household, (which was approximately 15 per cent of the 1973 average residential requirement) was attributed to changes in tertiary energy demand (net of behavioral responses). About half of this was due to improvements in the old housing stock and the rest was from the greatly improved thermal efficiency of new housing stock.

Part of the improvement is due to better control of air infiltration.

A comparison of tested air change rates of the typical houses provides an indicator of the reductions in air infiltration that have occurred in the last ten years. The average air change rate for 200 houses built between 1980 and 1982 was in excess of 4.8 air changes per hour (ACH) when tested at 50 pascals pressure[3]. A recent test of merchant or tract-built houses across Canada has found that the average is now about 3.0 ACH at 50 Pa[4].

The process is not complete. Demonstration programs have shown that air tightness test results of 1.5 ACH at 50 pascals and further reductions in energy requirements are achievable. Home owners, builders, suppliers and regulatory agencies are working together to bring about continued improvements.

It is not surprising that there has been an impetus for controlling air leakage in Canadian homes. What, in retrospect, is surprising, is the extent of the change, the orderly process it has followed (considering the building science implications of the changes and the resistance to change of the building industry as a whole) and the fact that a building "philosophy" has developed. This philosophy could be stated as:

> *"Recognize the house is a system; build as tight a building envelope as practical and provide needed ventilation in a controlled, mechanical manner."*

Tracing the development of this approach and how it was accomplished is the subject of this paper.

2.0 CONTROLLING AIR INFILTRATION -- THE STATE OF AFFAIRS AT THE END OF THE 70'S

The federal and some provincial governments initiated a number of programs designed to reduce energy consumption in residential buildings in the late 1970's. These programs included:

- CHIP -- Canadian Home Insulation Program
- COSP -- Canadian Oil Substitution Program
- The Heatline, a phone-in information service for consumers
- Research activities demonstrating "energy efficient" homes

These programs did not directly deal with controlling air infiltration and tended to focus on specific components of the building system.

By the end of the 1970's, building researchers and innovative builders were experimenting with reducing air leakage in houses to control energy costs. A number of well documented projects, such as the National Research Council of Canada's Mark XI houses[5] and the Saskatchewan Conservation House[6], as well as many individual projects, illustrated that achieving air change rates of 1.5 ACH at 50 Pa was possible and that energy could be conserved without loss of comfort.

2.1 Early Problems

Unfortunately, another major lesson of the early learning process was that, if the air sealing was not done properly, serious problems could result. Problems encountered with some "energy-efficient" homes built during the late 70's included:

- Poor ventilation and high humidity levels caused because needed ventilation was not being supplied. Associated with this was the possible build up of air contaminants that had been dissipated at higher levels of air infiltration.
- Condensation related water damage inside the building envelope. Even small amounts of leakage of humid air into areas where condensation could occur can result in damage or rot to structural parts of a building. Increased insulation levels could exacerbate the problem.
- Negative pressures developing in the house as a result of exhausting appliances operating in conjunction with a tighter building envelope. This could cause combustion appliances to spill combustion gases into the house.

As a result of the early research and developmental work on air infiltration, a number of conclusions were drawn:

- There is a need to positively control ventilation rather than allow it to occur accidentally through natural forces. Natural ventilation results in too much ventilation occurring at some times and inadequate ventilation occurring at other times.
- There is a need to address all locations of air leakage, not just the total amount. Small amounts of leakage into inaccessible areas can lead to serious rotting problems.
- The location of the air and/or vapour barrier is critical. Moisture damage can be exacerbated rather than reduced if the barrier is incorrectly located. To correctly locate and install the air and/or vapour

barrier requires a sound understanding of the building science principles involved -- a knowledge not previously required of builders.

- The combustion air and venting pressure requirements of fuel-burning appliances had to be recognized.

One important outcome was a general recognition that the house operates as a system in which changes in one component can affect the functioning of other components. With building envelopes in particular, it was recognized that they were a series of barriers to heat flow, air flow and vapour flow and, while each has its own requirements, they did interact.

The early problems proved to be surmountable and the successful efforts at controlling air infiltration demonstrated that, if it was done right, a better, more energy-efficient house could be built. To encourage such construction required more research and the overcoming of two major factors: the lack of knowledge on the part of most builders of the building science issues involved and the limited availability of equipment, tools and materials to do the job correctly. In Canada, government and industry chose to work together to overcome these factors and demonstrate that well-sealed, cost-effective homes could be built on a routine basis.

3.0 GOVERNMENT AND INDUSTRY WORKING TOGETHER IN THE 1980'S

Overcoming the early problems and transferring the knowledge to the building community was not a simple, pre-defined process. It required a co-operative, integrated process of research and development, demonstration and monitoring, education and legislative initiatives -- a process that is still ongoing.

3.1 Research

Various Federal government agencies, including Energy, Mines and Resources Canada (EMR), the Canada Mortgage and Housing Corporation (CMHC), the National Research Council, Canada (NRC); the Provincial ministries of Energy and Housing; industry groups such as the Canadian Home Builders' Association (CHBA); and manufacturers of relevant products sponsored or undertook research into the mechanisms of air infiltration, how to reduce it and the impact of reducing it. Specific research included:

- studies to understand the movement of air and vapour through the building envelope[7 8 9];

- the development of techniques which provided the necessary air barrier without compromising the other functions of the building envelope. The range of techniques developed included methods which used a combined polyethylene air/vapour barrier and ones which separated the functions, such as the "Airtight Drywall Approach"[10];
- the development of products such as spun bonded oleofin membranes which act as an air barrier but not a vapour barrier;
- modelling of air infiltration in houses[11] and heat loss from houses[12];
- studies on the prevalence, impact and causes of combustion product spillage from fuel-fired appliances[13 14 15];
- the development of heat recovery ventilators and other controlled ventilation systems.

As knowledge of the appropriate techniques for sealing building envelopes became available and suitable building products came onto the market, experimental and innovative builders began promoting low-energy construction. The Federal and provincial governments established programs to encourage further developmental research, educate builders and demonstrate to the public and builders the advantages of building energy-efficient homes which included infiltration control.

3.2 Demonstration Activities

It was observed that the Canadian building industry has tremendous inertia and did not readily embrace the concept of the energy-efficient house. To encourage more energy-efficient construction, education and promotion programs were needed. One prime example is the R-2000 Program established in 1982 by the Federal Government through Energy, Mines and Resources Canada. The goal of the Program was to demonstrate that building safe, comfortable, energy efficient housing was practical and affordable.

The R-2000 Program set a standard for houses based on the performance of the house. The key elements of the performance standard were:

- The house must be built by an approved R-2000 builder who has attended a special training course offered by the Program.
- The house must meet an energy budget established through the use of a simulation program. HOTCAN, developed by the National Research Council, Canada, was the simulation program used at first. It was later modified and refined for use by the R-2000 Program and renamed HOT2000[12].
- The plans for the specific house must be examined by a certified Plans Evaluator.
- The house must have a continuously operating mechanical ventilation system[16].

- The house must be tested for air leakage when complete and have an air leakage rate of less than 1.5 ACH at 50 Pa.

The focus of this performance standard has not changed since 1982, however, the standard has not remained static -- it has evolved as more knowledge has became available. For example, the use of the HOT2000 simulation program has expanded the range of house construction techniques and equipment that can be modelled and the ventilation standard has evolved based on actual monitored data.

The R-2000 Program has incorporated a wide array of functions, from validating early assumptions to developing an extensive training program. The Program has had to deal with quality assurance issues and has undertaken research into wide areas of building science to provide the considerable technical backup required.

The Program has taken an active roll in the development of new standards for the construction industry. As well, in conjunction with other agencies and groups, the Program has undertaken a comprehensive marketing plan to raise the public's awareness of the Program and its benefits.

The Program did not function in a vacuum -- other groups and agencies worked with and complemented the efforts of the R-2000 Program. The Canadian Home Builders' Association administers the program and ventilation training courses have been developed for them by the Heating, Refrigerating and Air Conditioning Institute of Canada[17]. Valuable research and program direction has been provided by CMHC and NRC.

A number of complimentary demonstration programs were undertaken as well. The Flair Homes Demonstration Program[18 19] is one notable example involving twenty homes which were built using different techniques to achieve tight building envelopes, energy-efficient and comfortable environments with good air quality.

3.3 Monitoring and Evaluation

Confirming that energy-efficient homes, such as R-2000 homes, work as advertised was an important facet of the improvement process. An extensive monitoring program was undertaken of the first 300 R-2000 homes registered to the Program and a number of conventionally built homes[20]. A dossier is maintained for all registered R-2000 dwellings. The information gathered forms one of the best data bases available on the performance of low-energy homes. Many other organizations have undertaken monitoring activities, as well.

The monitoring has contributed valuable feedback on the products and techniques used to build low-energy homes. The early manufactured heat recovery ventilators (HRVs) for example, had problems that were, in part, detected through monitoring the early R-2000 homes. It turns out that a major factor in HRV failures was inconsistency in the installation of the HRVs. The outcome has been the introduction of the HRV Installers' Course[17] by the HRAI with a resulting significant improvement in their field operation.

Through the monitoring programs and the ongoing analysis undertaken by various research groups, a number of concerns were raised and addressed.

One concern was the impact of well-sealed dwellings on the operation of combustion appliances. With the well-sealed house, it was suggested that, under certain conditions, combustion gases could spill into the house rather than go up the chimney due to the depressurization of the house. Simulation models have been developed by CMHC[13] and a number of field studies were conducted[14]. It has been found that the incidence of actual venting failure is low but significant, particularly when natural venting appliances are used in well-sealed houses. To avoid this problem, the current gas code requires the provision of outside combustion air for gas burning appliances.[21] Naturally aspirating appliances are not allowed under the R-2000 Program.

As energy consumption values and air quality statistics became available in the mid 1980's, the R-2000 Program, along with other interested parties, (for example, the Ontario Ministry of Energy, manufacturers and electrical utilities) began to actively promote R-2000 concepts to builders and the general public.

3.4 Education and Training

By the mid 1980's it was obvious that a good "product" (a variety of mechanisms and building techniques) had been developed to build tight, energy-efficient houses. However, there was a need to get the information on that product out to builders and establish a market with the general public. Again, government agencies took the lead in co-operation with industry.

In 1985, to more heavily involve builders and increase their awareness of the Program, the mandate for managing the implementation of the R-2000 Program was transferred to the Canadian Home Builders' Association from Energy, Mines and Resources Canada. A series of nation-wide Builder Workshops was instituted.

To date, this course has been attended by more than 12,000 people representing all facets of the building industry. The two or three day training course is important. The topics covered include the appropriate and best techniques for air/vapour sealing and the concept of the controlled supply of air through the use of mechanical ventilation equipment. By enforcing a high level of competence on the part of the builder, many of the earlier problems encountered with the construction of well-sealed houses could be avoided -- along with any negative publicity.

Energy, Mines and Resources Canada also developed courses for renovators[22] and for instruction in community colleges and trade schools.

The Canada Mortgage and Housing Corporation also sponsored training courses for builders of new buildings and for renovation contractors[23 24] They also publish a great number of publications on good construction practice. These are targeted at both builders and homeowners.

The Department of Building Research (DBR) of the National Research Council, Canada was also active. DBR sponsored a series of seminars called "Building Science Insight" and provided a great deal of information in the form of research reports and other various series-type publications.

As well, industry was encouraged and financially supported to offer training and education programs. The Heating, Refrigerating and Air Conditioning Institute of Canada, for example, conducted courses to train the correct procedures to installers for installing and commissioning HRVs[17].

Educating the builder about the features of the energy-efficient, well-sealed house was not a simple process due to the fragmentation and inertia of the building industry. An integral part of the process was to educate the consumer -- the home-buying public -- to request an energy-efficient, well-sealed product.

Advertising campaigns sponsored by the R-2000 Program, home builders, product manufacturers, provincial agencies and utilities promoted the concept of the energy-efficient, well-sealed house. The provincial home builders' associations presented annual home tours. Energy, Mines and Resources Canada, in conjunction with TV Ontario, produced a series of programs entitled "R-2000 The Better Built House". Newspaper and magazine articles documented the merits of living in a well-sealed house and R-2000 homeowners recommended R-2000 houses to their friends.

The proof of the success of the process has been in the demand by home purchasers for the concepts promoted through the advertising campaigns. The current Canadian home buyer is much more conscious of the levels of insulation used in construction, how well sealed the house is and the quality of the windows and doors.

Further proof of the success of the education and training campaign is the broad range of equipment and products available to control air leakage and, therefore, energy costs, and to provide needed ventilation. Equipment, such as prepackaged HRVs and high efficiency furnaces, and materials such as spun bonded oleofin membrane air barriers, gasketing, electrical box enclosures and other assorted products for both polyethylene and air tight drywall approaches (ADA) to air sealing are all readily available across Canada.

3.5 Codes and Standards

As the demand for a well-sealed, energy-efficient housing product has increased, so has the concern that the public be provided with safe habitation. Air and vapour barriers must be installed correctly and in the appropriate location, combustion appliances must receive adequate supplies of outside air and mechanical ventilation equipment must be installed correctly.

In a number of instances, the requirements and standards developed and instituted for projects such as the R-2000 Program have been, or are in the process of being, incorporated into national and provincial building codes and standards. In a sense, the voluntary standards of these programs provided a breeding ground for producing national standards and regulations for the building industry as a whole.

Some of the most significant codes and standards developed over the last decade that relate to the control of air leakage through the building envelope include:

CAN/CGSB-149.10-M86
Determination of the Airtightness of Building Envelopes by the Fan Depressurization Method
Canadian General Standards Board
This standard details the procedure for using a fan to determine the equivalent leakage area of the building envelope through the use of a calibrated fan test apparatus.

CAN3-A440-M84
Windows - Building Materials and Products
This standard defines the performance of windows, regardless of the material of construction. It addresses performance tests and outlines the standards for air leakage, water leakage and wind load resistance.

CAN/CSA-C444-M87
Installation Requirements for Heat Recovery Ventilators
Canadian Standards Association
This standard applies to the installation requirements for self-contained ducted heat recovery ventilators for new and existing buildings. It applies to equipment selection, minimum installation requirements and the information to be provided to the purchaser.

CAN/CSA-C439-M88
Standard Methods of Test for Rating the Performance of Heat Recovery Ventilators
Canadian Standards Association

This standard documents a procedure developed for testing and rating the ventilation and energy performance of heat recovery ventilators under standard conditions including a cold weather test (to -25°C and, more recently, to -40°C). The most important output of this standard is the independently produced design information for designers and builders.

CAN/CGSB-51.34-M86
Vapour Barrier, Polyethylene Sheet for Use in Building Construction
Canadian General Standards Board

A revision of the previous standard outlining the minimum criteria polyethylene sheet must meet to be acceptable as a vapour barrier. It now incorporates provisions for resistance to thermal decay.

CAN/CGA-B149.1-M86
Natural Gas Installation Code
National Standard of Canada

This standard contains a specific requirement for the provision of combustion air to gas burning appliances. The standard specifies the free area of the outdoor air supply required according to the rating of the total appliances in the structure or enclosure.

CSA Preliminary Standard F326.1-M1989
Residential Mechanical Ventilation Requirements
Canadian Standards Association

This preliminary standard defines the requirements for mechanical ventilation systems whose purpose is to provide the minimum controlled rates of ventilation air to the habitable spaces of self-contained single-family dwellings. Research and findings from the R-2000 Program provided important background to the development of the standard.

National Building Code of Canada
(the model for provincial codes)
Associate Committee on the National Building Code, National Research Council, Canada

The 1985 version of this code required that the vapour barrier be sealed. This, in effect, made it as air barrier, as well.

Three proposed revisions to the 1990 version of the code will have an impact on the control of air leakage through the building envelope. These include:

- Revised wording which recognizes the different functional requirements of the air and vapour barrier systems in the building envelope and requires a continuous air barrier.

- A requirement that a ventilation system capable of operating at an average continuous rate of 0.3 air changes per hour be installed in houses.
- Radon protection measures which include the sealing of all below grade entry points, a membrane under the floor slab and a gravel layer under the slab.

4.0 THE FUTURE

Although the various segments of the Canadian building industry have made a great deal of progress over the last decade in understanding and controlling unwanted air leakage in buildings, significant further advances are expected. Demonstration programs have proven that air tightness levels of 1.5 ACH at 50 Pa and space heating energy savings of 50 to 75 per cent after 1980 levels are achievable. As many of the developed concepts and techniques are incorporated into existing building codes and standards, all new housing will achieve the benefits of air leakage control.

Already the difference between regular, tract-built housing and energy-efficient homes, such as the R-2000 homes, is diminishing. It is not unusual to find tract built houses which meet R-2000 air tightness standards. It is logical to project that the average difference will continue to diminish to the point where it will be insignificant.

The changes in new building stock have already made a significant improvement in Canada's energy future. There is still room, however, for future developments in controlling air leakage, particularly in dealing with the existing housing stock. The lessons that have been learned about the importance of controlling air leakage and how best to do it can equally be applied to existing houses with some modifications. This process has already started in a significant fashion. Other concepts are also being explored including the "dynamic wall" approach[25] in which air leakage is not eliminated but rather controlled and used to advantage.

Continued development is expected in improved, cost-effective ventilation systems which meet the needs of consumers and Canada's harsh climate.

Probably the most important element in this transformation is the fact that a large and growing pool of engineers, architects, builders and trades contractors have been part of the process and have received the benefits of the improved knowledge and training in building energy-efficient houses. These people are ultimately the group which will have the greatest impact on the future of the Canadian housing industry.

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PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

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Poster 18

VENTILATION, AIR FLOWS IN BUILDINGS AND INDOOR AIR QUALITY - RD&D AND DEVELOPMENTS IN GERMANY

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INTRODUCTION

Ventilation, infiltration, indoor air flows and air exchanges determine two aspects of fast growing interest: the energy balance of buildings and the indoor environment. Whilst in the wages of the energy crisis RD (research and development) & D (demonstration) in the different areas had been focused on rational energy use now with view on the public awareness of the environmental situation (outdoors and indoors) also the aspect of indoor air quality stimulates widespread RD&D activities. The developments in the near future will be influenced by the demand to bring energy conservation into line with acceptable indoor air quality. This is expressed in several main RD&D programmes supported by Federal Ministries as well as in other institutional or industrial activities.

The areas of ventilation, infiltration, indoor air flows and air exchanges cover a wide range of technical subjects, first of all heating and air conditioning. An overview over the complex structure and organization of RD and D in the Federal Republic of Germany is given in a report by L. Trepte and A. Le Marié[1]. This paper presented at the 10. AIVC Conference as poster will summarize some new developments and thus complement the report.

1. FEDERAL RD&D PROGRAMMES

Most important for areas of energy conservation and indoor air quality are programmes and projects supported by the Federal Ministry for Research and Technology (Bundesministerium für Forschung und Technologie, BMFT) and the Federal Ministry for Youth, Family, Women and Health (Bundesministerium für Jugend, Familie, Frauen und Gesundheit, BMJFFG). As a matter of fact RD&D have priority in programmes of BMFT. The BMFT understands its role as an initiator and a catalyst to accelerate the introduction of RD&D results into practical application.

On behalf of the BMFT the Project Management for Biology, Ecology and Energy (PBE) in KFA Jülich GmbH coordinates the programme "Energy Research and Energy Technologies". This programme is structured by different focal points, from which in this connection "Rational Use of Energy and Solar Energy for Buildings" is of main concern, with the following activities:

- passive and active solar systems
- heat pumps and heat pump application
- improvement of insulation technics
- air exchange, ventilation, hot- air heating, heat recovery
- improvement of conventional heating systems
- energy conservation in buildings: consulting, information, systems.

An overview over the supported projects, as well as the principal researchers, duration, fundings etc., is given in a report edited annually by PBE[2].

Whereas the above mentioned programme is focused on energy aspects another one, also supported by the BMFT, is started 1987 and aimed at indoor air quality. With the working title "Indoor Air Pollutants" it is structured in the following way:

- pollution sources and indoor air quality measurement techniques
- connection between energy conservation and indoor air quality
- model simulation and development of models
- risk assessment.

In addition to these programmes and supporting activities also other Federal Ministries, e.g. the Ministry for Regional Planning, Buildings and Urban Development (Bundesministerium für Raumordnung, Bauwesen und Städtebau, BMBau), or Ministries of the states initiate and support projects in these areas.

2. UNIVERSITIES, FEDERAL AND OTHER RD&D INSTITUTIONS

In the system of German RD&D activities, especially for basic research but to some extent also for contract research and development several types of other institutes and institutions are incorporated. Independent of ongoing programmes they contribute essentially to progress in the areas of energy conservation and indoor air quality. These are e.g.

- universities, technical universities and incorporated institutes, examples:
    - University of Essen, Institute for Thermodynamics and Air Conditioning
    - Technical University of Berlin, Hermann-Rietschel-Institute
    - RWTH Aachen, Institute for Heat Transfer and Air Conditioning Techniques
- Max-Planck-Gesellschaft (MPG) with about 60 institutes, research above all in natural science and medicine, cooperating with universities, examples:

  Max-Planck-Institut für Physiologische und Klinische Forschung, Bad Nauheim, Max-Planck-Institut für Strömungsforschung, Göttingen

- National Research Centres (NRC) with twelve major research centres with some 16 000 employees, including 8 000 scientists and engineers. They conduct scientific, technological, biological as well as medical research. They cooperate closely with universities and many of the scientists working at the National Research Centres are also on the staff of universities located in the same state. Examples: Jülich Nuclear Research Centre (KFA Kernforschungsanlage Jülich GmbH), Jülich, Karlsruhe Nuclear Research Centre (KfK), Karlsruhe

- Fraunhofer-Gesellschaft (FhG) for the promotion of applied research with some 30 institutes

- Federal Institutions, subordinate institutions of the federal ministries. Carrying out research and tests they play an important role e.g. for formulating regulations, standards etc..
  Examples: Federal Health Office with Institute for Water, Soil and Air Hygiene, Berlin, Physical-Technical Federal Office, Braunschweig

- Other research institutions beyond universities, but carrying out research and development on a non-profit base. They usually have connections to universities and to some extent the staff of these institutions is also on the staff on an university.
  Example: Institute for Window Engineering (Institut für Fenstertechnik), Rosenheim.

3. STANDARDIZATION, INDUSTRIAL RD&D ACTIVITIES

In the last years in standards and guidelines and in industrial RD&D activities mainly energy aspects were placed into the foreground, among others as a result of the Regulation for Heating Insulation (Wärmeschutzverordnung) based on the Energy Conservation Law (Energieeinsparungsgesetz). In connection with new developments and a better understanding of indoor air quality requirements some of the major standards and guidelines will be revised. Examples are:

- DIN 1946 T2 "Room Ventilation, Technical Health Regulations", in preparation

- DIN 1946 T6 "Ventilation of Dwellings", draft

- VDI 3816 "Ventilation in the Case of Outdoor Air Pollution", draft.

The approaches include aspects of definition of indoor air quality, new definition of ventilation requirements, optimization of air flows and ventilation efficiency.

For joint research and technical associations because of the multiplicity is only referred to the following two examples:

- FLT Forschungsvereinigung für Luft- und Trocknungstechnik in cooperation with AIF Arbeitsgemeinschaft Industrieller Forschungseinrichtungen, RD&D interest at present: assessment of air flows, ventilation efficiency, legionalla and allergenes in HVAC systems, aspects of comfort

- FGK Fachinstitut Gebäude-Klima (PR-Association), main activities: information services concerning air conditioning, health, comfort and rational use of energy in buildings, support of cooperations, initiation of RD&D.

The industrial RD&D activities in the Federal Republik of Germany are very widespread and can be summarized in the following way: Further and advanced development of compact and maintenance free HVAC systems and components, development of demand controlled ventilating systems, heating-ventilating-systems, indoor air quality: measuring as well as controlling by filtering etc..

4. DEVELOPMENTS

Also if window ventilation still dominates in residential buildings in Germany, the growing awareness for health aspects and energy conservation will stimulate RD&D. In office and industrial buildings already more advanced HVAC systems are in practical use and it can be assumed that some output of this know-how and increasing benefit-cost-relations will also influence the residential buildings market positively. Looking in the near future the following RD&D efforts have to be undertaken (examples):

- Research and development fields:

  - air flows: modelling, measurements, quantification of ventilation efficiency, energetic optimization of air flows
  - sensors for indoor air quality
  - assessment of indoor air quality, health effects

- Practical application:

  - development of appropriate sensors and control units for indoor air quality (moisture, carbon dioxide etc.)
  - technics for air treatment (pollutant removal)
  - boyancy assisted ventilation
  - demand controlled ventilating systems, basic ventilation, demand ventilation, efficient ventilation
  - air heating systems, heat recovery
  - intelligent buildings, integrated building techniques, which link indoor environmental control, energy supply including lighting, security, communication and automation.

These are the subjects in present and starting German RD&D projects. Tomorrows technic for ventilating and heating will depend on the acceptance of the results by the market.

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PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

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Poster 19

AN OVERVIEW OF INFILTRATION AND VENTILATION DEVELOPMENTS IN FRANCE

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AN OVERVIEW OF INFILTRATION AND VENTILATION DEVELOPMENTS IN FRANCE

Synopsis

France is one of the European countries where ventilation has the most advanced regulation.

Vertical ducts have been used for a very long time, making easier the transfer from passive to mechanical ventilation (which covers now 90 % of the blocs of flats and 70 % of the individual housing).

1969 regulation has been based on a continuous air exhaust from the service rooms and air replacement through inlets in the habitable rooms. This has never been changed since then and is a part of french regulation's features.

The other main characteristic is to include the thermal loss du to ventilation in the calculation of the total loss of the housing.

Since 1983, it is possible to install a system which modulates and distributes the ventilation flows according to the needs (measured from humidity level) in the different rooms.

When you buy a house, you find heating, water supply, electricity supply. Good quality air supply, through a global ventilation system, must be provided as well. This is why a regulation is needed, with an obligation of result.

Essential requirements have to be met in every house, every places; whatever the wheater, the occupancy may be, at the right energy cost.

1. HISTORICAL INTRODUCTION

The present state of ventilation in France is the result of a long sequence of habits and requirements.
In order to understand it, one should go back as far as the after-war reconstruction period.
In the early 50's, one searched for higher and higher buildings because they were less expensive and quicker to build, making possible to house as many people as possible within the shortest time.
At that time, a paragraph of the building regulation expressed that all dwellings should have a possibility of heating accomodation.

As the central heating was not as wide spread as it is now, that particular point of the requirement often turned to be a mere duct for fumes ; each dwelling had its own duct connected with the roof level.

Due to that requirement and its interpretation, the contractor considered a vertical duct for air transport, joining each flat to the roof as a standard.

Once that practice was adopted, it remained valid even later, when central heating proved reliable.

2. THE SHUNT

In 1955 a new type of duct appeared:the "shunt". It was in fact a double duct composed of a smaller one used for the individual flow of each dwelling and a bigger one as a collector to gather the individual air flows into one (see fig 1).

In that way, the shunt duct made it possible to gather more than one dwelling in one plant saving space (for instance a 10 storey building needs only 20 x 54 cm instead of 15 x 150 cm in the previous system) and decreasing the cost of construction. The ventilation was still limited to the people's opening the windows. Occasionnaly, some grilles in lower and higher position were fitted in the service rooms like kitchen, bathroom..., using passive ventilation.

3. THE REVERSED SHUNT

At that time a new tendency appeared i.e. to group the service rooms in the middle of the buildings, leaving its "nobles" outside space to the habitable rooms.

Such a design allowed to use both sides of a building. The existing shunt duct made building construction easier by permitting the ventilation of the service rooms (where the pollutions due to occupants are stronger) although they had no longer any common wall with the outside atmosphere.

The ventilation was then ensured by a double shunt system :
It associated a normal shunt for the exhaust of foul air from the service rooms to the roof through higher grilles and a reversed shunt for the fresh replacement air, coming from the ground floor through lower grilles (See fig 2).

As foul air replaced fumes in those ducts, that design enabled to collect seven flows instead of five. That ventilation was still passive : "power" is given by the combined actions of wind and thermal draught (stack effect).

Each service room was ventilated reparately, being connected with a double shunt shaft.

There are two main objections to this principle :

- with the wind falling, the air flow may be strongly reduced ; even if a static fan may give an illusion of efficiency.

- the "motor" of the stack effect is the difference between the inside and the outside air temperature : It races in winter and stoppes in summer.

Thus, all dwellings are over ventilated in winter, causing unpleasant cold draughts, and under ventilated in summer, leaving bad odours, moisture in place.

- The habitable rooms are not ventilated in a coherent way, only submitted to the cross ventilation which can be either in the right or in the wrong direction spreading various pollutants from the service rooms. At that time,however, in France, the air leakages were rather high and there were only little condensation problems.

4. 1969 - THE FIRST REGULATION

4.1 1969 Regulation

The 1969 regulation,(the first required residentially in France), was designed to overcome all the drawbacks we talked about previously. The ventilation was required to continuously provide one airchange per hour in each habitable room. It had to supply fresh air to the habitable rooms and to exhaust stale air from the service rooms (see fig 3).

Actually, that was the very begining of the central mechanical ventilation in France.
This technique had already been know and used in switzerland and in sweden for a little while.

Its advantages are significant :
Whatever the weather, wind, temperature, may be the flows remain constant and well known in each service room.

In 1969 regulation - as in the further ones - the passive ventilation was still allowed, requirements were then to be met on an average winterday.

4.2 Evolutions

It became necessary to controll that the requirements were fulfilled either with a technique or with another.

Several control organisms (CEP - SOCOTEC ...) were set up. In a short time, they found that the mechanical ventilation was the best solution to comply with the requirements. It was easier to control, more constant and closer to the requested rates.

The existence of a regulation made possible a great extension of the ventilation market and stimulated the development of many new products, new technologies for the exhaust or inlet valves.

The quality of the different components of the complete system improved : the inlets took diffusion into account, tried to fight cross ventilation and wind pressure effects. The outlets allowed the flows to remain fixed, independant of the pressure in the ducts, without any adjustment on side , allowing to install extended networks with one bigger fan on the roof. The noise of the elements was strongly reduced and new requirements appeared, in order to limit sound levels. Fans became more and more reliable, and their lifetime increased, reaching 10 to 15 years of continuous work.

Additionally, it was less expensive to use central mechanical ventilation rather than passive because ducts were so much smaller (which was facilitated by the increase of the air velocity in those ducts) thus saving building space.

Furthermore, the " code de la construction " which is - in France - the practicable interpretation of essential requirements, asked the ventilation to cope with condensation problems : " condensations must not occur but momentarily."
Some cases were brought before the courts, where the dwellings had become insalubrious within a few month after the first occupancy.

After a mechanical ventilation had been set, the court noticed that condensation had disappeared and condemned the builder to refurbish the house and to use a mechanical ventilation instead of a passive one.

So, in the early 70's the market of the control mechanical ventilation was growing up at a rate of % per year .
Assuming this growth to continue for a while, and believing that, when writen, a requirement becomes effective at once, the french authorities have decided to include the energy loss due to ventilation in the total energy costs of the buildings.
Although the requirements were not met in every house, the energy costs estimates turned out to be good enough, because of the high ratio of mechanical ventilation.
The consumed energy is then calculated by the formula :

$$E = \quad (Ki\ Si) + 0.34\ Q$$

where : E is the total loss in W/°C

Ki are the thermal coefficients of walls in W/m2/°C
Si are the areas of walls in m2
Q is the total flow of exhaust air in m3/h.

This point of the french regulation, which has continuously been used since then, is very speciic and very important :
It makes possible to compare every loss whatever the sources are, hence allowing to choose either a saving energy ventilation, or a normal with a larger thickness of insulator.

5. 1974 - THE ENERGY CRISIS

1974 is the year of the first energy crisis and attention was focused on the heating bills.
At that Time, about one third of this bill was caused by ventilation.
The dropping of the continuous ventilation was considered as a possible energy saving measure.

5.1 1978 - The Survey

In 1978, the french authorities decided to undertake a general survey about ventilation, collecting datas from all parts of the country, in order to determine if it was possible to reduce the air-change rates, to allow intermittent use...
The considerable amount of datas which was collected permitted to know, very precisely, how the different types of houses were occupied (see table 1)
It made possible to link the number of habitable rooms with occupancy, then gave a percent of residences which were over, well or under ventilated (see table (2)).

A significant result of the survey was to find out that almost 50 % of dwellings were over ventilated (when basing on occupancy rather than on volume).

5.2 The consequence of the Survey - 1982 Regulation

5.2-1 The consequence of the survey

The main consequence of the survey has been to change the reference for ventilation rates : they have, since then, been linked to the number of habitable rooms.
It has been decided also to lower the airchange rates in order to minimise energy loss.
New labels were set up, corresponding to different class of energy consumption thus to differents amounts of money given as incentives
to build houses as thrifty as possible.

Some competitions and long time monitorings were set up to find new solutions and to make everyone involved in the energy saving.

5.2-2 1982 - Regulation

The 1982 regulation was in fact decided in 1980 but the requirements were to be effective in 1982, in order to give the industry sufficient time to develop products according to the new rates.
These rates have been calculated as follows since then :

$$Q = 15 (N + 2)$$

Where Q is the total exhaust flow in m3/h
N is the number of habitable rooms.

Additionally, a higher level for kitchen has been proposed, depending on the size of each kitchen.
This additional flow had to be intermittently operated by the owner to cope with the problems of odours during cooking for instance.

5.3 Consequences

A consequence of this regulation, which is often forgotten, has been to reduce the air changes from one per hour to about 0.75 to 0.65 .....
While ventilation was changing, the thermal insulation improved, the double glasing became more and more used and the air leakage lower
We would see that all these changes put together have brought some desagreements :
The improvement of thermal insulation made the influence of thermal bridges more effective : All the cold has been located on little areas at the breakpoints of insulators.
Thus, condensation occured always in the same zones.

The glazing was previously playing the role of a large cold area where almost all water vapour condensed. The double glazing removed this possibility.

The increasing airtighness decreased the cross ventilation thus the total airchange became lower and lower, which means the humidity became higher and higher .
We can set the equation :

Higher humidity
\+ Removal of cold areas
\+ Thermal bridges at the insulation breaklines
\+ Decreasing airchanges
= More condensation problems, more mould on the walls.

It semms obvious that the ventilation needs are strongly connected with the occupancy. In 1969 requirements the needs of large families were not met. The 1982 regulation increased the number of such families.
As shown in table 2 we can see that the link between the number of habitable rooms and the occupancy is not accurate enough to be a sufficient criteria for ventilation rates.

5.4 1983 Alternative

To avoid this increasing risk of condensation, a research was undertaken, trying to find out a solution which would be more accurately linked to the real needs rather than to the average needs.

At that time, the pollutant better connected with occupancy was found to be the water vapour.It is strongly related to human activity : Breathing, cooking, washing... Each of these activities gives off some vapour, with a specific quantity which can be averaged.
That enabled to build some possible scenarios and to prove, before manufacturing any product, that it was possible to save energy while adapting ventilation closely to the needs.

All this research was presented to the french authorities and the regulation has been amended in 1983 to authorise the new system, allowing to reduce the ventilation rates during under-occupancy periods and re-establish normal ones when necessary.

At that time, the exhaust flows have been chosen less or equal to the requirements' level. Which means that the new system left under ventilated the dwelling which previously were. In such cases, the improvement was due to the inlets, ensuring a better distribution of air. (The higher the humidity, the larger the apperture
(see fig. 4).
We can try to explain why the flows were set lower than requirements' ones : the amount of investments required to develop this technology was quite important ; thus the new products were rather expensive and it was a necessity to sell maximum energy savings. In the amendment to the regulation, that saving was taken into account in the general heat loss of the house, thus giving points to get financial incentive for building.

That was only a first step to meet the real needs; now, some systems use higher flows, remaining connected with the humidity. They can fit more precisely whith the needs, even if the premises are over occupied (according to french average occupancy).
The modulation allows the exhaust flow to be as low as about 45m3/h (for a two bathroom dwelling) and as high as 175 m3/h continuously when necessary (without accounting for the additional flows in kitchen or W.C)
Obviously, we must not forget the inlets which modulation allows to admit air where it is the most necessary.
Inlets and outlets have the same proportional action in the range of 35% to 70% Hr.

CONCLUSION

As a conclusion and to make the french approach as clear as possible, let us have a glance ont its basements :

1) The occupants, while breathing, cooking..., alter their indoor air quality, thus ventilation is necessary.

2) The occupants are not able to appreciate the quality of the air they breath neither the ventilation rate which would be needed to make it better.

3) Identical dwellings have not the same needs. These needs change during the day itself.

4) Ventilation has an energy cost.

5) A dwelling which can be occupied by N Persons must be able to provide them a good air quality without causing any nuisance for them and their neighbours (such as odours, moisture, noise...) and for the right energy cost.

6) The ventilation system is a part of the housing, as well as heating, water supply..., and all means must be set up to ensure its proper use. The obligation of result should be a requirement instead of an obligation of means.

7) To ensure possibility for evolutions and clear competition, an evaluation method upon the efficiency of the systems should be workdown. Such a method is now under consideration in France.

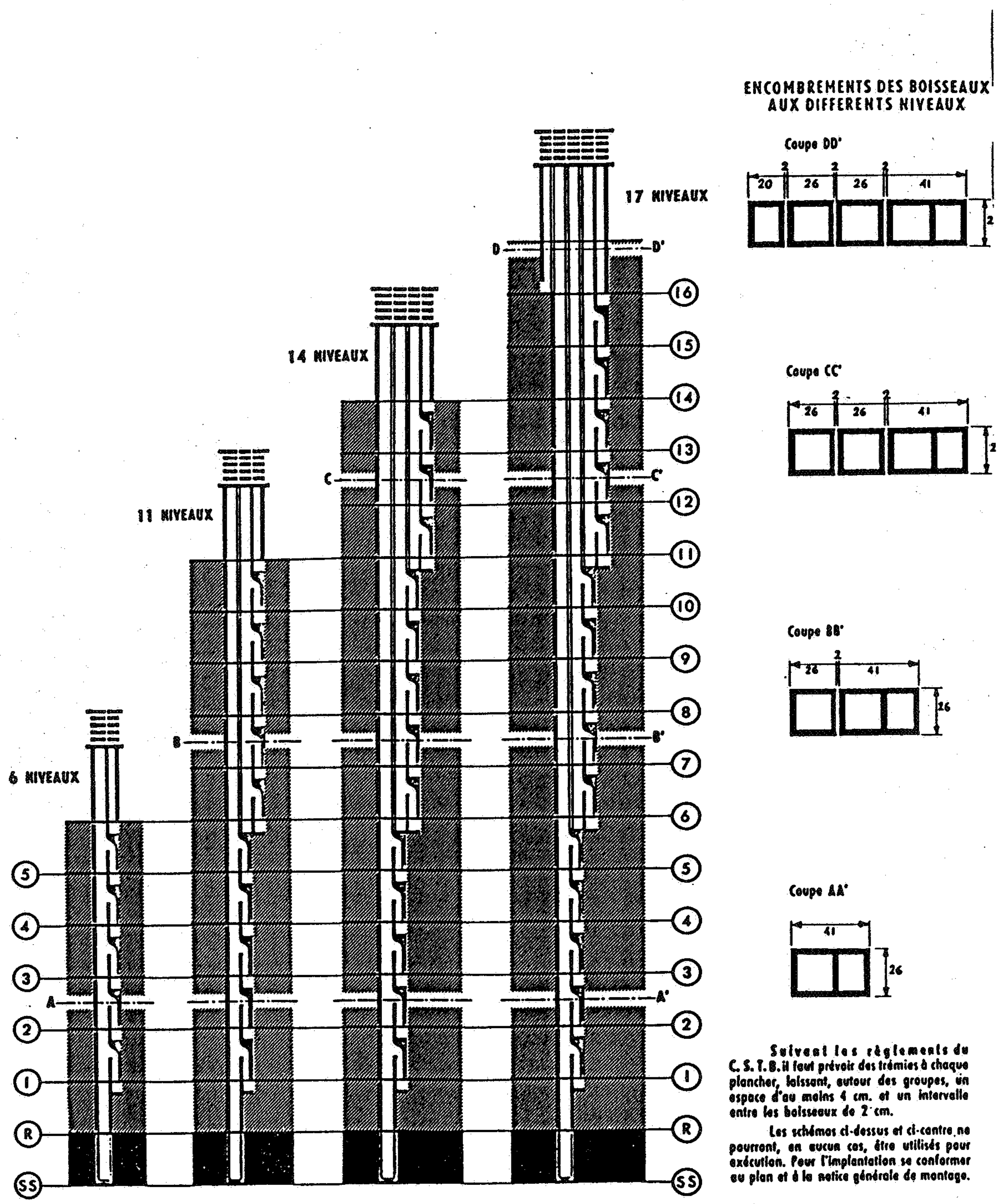


TABLEAU DES APPELLATIONS DIMENSIONNELLES DES BOISSEAUX U.M.G. POUR FUMÉE

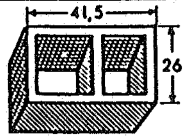
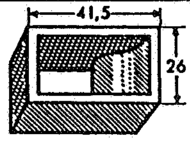
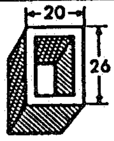
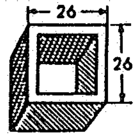
ÉLÉMENT COURANT B 41 h = 33	ÉLÉMENT DE RACCORDEMENT BR 41 h = 25	ÉLÉMENTS INDIVIDUELS B 14 h = 33	B 20 h = 33
 	 	 	

FIG 1 : SHUNT DUCT

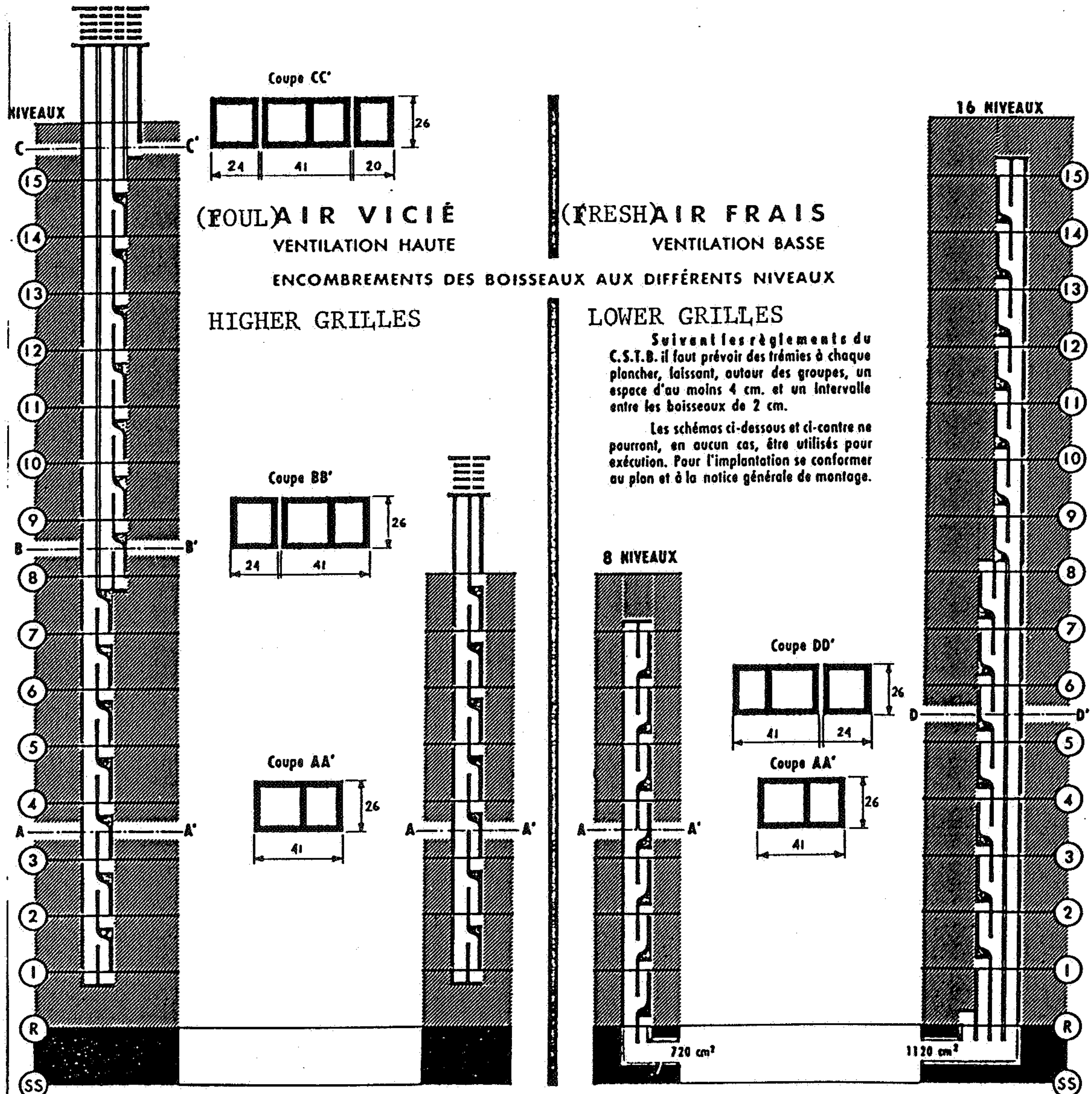


TABLEAU DES APPELLATIONS DIMENSIONNELLES DES BOISSEAUX U.M.G. POUR VENTILATION

ÉLÉMENT COURANT V 41 h = 33	ÉLÉMENT SPÉCIAL VD 41 h = 33	ÉLÉMENT DE RACCORDEMENT VR 41 h = 25	ÉLÉMENTS INDIVIDUELS V 16 h = 33	V 20 h = 33

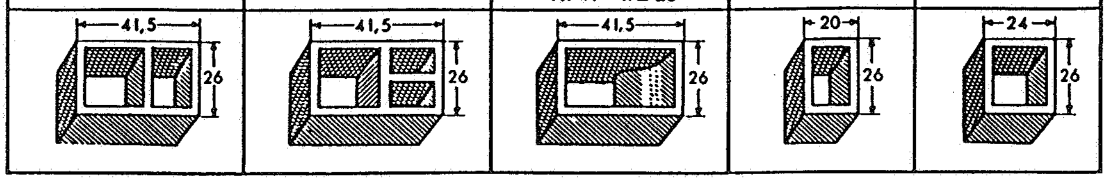


FIG 2 : REVERSE SHUNT SYSTEM

Table 1. 1978 SURVEY IN FRANCE: OCCUPANCY ACCORDING TO THE NUMBER OF PRINCIPAL ROOMS IN THE HOME.

OCCUPANTS \ PRINCIPAL ROOMS	1	2	3	4	5	6	TOTAL*
1	933	1304	1028	550	212	134	4162
2	278	1067	1727	1356	618	404	5453
3	64	331	1144	1120	613	344	3617
4	18	118	561	1069	687	486	2939
5	4	29	160	498	387	336	1415
6+	3	17	67	251	331	383	1053
TOTAL*	1300	2867	4690	4845	2848	2089	18641

* TOTALS are in thousands, and have been rounded off.

_ TABLE 1 _

Table 2. 1978 SURVEY IN FRANCE: FRESH AIR PER OCCUPANT IN m3/hour BASED ON 1969 REGULATION FOR CONSTANT FLOW VENTILATION

OCCUPANTS \ PRINCIPAL ROOMS	1	2	3	4	5	6
1	45	60	75	90	105	120
2	22	30	37	45	52	60
3	15	20	25	30	35	40
4	11	15	19	22	26	30
5	9	12	15	18	21	24
6	7	10	12	15	17	20

TOO MUCH AIR: MORE THAN 30m3/h PER OCCUPANT (49%)

CORRECT AIR: MORE THAN 20m3/h PER OCCUPANT, and LESS THAN 30m3/h PER OCCUPANT (39%)

TOO LITTLE AIR: LESS THAN 20m3/h PER OCCUPANT (11%)

_ TABLE 2 _

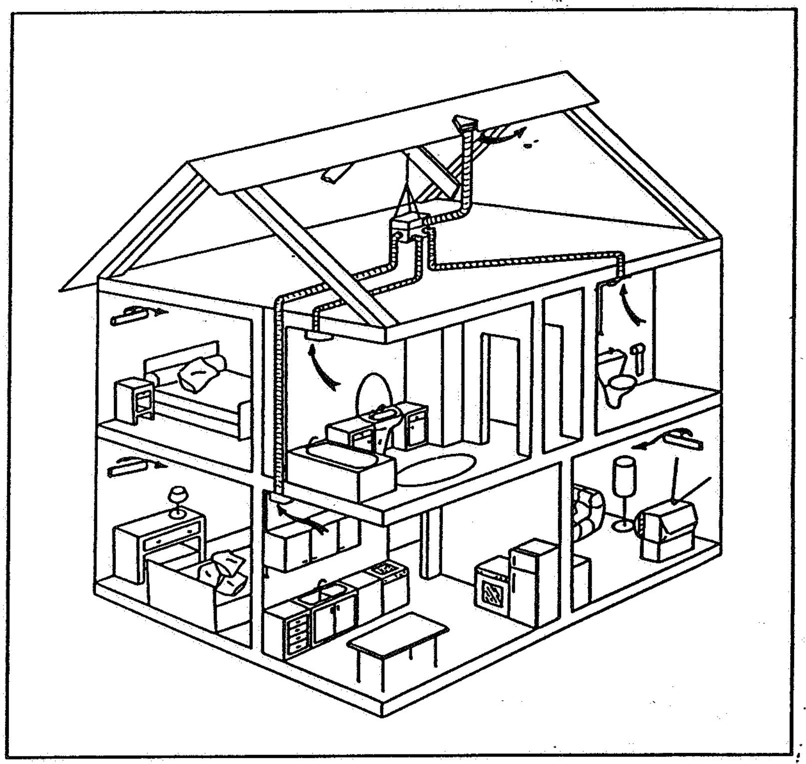
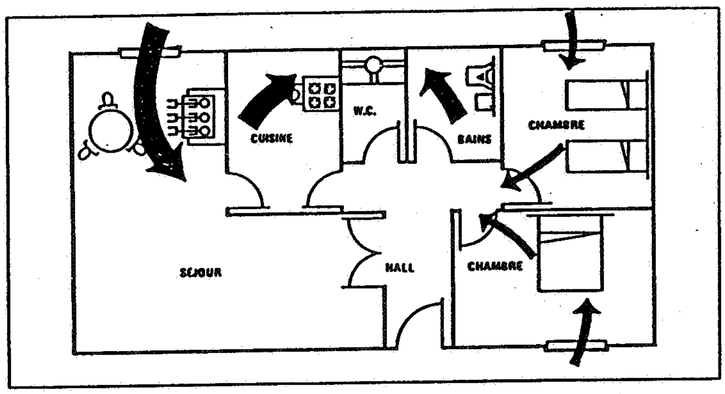
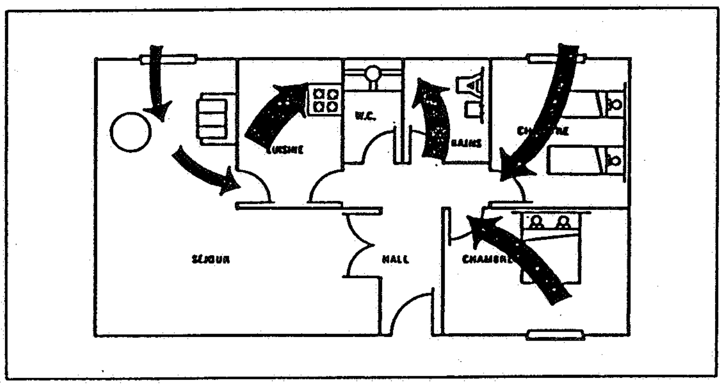


FIG 3 / AIR FLOWS THROUGH THE HOUSE

DAY OCCUPANCY



NIGHT OCCUPANCY



FYG 4 : HYGRO MODULATION

PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

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Poster 20

Automated Tracer Equipment for Air Flow Studies in Buildings

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1 Synopsis

This paper describes tracer gas methods and equipment developed to measure infiltration and inter-zone air flow rates in New Zealand houses. Air flows in houses have been studied in detail, in order to understand the role of ventilation in controlling indoor moisture, and the role of air flows through the construction cavities in transferring moisture to parts of the structure most sensitive to moisture. The main technical content of this paper, however, concerns an automated tracer gas detection and delivery system based around a gas chromatograph and an electron capture detector. Constant composition and decay modes of operation have been automated and the control loops for both modes are illustrated with a flow diagram. Favourable comparisons between air flows measured and calculated by various methods are presented. Against this background of consistency, a passive technique is shown to give plausible infiltration rates.

2 Tracer Techniques

Tracer dilution methods have been widely used to study air flows in buildings, but with manual control and data analysis methods, it can be time consuming and impractical to monitor air flows over periods of days and weeks. Automation with computers and process control technology has been employed by the Building Research Association of New Zealand (BRANZ) to make long term airflow studies practical.

The tracer gas and the detector type have an important bearing on concentration analysis and control procedures. In this case a gas chromatograph coupled to an electron capture detector (ECD) was used to detect halogenated compounds used as tracers. One advantage of an ECD is that ideal working concentrations of tracer are very dilute (parts per billion) so that transporting and applying the tracer gas are logistically easy and there are no potential air quality problems when working in occupied spaces. One disadvantage is that air samples must be analysed discretely rather than continuously and the consequent time delay can be a problem when controlling a tracer concentration to a constant level.

A computer running Basic has been used as a process controller and data analyser. This made it easy to adapt the equipment to a range of different jobs that have contributed to the air infiltration program in recent years. Early measurements of house-averaged infiltration rates[1] were made using a single tracer gas, SF_6, and the decay method with automated tracer top-up. As more sophisticated multizone air tracing tasks have emerged, the equipment has been reconfigured to analyse multiple tracers and control tracer concentrations to target constant values. Recent studies of airflows between the construction cavities and the living spaces of houses[2] have used two tracer gases in constant composition mode.

3 Gas Chromatography Hardware

The tracer gas equipment described here is fully automated and can operate in one zone in constant composition or decay mode, or it can deal with two zones simultaneously using two separate tracer gases in constant composition mode.

3.1 Hardware Configuration

The major components of the system are a computer controlled gas chromatograph with an electron capture detector (ECD). Other major pieces of hardware are shown in Figure 1. The Capricorn weather monitor can measure wind speed and direction 10 m above ground on the building site, together with temperatures inside and outside the building. Weather data, and the results of tracer investigations were written to disk and, in the case of constant composition multizone results, copied to a larger computer for analysis.

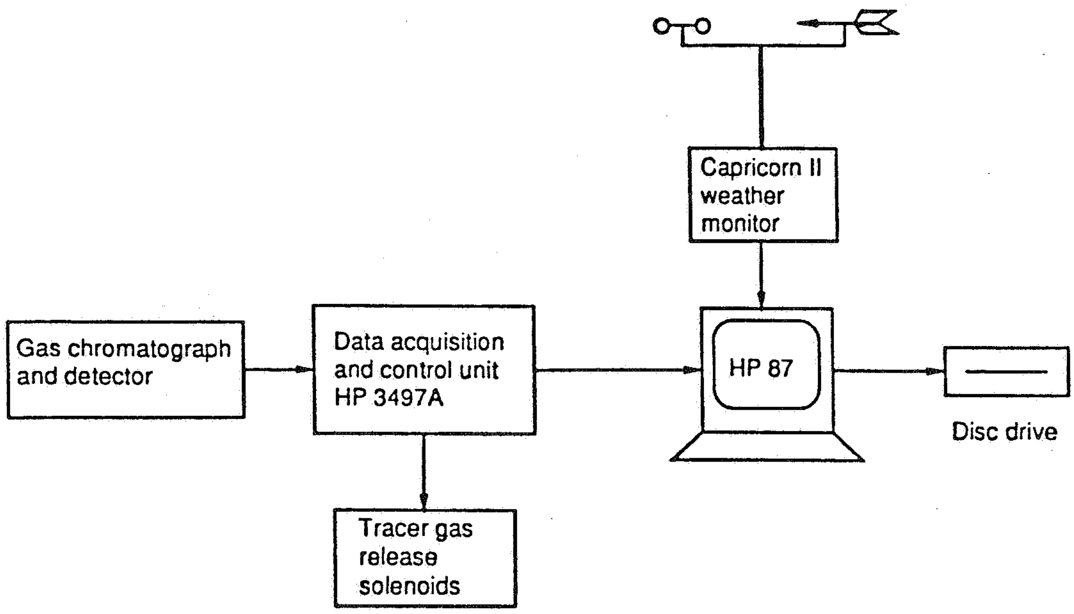

Figure 1: Schematic of automated gas chromatograph

3.2 Tracer Sampling

When working in constant concentration mode in two zones and with two gases, the GC samples air from a zone, measures both tracer concentrations, tops up the chosen zone to target concentration, and then moves on to the next zone. It repeats the process every three minutes, stepping sequentially between zones and writing tracer concentrations and injection volumes to disk.

In automated decay mode, the top-up process takes place after a sequence of 10 concentration measurements have been processed into an infiltration rate. This takes about 30 minutes and the sequence then continues until an interrupt is received from the keyboard.

Sample handling and tracer metering into the zones is achieved with a network of small bore tubes drawn schematically in Figure 2. Here the equipment is illustrated working simultaneously in two zones of a building.

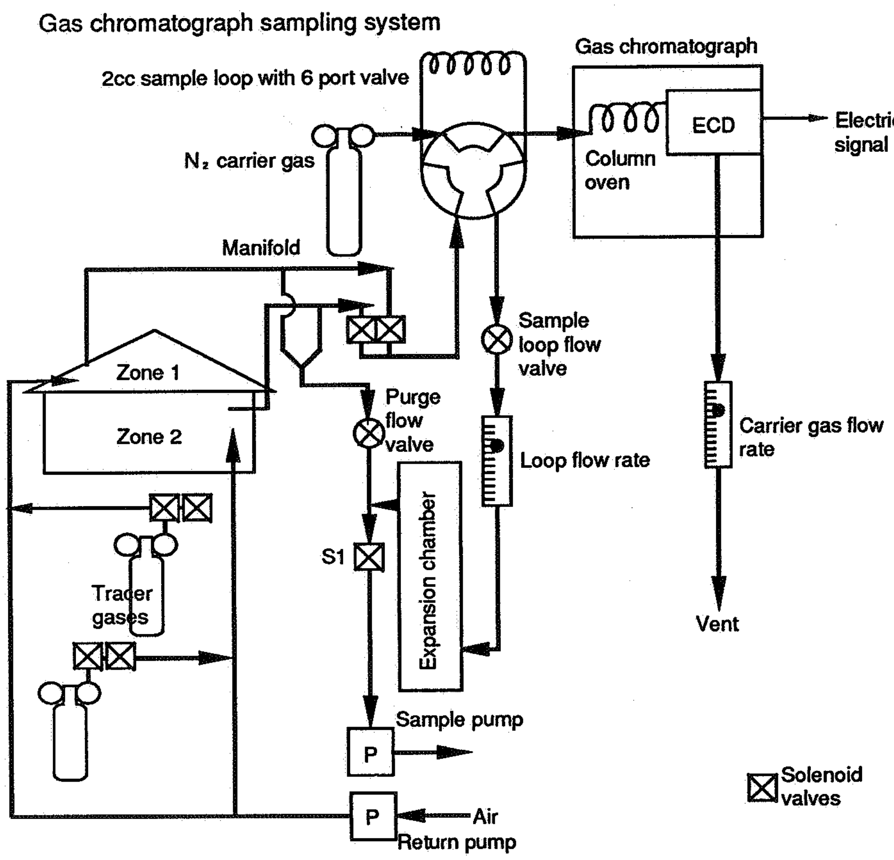

Figure 2: Tracer gas sampling and top-up network

Two air flow circuits were found necessary to ensure that transport delays in the sampling process were kept to a minimum. It would not have been possible to duct all the sample air through the sample loop because the six-port valve has too high an airflow resistance for an adequate air flow to be easily achieved. The first loop is a high air flow ($100cc/s$) circuit that maintains an up-to-date air sample at the GC and delivers top-up tracer gas back to the zone. The second, low air flow rate circuit ($1cc/s$) maintains an up-to-date sample in the sample loop. The six-port sample valve is switched to flush an up-to-date air sample through the GC.

Further notable aspects of design are:

- An expansion chamber which dampens pumping pressure fluctuations to give a steady airflow through the loop flow rate meter.
- A solenoid (S1) which isolates the sample loop from pumping pressure oscillations prior to sampling, thus ensuring the sampling loop always captures the same sample size.
- Location of as many pumps and solenoid valves as possible downstream of the loop to prevent contamination reaching the detector.
- Complete isolation of sampling and tracer dosing networks to prevent cross contamination.

3.3 Tracer Dosing and Mixing

Hardware for topping up tracer concentration in a zone is illustrated in Figure 3. In principle it is similar to the tracer discharge system described by Kumar et al[3]. It releases discrete shots of tracer gas from the small pressure vessel located between two computer controlled solenoid valves. The sequence of events leading to a shot of tracer being released is as follows:

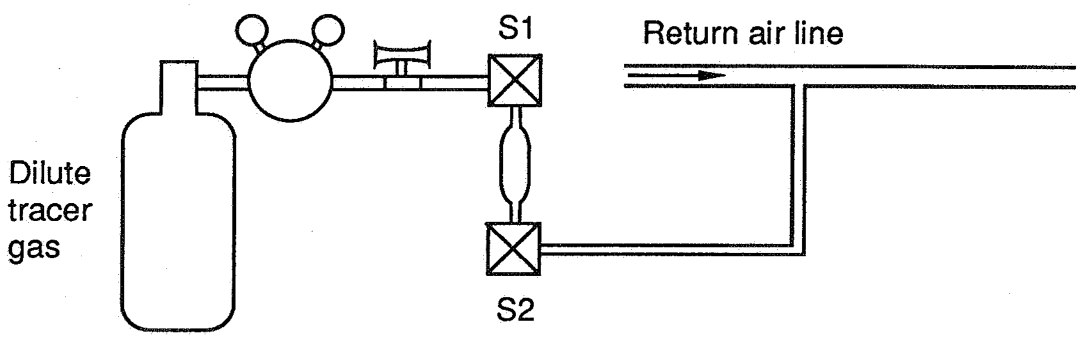

Figure 3: Tracer gas metering system

1. Solenoid valve S1 on low pressure side of regulator opened.
2. Pressure of tracer gas in pressure vessel equalises to regulator setting.
3. Solenoid valve S1 on low pressure side of regulator closed after 200 ms.
4. Solenoid valve S2 on pump side opened.
5. Tracer gas pressure relaxes to atmospheric pressure and excess volume carried to zone by return air hoses.
6. Solenoid valve S2 on pump side closed after 500 ms.
7. Wait 100 ms and return to 1.

Shot sizes were measured for a range of regulator settings and the required number of shots was calculated by a control function within the main program. Working concentrations of tracer gases and shot sizes found convenient in houses are shown in Table 1. The working concentration and amount of SF_6 were chosen to avoid air quality problems in an overspill. In the case of freon-12 (CCl_2F_2) the concentration was chosen to avoid liquifaction.

Tracer type	Concentration	Delivery pressure	Shot size
SF_6	5%	100 kPa	$1cc$
$freon - 12$	1%	100 kPa	$50cc$

Table 1: Details of working tracer gases

Tracer gas delivered to the zone in the return air leg of the tracer dosing circuit was released in front of 400 mm portable fans running at slow speed. This mixing process has been found to minimise tracer concentration differences in the living spaces of houses. In subfloor and roof space construction cavities mixing fans have been shown to give homogeneous gas concentrations, at the same time not causing pressure differences that alter the natural infiltration driving forces. This has been confirmed by showing that major changes in the location and number of mixing fans have not changed the infiltration characteristics of the space.

3.4 Tracer Detection

The gas chromatograph is a Shimadsu GC-Mini 2 fitted with an electron capture detector but without the temperature programming option. Important aspects of configuration and operation, when used to detect SF_6 and freon-12, are as follows:

The hardware specification for the gas chromatograph is as follows:

Sample loop volume	2cc
Electron capture detector source	Ni^{63}
Operating temperature of ECD	140 C
Chromatograph column packing	Molecular sieve
Column operating temperature	100 C

The operational characteristics can be summarised as follows:

Retention time for SF_6	75 s
Working concentration range	1 - 60 ppb
Retention time for freon-12	100 s
Working concentration range	200 - 1500 ppb
Retention time for oxygen	120 s

Working concentration ranges of 1-60 ppb for SF_6 and 200-1500 ppb for freon-12 were found to be consistent with simple linear relationships between integrated output from the ECD and tracer gas concentration.

For SF_6 the relationship was:

$$\text{concentration } SF_6 \text{ ppb} = 1.29 + 60.6 \text{ (area in volt.sec)}$$

and for freon-12:

$$\text{concentration freon-12 ppb} = 5.1 + 1004.4 \text{ (area in volt.sec)}$$

4 Analysis and Control Functions

A control program was written to operate the 6 port sampling valve, analyse the chromatogram into tracer concentrations and run the tracer metering solenoids to top-up tracer concentrations. All of these functions, together with copying weather data and infiltration records to disk, were accommodated within one program. The program structure is illustrated in Figure 4.

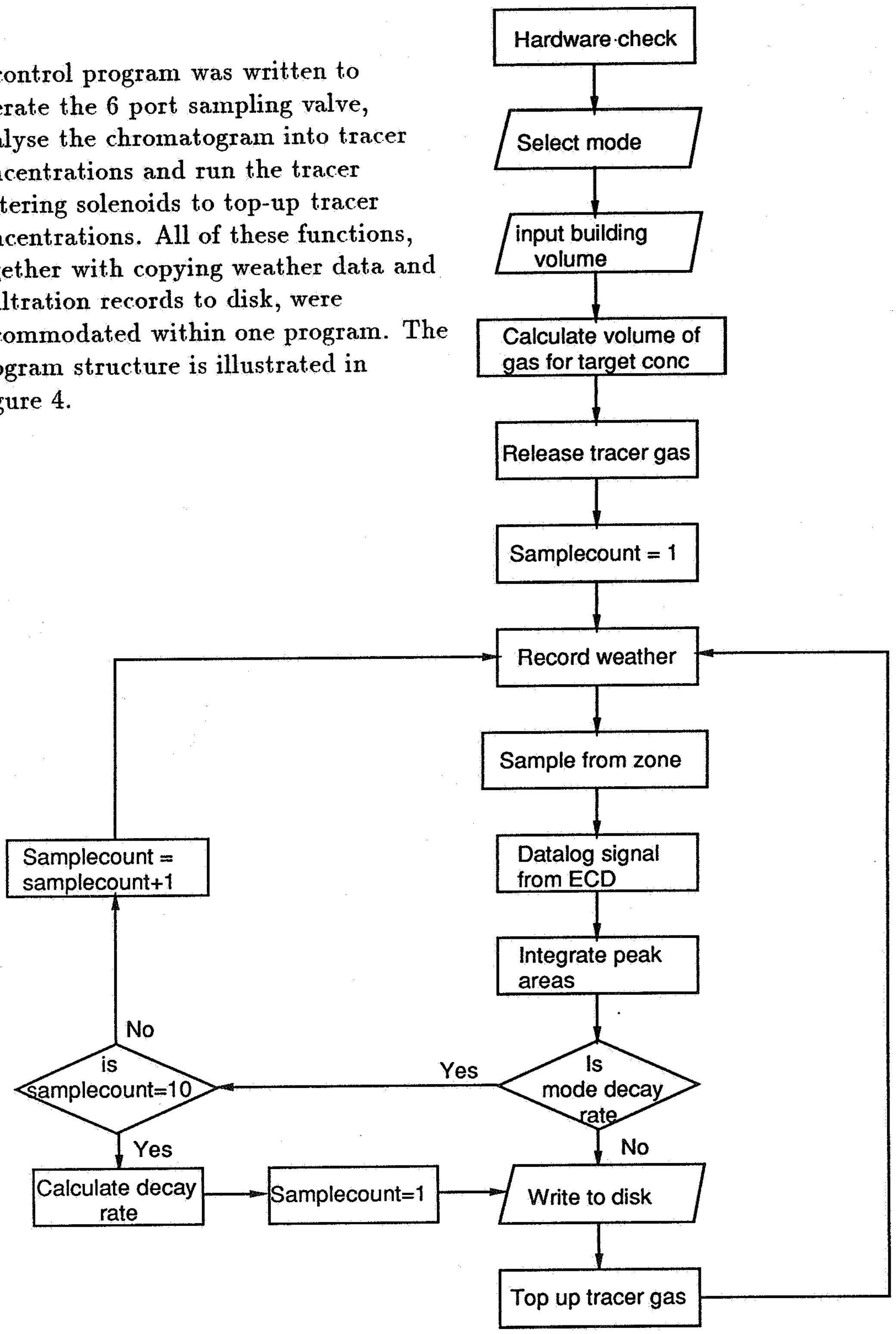

Figure 4: Flow diagram of control and analysis program

4.1 Analysis of Tracer Concentrations

Tracer concentrations were determined from integrated chromatograph peak areas rather then the peak height. This added complexity was considered necessary to allow columns to be swapped without having to have alter calibration constants resident in the program. This turned out to be too simplistic a view, however, as both column changes and variations in carrier gas pressure were shown to influence the calibration constants. As a consequence, it was always necessary to adjust the carrier gas pressure to ensure tracer gas retention times were at standard values. In these circumstances, peak height is an equally satisfactory variable to use in calculating tracer concentrations. Figure 5 gives peak area and peak height for SF_6 over the working range of concentration and both variables are shown to be a linear function of concentration.

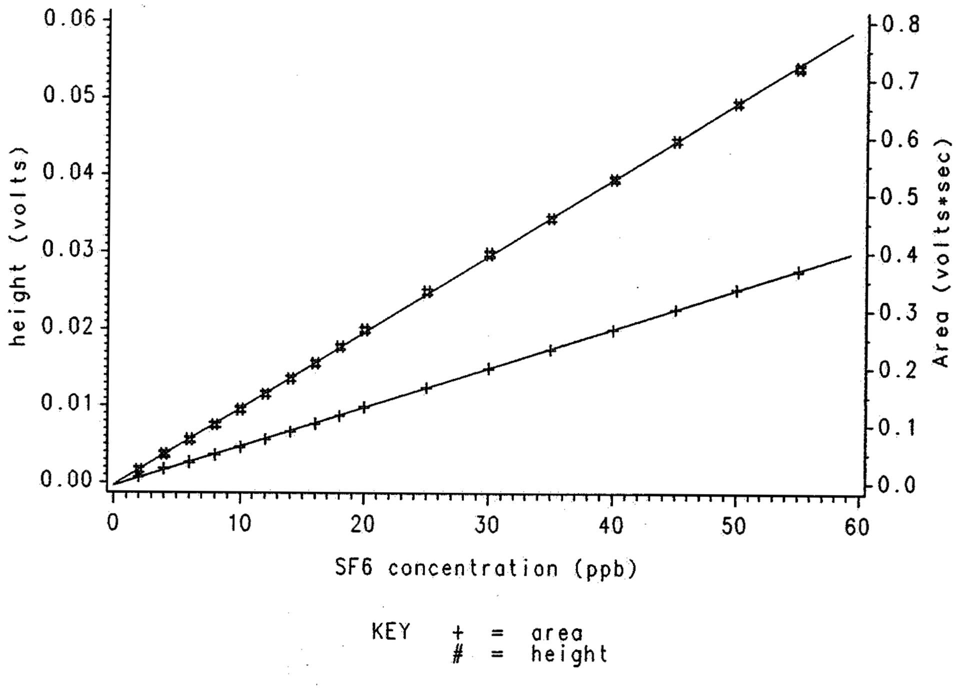

Figure 5: Peak area and height relationship with concentration of SF_6

The accuracy achieved in freon-12 and SF_6 concentration measurement is considered to be around 5% in the calibrated range.

An integration procedure found to be a reasonable compromise between speed and noise rejection is illustrated below. First the analog signal from the electron capture detector was digitised and logged to memory at the rate of 7.69Hz. Then, to eliminate an interference problem with AC pick-up, adjacent data values were averaged to give a new filtered data set. Peak identification, integration and concentration determination were then carried out as follows. Figure 6 is an example of a chromatograph peak and the associated text illustrates how the beginning and end of a peak were identified, with the help of intermediate decision points.

The slope at data point n, Dat(n) is defined as :Dat (n + 4) - Dat (n)

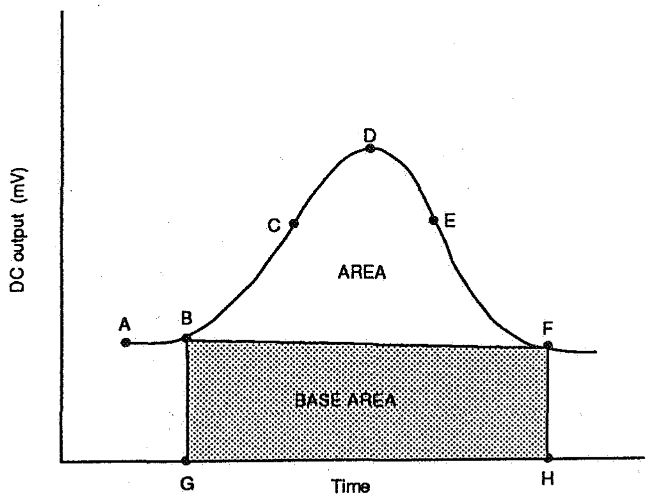

Figure 6: Example chromatograph peak

Point A. Start of data logging.

Point B. Slope> 5. 10^{-5} v/s. This was the minimum slope that ensured base line drift (A-B) did not prematurely trigger peak area analysis routine. This is the start of area integration.

Point C. Maximum slope. Start testing for the maximum Dat(n).

Point D. Maximum height. This defines the retention time. Start testing for minimum slope.

Point E. Minimum slope Start testing for off slope.

Point F. slope > -5. 10^{-5} v/s. End of peak area integration.

The line joining G-H is the zero. Integrated area is the total area under the curve B C D E F H G. Area B F H G is calculated and subtracted to give the effective peak area.

Where multiple gases were being detected, the gas type was determined from the time between sampling and arrival of the gas at the detector (the retention time).

4.2 Tracer Concentration Control

A number of tracer concentration control functions have been investigated but in most cases simple proportional control has proved adequate. It has been used to top-up tracer concentrations in both constant composition and decay modes. Because data analysis using a derivative method[2] with averaging over two hour time periods does not assume constant concentration, the main role of the control function has been to keep tracer concentrations within detectable range. The limiting factor in using more sophisticated control functions has been the speed at which concentration measurements could be made. With two zones being controlled, this transport delay was approximately six minutes and generally in excess of the tracer mixing time constant.

5 Results of Consistency and Reproducibility

Now, after a number of air infiltration and multizone airflow studies have been completed, a range of data can be brought together to review infiltration in New Zealand houses. The opportunity can also be taken to examine the consistency between different ways of measuring infiltration rates, and with infiltration rates calculated using numerical models having a physical basis.

5.1 Living Space Infiltration Rates

Infiltration rates have been measured in nine houses using the automated decay method. In all cases, wind speed and inside and outside temperatures were measured on site, together with the overall airtightness characteristics of each house. Infiltration rates averaged over the 1-3 day duration of the experiment were calculated using the LBL simplified method[4] and appear on Figure 7 compared with measured values.

The quality of agreement between measured and calculated infiltration rates has been found to be within 10% where wind speed measurements were made on site. This margin is equivalent to an uncertainty of about one site wind exposure class. Where wind data has been taken from a weather station some distance away, agreement has been found to be much less satisfactory.

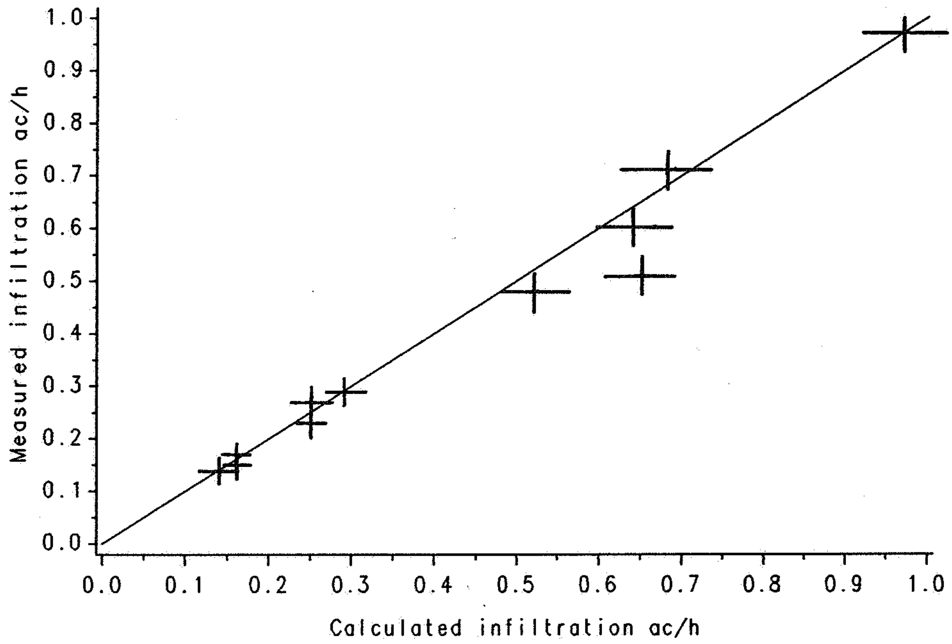

Figure 7: Comparison of measured and calculated infiltration rates with wind speeds measured on site

5.2 Comparison of Decay and Constant Composition Results

Consistency has been observed between results of applying the constant composition and decay methods. Figure 8 illustrates this by comparing one hour averaged infiltration rates measured in the living space of a house using both methods. Although the two methods were not run simultaneously, it has been possible to demonstrate consistency by plotting infiltration rates against the dominant influence of wind speed.

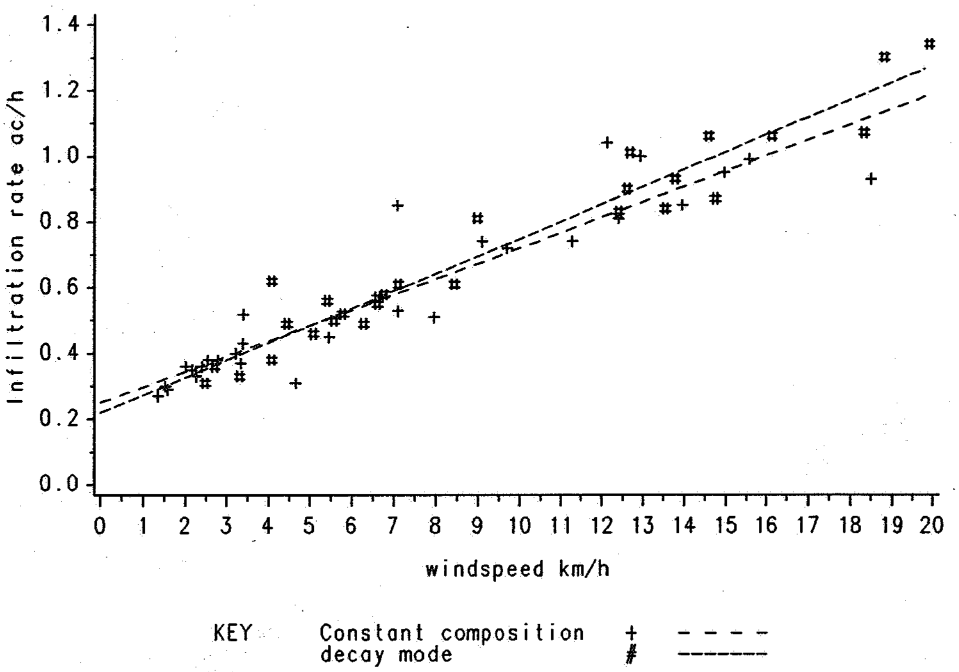

Figure 8: Infiltration rates measured with decay and constant composition methods

5.3 Multizone Results and Predictions

Multi-tracer studies of air flows between the living space, subfloor and roof cavities of houses have been completed and described[2]. Airflows between zones, and infiltration rates into the zones, were measured over periods of several days in five separate houses with the constant concentration method. The airflows measured in these houses have also been modelled using the Walton computer program[5], basic airtightness data for each zone, and wind speed and temperatures measured at each of the five houses. The study[6] allowed a comparison to be made of measured and calculated inter-zone airflows. Figures 9 and 10 are reproduced from[6] to show the extent of agreement between measured and calculated data.

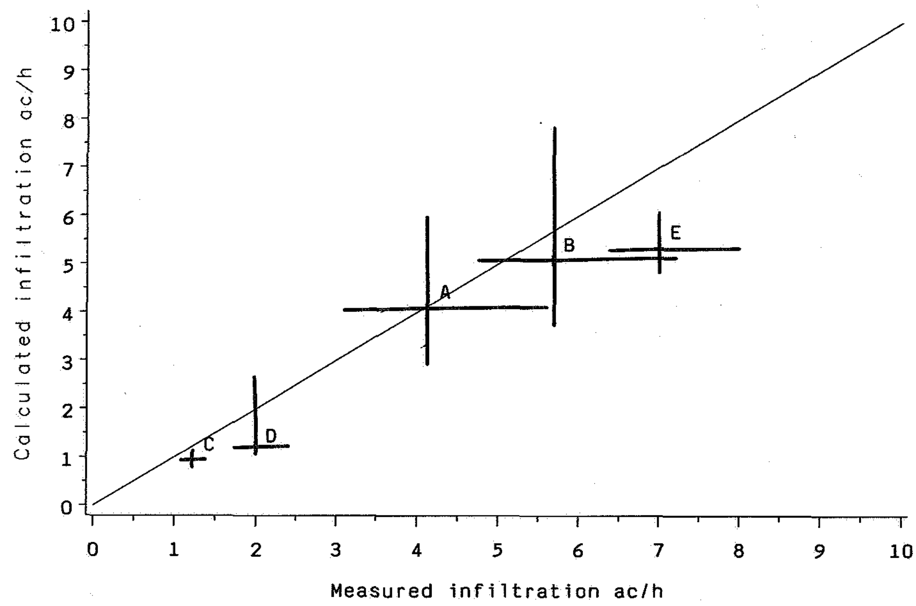

Figure 9: Measured and calculated infiltration rates in the crawl space of five houses A-E

The study has shown that there are no major systematic differences between the results of multi-tracer measurements and calculation but that there is scope for bringing the two closer together. This will involve work in the following areas:

1. Establishing a more comprehensive database of wind pressure coefficients.

2. A better understanding of the distribution of leakage openings in a building, in particular those linking the living space and major construction cavities.

The results of modelling infiltration into construction cavities have generally come within 75% of measured infiltration rates averaged over the same two hour period. On the same basis, 30% to 40% of data points agreed within 25%. In Figure 9 the infiltration rates averaged over the 2 to 3 day duration of the experiment are shown with error bars representing two standard errors of the mean, calculated on the basis of a log-normal distribution of infiltration rate.

The inter-zone air flows compared in Figure 10 were modelled on the fairly severe assumption of a uniform distribution of leakage openings over the building envelope. The data in Figure 10 shows that in some cases the air flows were well reproduced from these simple assumptions but in others, better definition of the linking air flow resistances was needed.

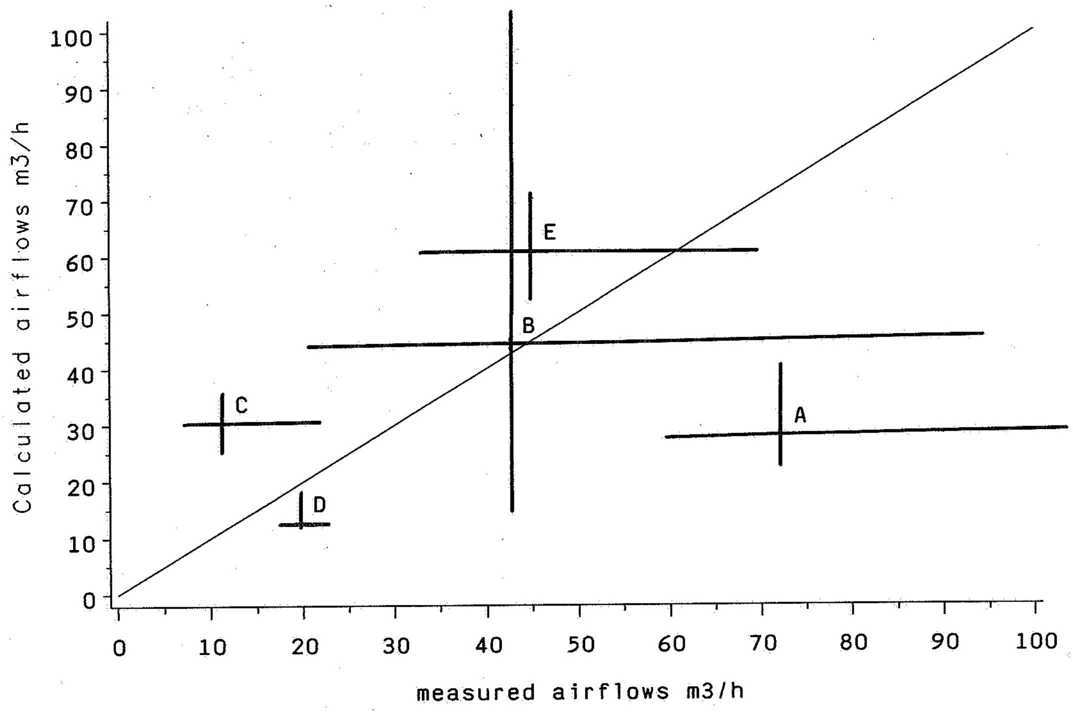

Figure 10: Mean values of calculated and measured air flow rates between living spaces and roof spaces in houses A-E

5.4 Comparisons with a Passive Method

The passive Brookhaven National Laboratory/Air Infiltration Measurements System (BNL/AIMS) developed by Deitz[7] promises to give easily measured average infiltration rates and inter-zone airflows. Recent comparisons between this and traditional tracer methods by Harrje et al[8] and Piersol et al[9] show reasonable agreement in some cases and important differences in others. Both authors highlight adequate mixing as being essential in arriving at agreement between two simultaneously applied tracer gas methods. Major tracer gas concentration gradients caused by open windows for instance, will give results that depend on the location of tracer sources and concentration measuring points. In buildings with ducted air distribution systems, such as the houses surveyed[9], there is generally no problem in achieving uniform tracer gas mixtures. This convenient mixing system is absent in most New Zealand houses and it has been customary to mix the air with portable fans during infiltration measurements. In the following comparison of infiltration measurements, those conducted with the PFT method were carried out without fan assisted mixing, relying instead on natural thermally driven air flows within the building.

Perfluorocarbon tracer (PFT) emitters and capillary adsorption tube samplers (CATS) were maintained for two weeks in two houses (A and B) located in Wellington, New Zealand. Basic airtightness and construction data for the two houses are given in Table 2, together with an infiltration function determined using airtightness data and the simplified LBL method[4]. Infiltration rates measured with the two tracer techniques are separately compared with the average of infiltration rates calculated using the infiltration function.

	Details	House A	House B
House	Volume m^3	195	225
	Surface area m^2	252	290
	Floor area m^2	72	94
Airtightness Details			
	Air changes at 50 Pa	9.4	3.5
	Leakage function m^3/s	$0.0417\Delta P^{0.64}$	$0.0195\Delta P^{0.62}$
Measured and calculated infiltration rates in ac/h			
Calculated using infiltration function		0.25	0.16
Measured using tracer decay method		0.23	0.15
Calculated using infiltration function		0.18	0.08
Measured using PFT method		0.12	0.08
Infiltration functions			
House A	$Q = 0.049(0.133^2\Delta T + 0.0405^2 V^2)^{0.5}$		
House B	$Q = 0.018(0.144^2\Delta T + 0.0765^2 V^2)^{0.5}$		
Where V is the wind speed in m/s at roof height and ΔT is the indoor outdoor temperature difference in C			

Table 2: Details of infiltration study in houses A and B

Although infiltration rates measured with the tracer decay method and with PFTs were conducted at different times, respectable agreement is indicated with the infiltration function in both cases.

6 Conclusions

A gas chromatograph and electron capture detector has been automated to carry out long term surveys of air flows in buildings. Using halocarbon tracer gases in ppb concentrations, the system is shown to be versatile and to give results that are consistent between two modes of operation and with the results of accepted numerical modelling techniques. In more detail, the comparisons between tracer gas studies and modelling show:

1. No major systematic differences between the results of air flow studies using tracer gases and numerical modelling have become apparent.

2. Further improvements in the agreement between tracer measurements and modelling can be expected as more extensive databases of wind pressure coefficients and more detailed descriptions of the distribution of leakage openings in buildings become available.

3. The BNL/AIMS passive tracer method was used to measure infiltration rates in two unoccupied houses and the results were found to be consistent with conventional tracer measurements and numerical modelling.

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PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

10th AIVC Conference, Dipoli, Finland
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Poster 21

INFILTRATION AND VENTILATION DEVELOPMENTS IN NORWAY

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OVERVIEW OF INFILTRATION AND VENTILATION DEVELOPMENTS IN NORWAY

Introduction

This paper gives an overwiew of air infiltration and ventilation developments and trends in Norway. The paper is devided in an infiltration part and a ventilation part.

Some key figures for Norway:

Inhabitants	4.4 mill.
Low rise housing	1.3 mill.
Block of flats	0.3 mill.
Degree days (base: 20 °C)	3500 - 7600 °C D

INFILTRATION

Building code

In Norway we did not have any quantitative code for the airtightnes until 1981. From then the n_{50} -value, one can obtain from a pressurisation test, should be as follows for the different kinds of buildings:

Low rise housing	$n_{50} < 4$ ach
Other buildings up to 2 stories	$n_{50} < 3$ ach
Other buildings more than 2 stories	$n_{50} < 1.5$ ach

In adition we have a functional requirement:

> The building constructions shall be so airtight that the effect of the heat insulation will not be reduced and so that there will not be any draught or moisture problems.

Building constructions and components

After the oil crise in 1974 towards 1980 the buildings became more airtight because the builders became more aware of the tightness´ effect on the energy consumption. In 1979 airtightness of 71 randomly picked single family houses were measured. The houses were located in different parts of Norway, see fig. 1.

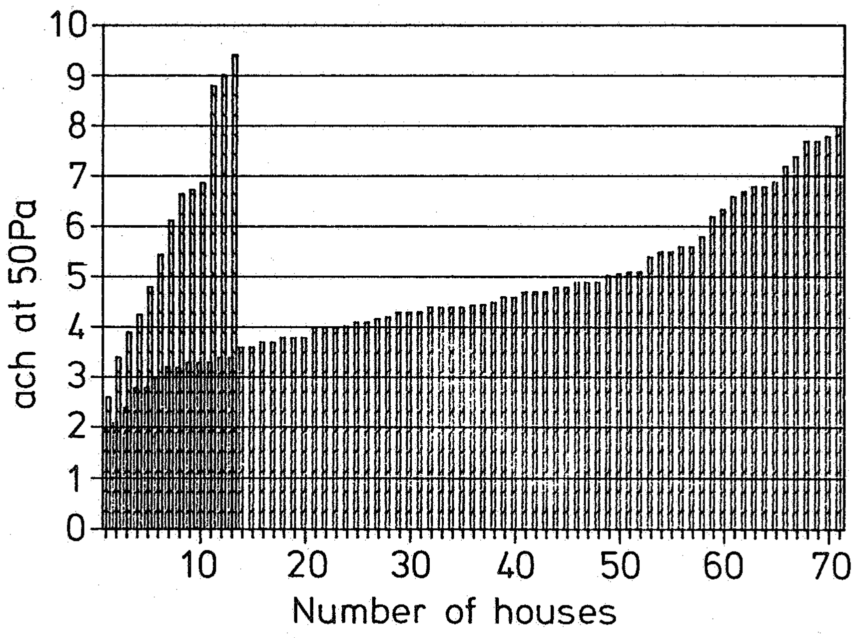


Fig 1. Results from airtightness measurements of 71 randomly picked new single family houses in 1979 and 13 new single family houses and rowhouses in 1988.

The number of houses we have measured is too small for drawing any conclusions but they indicate that new houses in Norway have not been more airtight the last decade. From other measurements we can see the same tendency.

However in the same periode we have seen a lot of new developments in building constructions and components that should have made the buildings more airtight. Among these are:

- thicker polyethylene film as vapour barrier
- new techniques for making the joints in the vapour barrier tighter (welding, sealants, clamping)
- use of rubber gaskets between different kinds of constructions
- more use of sealants in general
- new and lighter wind barrier with larger format which means fewer joints
- tighter windows and doors

A typical single family house construction from today is shown in figure 2.

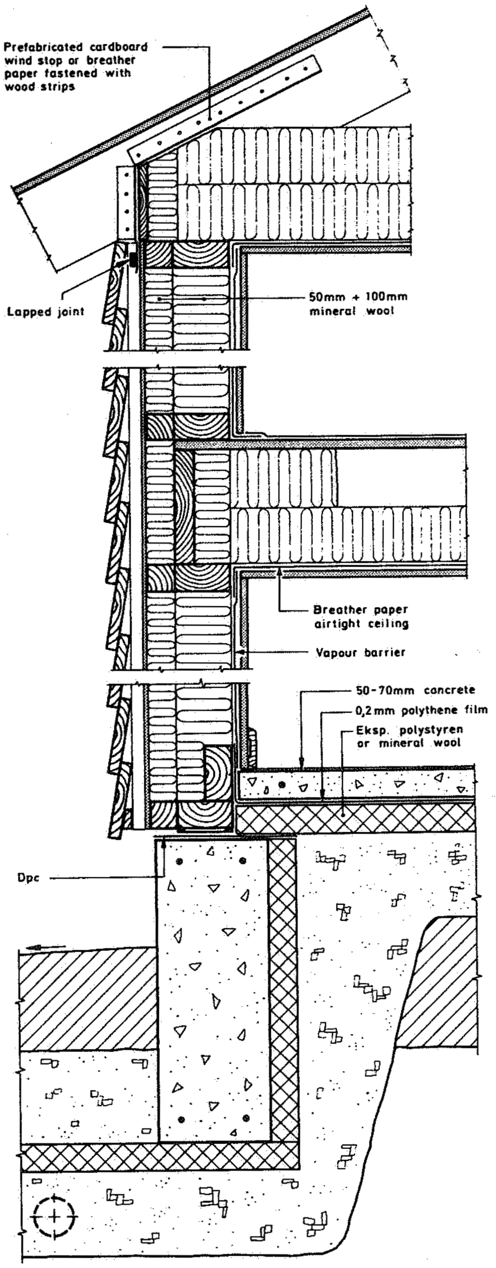


Figure 2. A typical single family house construction in Norway today. More than 90 % of the houses in Norway are wood frame constructions.

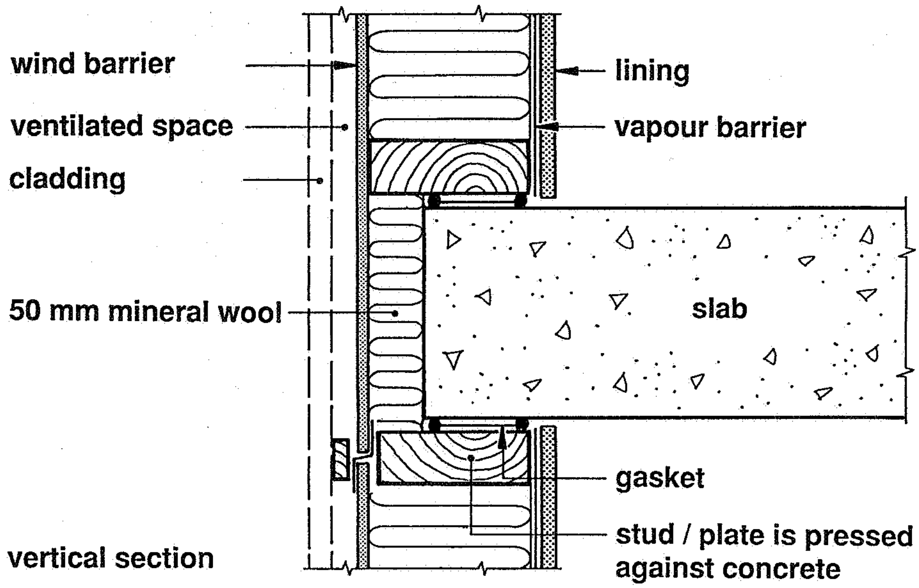


Figure 3. A typical multi family house construction.

There might be several reasons that the buildings are not more airtight today than 10 years ago. The most probable are:

- the form of the houses are more complicated today than 10 years ago
- the sick building syndrome has wrongly been connected to tight buildings and has given the builders an excuse not to put an emphasize on the tightness
- the high building activity in Norway the last years has resulted in lack of qualified persons to the building industry

Research Activity

Rather few research projects have been carried out within this field in Norway the last years. Here is a summary of the most interesting projects in which one of the goals was to improve the tightness of the constructions.

The effect of Vapour Barrier Thickness on Air Infiltration

This project was a comparativ study of the air leakage performance of the two vapour barrrier thicknesses 0.06 mm and 0.15 mm. The study considerd 10 identical single family houses, five which were constructed using the thin film and five which used the thick film. The results show that the houses with 0.06

mm film on average were 17% more leaky than the houses with the 0.15 mm film when they were new. The n_{50}- values were 3.4 ach and 2.9 ach respectively.

The Importance of Wind Barriers for Insulated Wood Frame Construction

The main goal of this research project was to get more information about the influence of wind on the heat loss from wood frame constructions. The project was devided into three parts: calculations, hot-box measurements and wind pressure measurements on a rotable test house. The results show the importance of protecting the insulation layer with a wind barrier to achieve full effect of the insulation in wind exposed constructions, see figure 3.

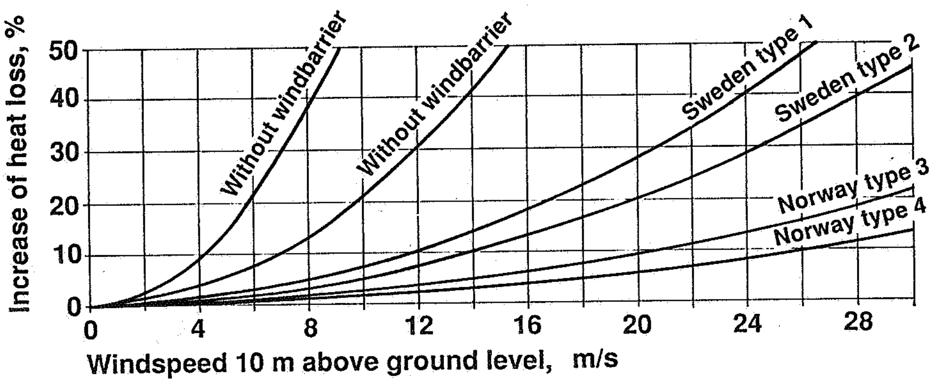


Figure 4. Estimated increase of heat transfer through a stud frame wall caused by wind blowing from outside into the insulation and out again. The two curves without wind barriers represent two different ways of mounting the insulation.

Air Leakage in Old Timber frame Houses with Blown-in Mineral Wool Insulation

Air leakages of 10 old timber frame houses were measured before and after blowing mineral wool into the cavities in the wall and roof constructions. The mean air leakage before insulation was n_{50} = 10.1 ach and n_{50} = 9.0 ach afterwards. This gives an average improvement of 10%. Improvements of 20% were registered on houses where both walls and ceiling were insulated. A couple of houses had no improvement at all.

Infiltration Rates Measurements in Buildings

Infiltration rates meaurements have been carried out in 8 different buildings to find the influence of the weather and the users. The measurements have been carried out with the constant consentration tracer gas technique using SF_6 as the tracer gas. This project is not yet reported but some of the preliminary results are shown in the figure below. The most interesting findings are

- in dwellings about one half of the ventilation is caused by the inhabitants
- in commercial buildings there is a very wide range of what people can accept of high temperature and air quality before they open their windows

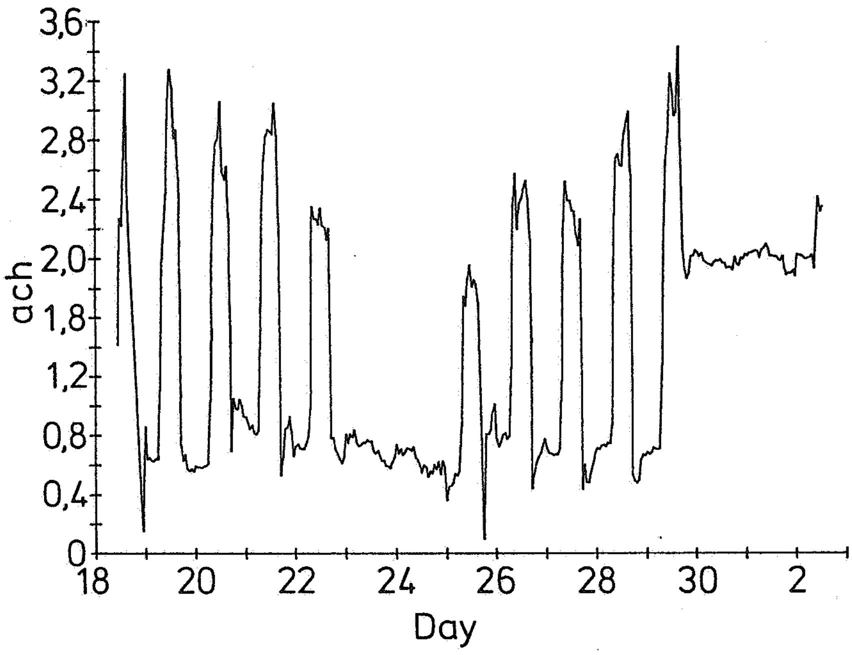


Figure 5. Example of results from air change rates measurements in an office building using the constant consentration tracer gas technique. The building have balanced ventilation system which is turned off during the nights and normaly the week-ends. The last measured week-end it was on and no people was present in the building. When comparing that week-end with the week-days we can see how much the people influence the air change rate. During the first week-end the air change rate is only caused by infiltration.

Infiltration work in the future.

The more we insulate our buildings, the higher is the relative influence of infiltration on the energy consumption in the buildings. We therefore have to carry on with measurements on parameters which effect infiltration including the effect of the occupants. These measurements will give us a better understanding so that better constructions and better calculation methods can be developed.

In Norway we can see a trend in using larger, prefabricated elements in the building industry. To ensure good quality and airtight constructions in the new houses we have to develop better systems for sealing the joints between the elements.

VENTILATION

Ventilation is a mean to secure acceptable indoor air quality. There are a number of consequences of indoor air pollution on both inhabitants and building. These may be grouped in three types.

> Health risk for inhabitants. The sources are among others: Formaldehyde, tobacco smoke, radon, combustion products, organic compounds, humidity and moisture (moulds and fungis)
>
> Annoyance of inhabitants. Main factors are: Body odour, other odours and irritants.
>
> Damage to building fabric. Main factors are: Humidity and moisture.

Many of the pollution sources have their origin in building materials, building furnishing and decorations and processes and activities taking place within the building. Ventilation requirements has to take all the mentioned factors into account.

CODES AND GUIDELINES

The Norwegian regulations concerning building design and construction are composed of laws, codes and guidelines. There has been a shift from codes containing detailed requirements to codes based on functional requirements supported by guidelines containing information on how to fulfill the codes. The guidelines give examples of solutions that complies with the functional requirements, other approved solutions may be accepted as well. A collection of approved solutions is found in the Building Research Series published by The Norwegian Building Research Institute. These series cover solutions to design/construction details and technical installations as well as planning solutions.

Functional requirements

> Building shall have ventilation securing a satisfactory indoor climate in each and every room.
>
> The ventilation plant shall be built in such a way that good energy economy is secured, it is easy to control and easy to maintain.

Those parts of the plant that must be cleaned in order to function properly has to be made in such a way that all these parts can be cleaned from dust and other airborne depositions.

The ventilation plant shall be built in such a way that the capacities according to the requirements can be measured and adjusted.

Before the building is taken into use the ventilation plant should be controlled and adjusted in order to satisfy designed capacities.

The air quality within a room shall be kept at a level that will not, the use of the room taken into account, lead to discomfort or health risk.

The air flow between rooms shall be from a room with a low concentration to a room with a higher concentration.

The ventilation plant shall prevent the spread of gases and particulate matter that smells and/or are dangerous to health. Such pollutants shall be captured by means of local exhaust.

The ventilation plant shall prevent the spread of contaminants to the outside if this represents a danger to health or discomfort.

The guidelines supporting the building codes give recommended minimum ventilation rates which for public and commercial buildings are given as $m^3/h\ m^2$ floor area. For dwellings the requirements are a certain crossection of stacks for natural exhaust from specific rooms or specific exhaust air flow rates from the same rooms when applieing mechanical exhaust.

Public and commercial buildings

Room	Outdoor air	Pers./m2	Outdoor air/pers.	IEA Annex 9 Outdoor air req.
Work rooms	5 m3/h m2	0.3	16.5 m3/h	0.5-1.0 ach for
Assembly halls	10 "	1.5	6.7 "	moisture and
Sporting halls	10 "	0.3	16.5	humidity.
Kitchen	15 "	0.3	50	20-30 m3/h pers
Offices	5 "	0.1	50	for body odour.
Stores	5 "	0.3	16.5	30-70 m3/h pers
Class rooms	7 "	0.5	14	for tobacco smoke.
Waiting rooms	3 "	0.6	5	

As we can see from the tables, the ventilation for office rooms can be said to be of a high standard compared with the recommendations found in the IEA annex IX, minimum flow rates, while the standard is very low for class rooms and assembly halls. This is also reflected in practice where one almost without exceptions have complaints in classrooms, kindergardens and assembly halls ventilated according to the recommendations in the codes.

Dwellings

Room	Natural exhaust. cm^2	Mech exhaust m^3/h	Suppl. air grilles , slots, ducts
Living rooms	-	-	Openable window or 100 cm^2
Kitchen	150	60	Slot over under door 100 cm^2
Bathroom with or without WC	150	60	"
Sep. shower or sep. WC	100	40	"
Washing- and drying room	150	80	Openable window or 150 cm^2 in outer wall, 150 cm^2 between adjacent rooms

The ventilation standard will vary according to how many rooms requiring exhaust there are in the house. Typically, the ventilation, using mechanical exhaust, will vary between 0.5 and 1.0 ach. In natural ventilated dwellings the ventilation is not under control and the ventilation standard is usually rather low. Many homes financed by the The State Bank for Housing have only natural ventilation. The effect of this is that the ventilation in houses financed by the State Bank for Housing will be rather poor compared with larger dwellings of higher standard.

General comments

The recommendations in the codes obviously do not take properly into account all the pollution sources in a building not even the number of persons occupying the space. For this reason one can say that the ventilation standard is rather uneven.

The shortcomings of the present guidelines are obvious and they will be revised in the future. The Nordic Committee for Building Regulations is working on a revision of the air quality section of The Nordic Guidelines for Indoor Climate. The outcome of this work will have a great impact on the future Norwegian air quality and ventilation guidelines.

SYSTEMS AND COMPONENTS

Codes and guidelines are still governed by the view that the human beeing is the main source of pollution indoor, which is no longer the real case. Up to the sixties this was probably more true since the general practice before that time was to use well known "natural" building materials. Since the sixties the development in the field of materials for the building and its furnishing has been allmost explosive in an historical time perspective. This is especially the case for surface materials like panels, wallcovers, carpets and paintings, introducing a large, ever increasing number of chemical cmponents in the indoor environment, components which not at all are good for our health.

Dwellings

The building codes are generally the bases for ventilation requirements.

Natural ventilation was dominating up to mid sixties. However, mechanical exhaust was used for kitchen ventilation either as a genaral exhaust or through exhaust hoods above the stoves.

Gradually total mechanical exhaust was taken into use, first of all in dwellings not financed by the State Bank of Housing. Exhaust air was taken from kitchen, bathrooms, WC and washingrooms, in the same way as for natural exhaust. The principle of exhaust ventilation is shown in fig.6.

The change to mechanical exhaust led to better standard of ventilation. Especially, problems with moulds and fungis in the "wet-rooms" were reduced. Nevertheless, mechanical exhaust is not without problems. Fans and ducts are not maintained in a proper manner. Fans stop running and ducts become dirty. When this happens the ventilation becomes worse than with all natural exhaust, because of the smaller dimensions of the ducts. This is still a problem.

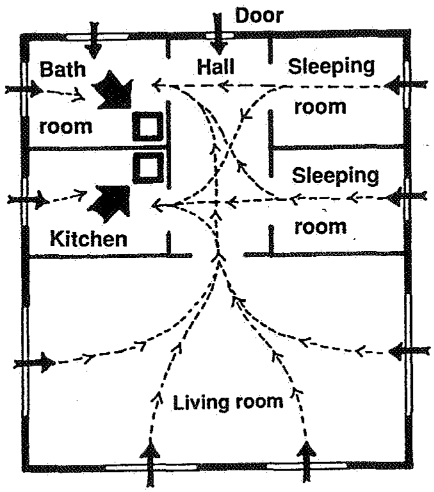
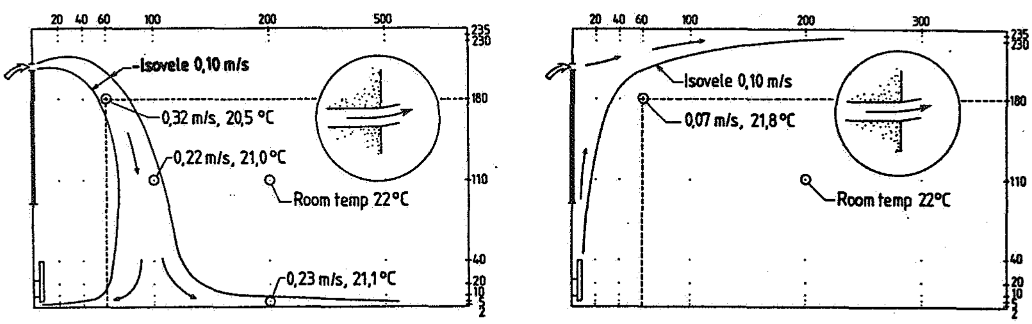


Figure 6. The main routes of the ventilation air flowing through a building with exhaust ventilation.

After the oil embargo in 1973 the standard of ventilation was reduced because of the great focus on energy saving, where the simpliest mean was to reduce the infiltration and/or ventilation of the buildings. This was done by tightening the building and closing the air valves. Buildings with natural ventilation suffered most from this. Buildings with mechanical exhaust suffered also because the fans were not sized for the resulting smaller inlet flow crossections. The need for ventilation because of the new development in building materials required in reality an opposite approach.



a. Air flow pattern with a slot outlet. The supply air temperature °C. Outdoor air flow rate 5 l/s. Panel heater 0 watt.

b. Air flow pattern with a slot outlet. The supply air temperature °C. Outdoor air flow rate 5 l/s. Panel heater 350 watt.

Figure 7. Exhaust ventilation. When the heater is on, draught can be avoided.

Mechanical/natural exhaust ventilation are installed with outdoor air supply openings/valves in the outer walls, generally located in the upper window frame, but also other locations are used. This may lead to drauhgt problems. Investigations carried out at the Norwegian Building Research Institute have shown that this is of no problem when the room heaters under the windows are not shut off, fig.7. However, when using radiators with thermostats the heat will go on and off, resulting in draught risk.

There has been some activity developping components overcoming this problem. One method has been to integrate the radiator and the outdoor air supply, fig.8. Another has been to develop supply devices (diffusers) giving draughtfree ventilation with unheated outdoor air. The principle is to create a more intense mixing with room air, in and close to the diffuser.

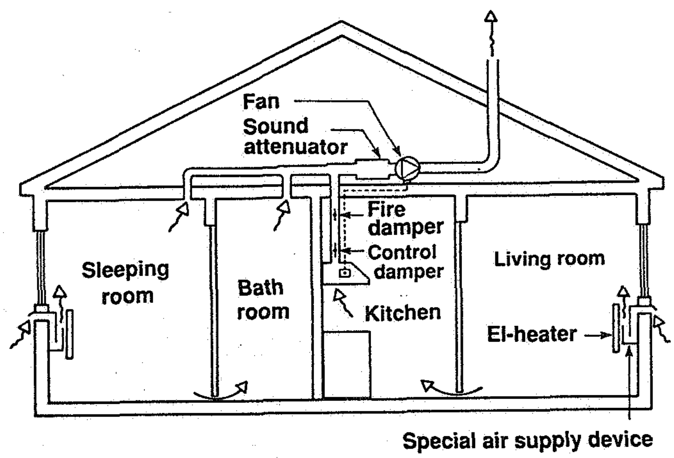


Figure 8. Mechanical exhaust ventilation. The fan is placed in the attic. The figure shows how draught problems can be solved by using special components integrating the room heating device and the air supply device.

The oil embargo also resulted in an activity developping balanced ventilation systems with heat pumps and heat recovery. Such systems allow a higher standard of both ventilation and thermal comfort, see fig.9, without an increase neither in energy consumption nor in the total cost per year of owning and running the building. However, the State Bank of Housing does not finance such solutions.

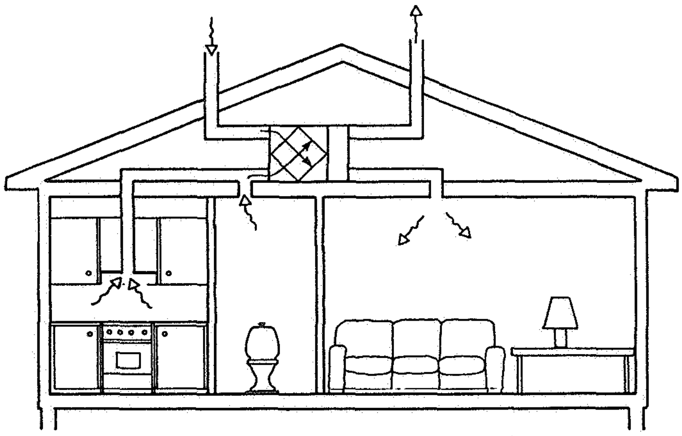


Figure 9. Building equipped with balanced, mechanical ventilation and heat recivery unit.

Systems with heat pumps using the exhaust air as heat source for heating the warm water sypply, is increasing in popularity, fig.10. Such systems are now made for also preheating the ventilation air.

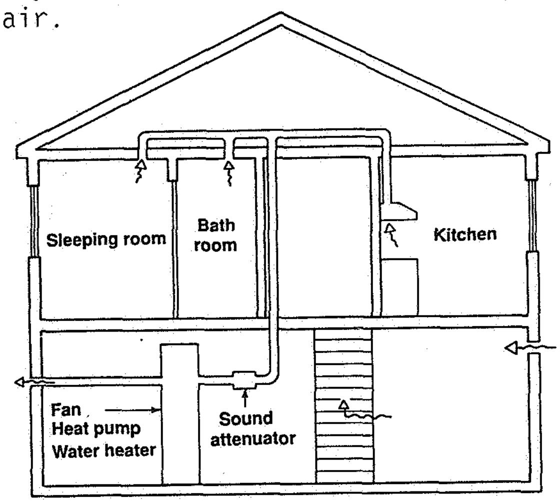


Figure 10. Example of mechanical exhaust ventilation combined with a heat pump. The fan is placed in the basement. Fresh air is taken throug the basement and is supplied to the living rooms via the stair well. In this way a certain preheating of the fresh air is achieaved.

The future

Future ventilation systems will certainly be balanced systems with heat recovery and/or heat pumps because we will have tighter constructions in the future. The solutions will take advantage of the possibilities for using cheap microprocessor technology in what we may call "intelligent" ventilation. This will be some kind of demand controlled ventilation. The strategy will be to supply ventilation air to that part of the building which is occupied or in use. In day time to the living area, in night time to the sleeping area in combination with timer control of for example bathroom exhaust, either user activated or occupancy sensor activated. This will lead to much better air quality and comfort without a substantial increase in total ventilation airflow requirements. A certain minimum basic ventilation must be secured all the time to unoccupied areas. The exhaust air will be from the same zones as we find in existing practice in addition to other "intelligent" exhaust points.

The use of heat pumps and solar assisted preheating of the ventilation air will be common.

Public and commercial buildings

The minimum ventilation requirements in the codes are generally the bases for outdoor air flow rates. Higher flow rates are occasionally used. On the other hand, source control are generally not properly applied. However, due to lack of maintainance and adjustment, lower flow rates than the design values are generally found in practice.

Mechanical balanced ventilation has been the most common practice the last twenty to thirty years, allthough mechanical and natural exhaust systems has been used to some extent. Natural and mechanical exhaust has been most common in smaller office buildings, schools and kindergardens. Because of the very low ventilation requirements in the codes for schools and kindergardens most of the sick building problems are found in these types of buildings.

The oil embargo led on the one hand to reduced ventilation through more recirculation (reduced outdoor air supply) and no ventilation during nonoccupancy and on the other hand to more energy economic ventilation like the development of heat recovery equipment.

Systems that has been used:

Induction systems with heating and cooling facilities. Not widely used to day.

Single duct systems with zonal reheating, with/without fan coils for cooling.

Single duct systems with zonal reheating and cooled ceilings.

Single duct systems with separate roomheating without/with cooled ceilings or fan coils for cooling.

Dual duct systems. Very little used.

VAV systems.

All these are mixing ventilation systems, fig.11.

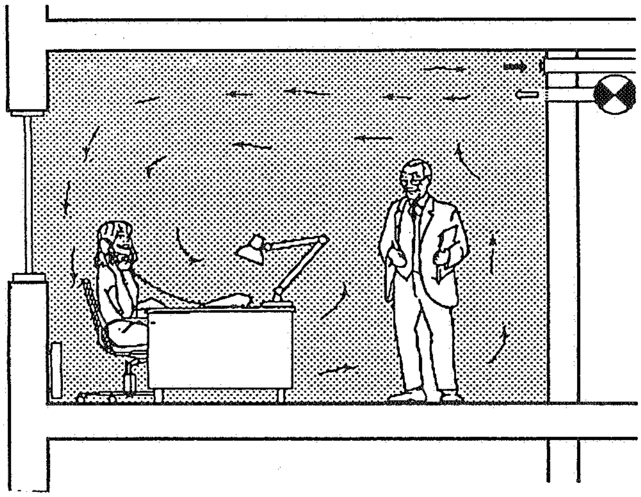


Figure 11. Mixing ventilation. Good mixing results in even concentrations in the whole ventilated space. The concentrations are the same as in the return air duct.

All-air systems have been demonstrated to have a substantial reduced ventilation effectiveness when the heating demand requires supply air temperature more than a few centigrade above room air temperature.

Recirculation leads to unfavourable spreading of pollutions and it is now a trend to less use of such solutions. Heat recovery is now allmost 100 % used and it is for this reason less arguments for using recirculation as a mean for energy saving.

In spite of increased knowledge and technical development the sick building problems seems to be increasing all the time in Norway. The reasons are likely to be that ventilation requirements and system solutions are not adjusted to the ever increasing sources of bad air quality or on the contrary, the pollution sources are not properly identified in order to apply the right measures for eliminating the sources. Another problem seems to be ventilation plants which are not properly cleaned neither at the time the building is taken into use, nor afterwards.

Norway has during the last decade been among the leading parties in the field of research in ventilation efficiency. The outcome of this is a new ventilation strategy which is called displacement ventila- tion, fig.12.

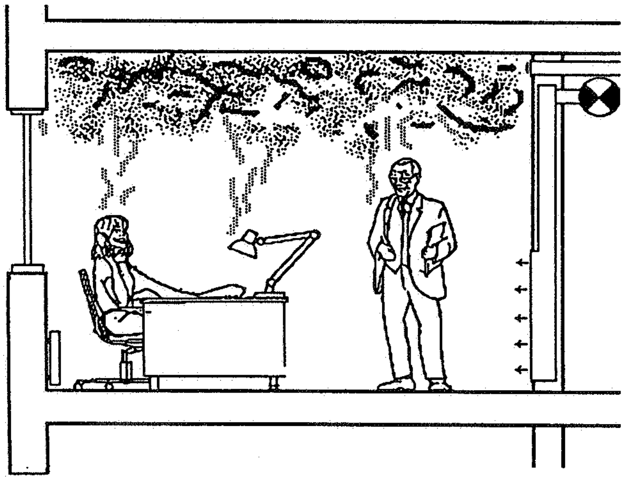


Figure 12. Displacement ventilation. The pollution concentrations increase with height. The air is transported upwards by convective currents. The concentrations in the breathing zone are lower than in the return air duct.

Another name is soft ventilation becasuse the ventilation air is supplied with low velocity (low momentum) directly to the occupied zone. Allthough this ventilation strategy for a long time has been utilized in industrial ventilation , it has been too little knowledge of the physical behaviour and theoretical discription of the method. The lack of theoretical treatment may be the reason for that, allthough treated in the litterature more than 100 years ago, there has been little application of the principle in the past.

The concepts of ventilation effectiveness is based on the age concept where age is defined as the time elapsed since the ventilation air entered the room, fig.13.

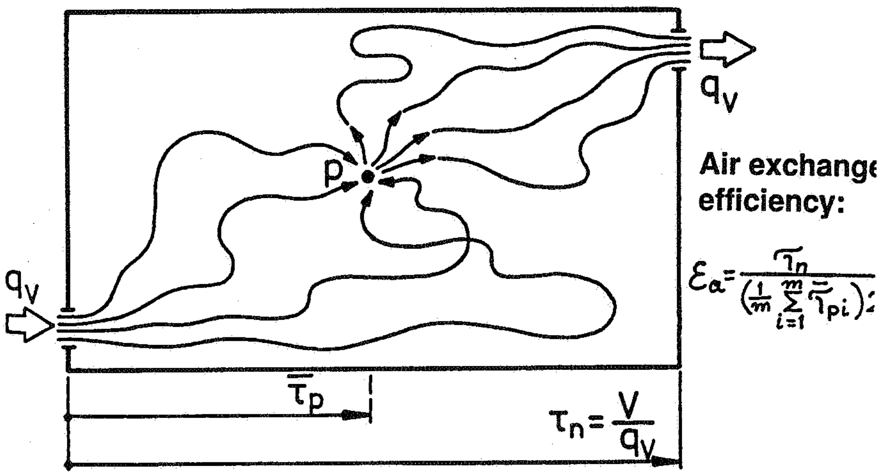


Figure 13. The local mean age, $\bar{\tau}_p$, of thew air is defined as the average time which will elapse after the air entered through the supply opening until it arrives at a certain point. The average age is the mean of the local ages.

Local age is the age of the air at a point in the room. Average age is the average age of all the air in the room. The shorter the age relative to the nominal time constant the better the ventilation potential is. The relative age is called air exchange efficiency. The shortest possible ages are found when the ventilation air flows like a piston through the room (plug flow). Displacement ventilation is to create a flow pattern that is as close to plug flow as possible, fig.14. The realative air quality or contaminant removal effectiveness is measured as the ratio between the concentration of pollutions in the return air and the concentrations in the breathing zone. The heat removal effectiveness can be defined in the same way, changing concentration with temperature.

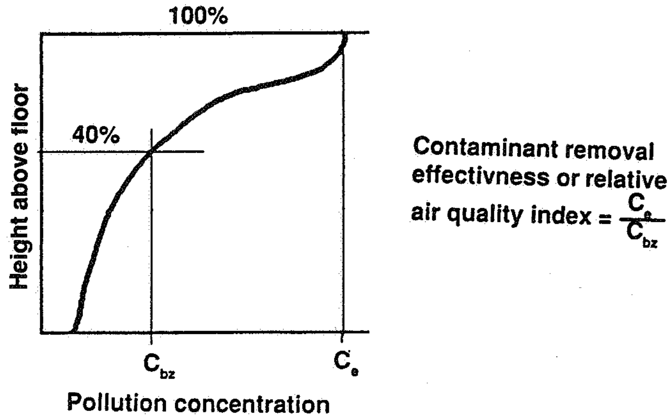


Figure 14. The shape of the vertical concentration gradient shows the displacement effect.

It has been demonstrated that the displacement principle results in a better ventilation effectiveness than mixing ventilation, and it also makes it possible to utilize free cooling with outdoor air in a much larger scale. However, it has also been demonstrated that it is necessary to adjust the ventilation flow rates to the number and strength of the heat sources present in the ventilated space. In normal Norwegian office rooms it is required to have ventilation flow rates as high as 7-8 m^3/h and m^2 floor area in order to have a significant displacement effect. For all ventilation systems there is for thermal comfort reasons a limitation in cooling capacity. Displacement ventilation is no exception to this. The cooling capacity is limited because of the limitation in vertical temperature gradients in the occupied zone. Increasing the cooling capacity means increasing ventilation air flow rates, better diffusers or installation of separate cooling systems like radiative cooling or a combination of radiative and convective cooling. There is at present research and development activities in all these areas.

Using displacement ventilation it is necessary to have separate heating systems. The trend in Norway is now to separate all three functions, ventilation, cooling and heating. Doing this one can optimize all three functions and get a more relieable operation of the HVAC systems in a building.

Ventilation strategy in the future

The objective is to secure healthy buildings with a comfortable indoor climate.

There will be strict requirements to the building, its furnishing, activities and processes. Among other things we will see building materials with health declarations.

Since there is difficult to use only quite "clean" materials there will be a requirement for increased ventilation during the first year of a new building or the first year after a renovation.

It is of great importance that the user can have an influence on his environment, therefore, individual control will be a basic requirement in future ventilation.

Of energy economic reasons, demand controlled ventilation will be a requirement, like a requirement for high ventilation effectiveness (displacement ventilation, spot ventilation).

Another important requirement is ventilation plants which are easy to clean and maintain, equipped with operating instructions which the building owner easyly can understand and also are easy to use.

Last but not least it will be necessary to put into practice a better quality control procedure in the building construction phase and a procedure for repeated control of the indoor air quality, for instance every second year.

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PROGRESS AND TRENDS IN AIR INFILTRATION AND VENTILATION RESEARCH

10th AIVC Conference, Dipoli, Finland
25-28 September, 1989

Poster 22

AIR INFILTRATION AND VENTILATION. PROGRESS AND TRENDS IN SWEDEN.

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1 BACKGROUND

The climatical conditions in Sweden are such that it has almost always been necessary to tighten the houses quite thoroughly in order to avoid cold-draught and to make as good use of the heating as possible. Devices for intentional ventilation, such as ducts for the exhaust of "used" air, have been installed in Swedish houses for centuries.

Thus the concept of infiltration and ventilation is not a new one. The more general introduction of central heating in the beginning of this century led to a relatively high degree of dependance on imported fuel. As a consequence of rising heating costs, aims to protect the environment, increased demands on indoor comfort etc. Steps to make houses more airtight and the ventilation more reliable and efficient have been taken during the last decades.

It was not until the seventies that a more systematic approach to the problems of airtightness and ventilation was undertaken. The Swedish Council for Building Research initiated the formation of the so called "Tightness group" in 1977. The group consisted of researchers in building physics and ventilation technology and practitioners - consultants and contractors.

Since the start in 1978, Sweden has been a member of the IEA annex V, Air Infiltration and Ventilation Centre. The collaboration between scientists and practitioners from the different countries in AIVC has proved to be successfull and of substantial value for Sweden, Boysen (1989).

2 AIR INFILTRATION

2.1 Air tightness of buildings

2.1.1 Benifits and drawbacks of airtight buildings

Nowadays, there is a consensus among researchers and practitioners in Sweden that there are a number of benifits of airtight buildings.

Airtight buildings

- are a prerequisite in order to make the ventilation of buildings perform well, thus having control of flow rates etc.
- prevent cold draught from air leaks.
- conserve <u>energy</u> because of low uncontrolled leakage flows.
- conserve <u>power</u> for heating in cold, windy situations.
- prevent rapid cooling of the indoor air in cases of heating equipment failure or uneven heating power deliveries e.g. in crisis situations.

- prevent moisture problems and -damages due to moisture convection (normally due to exfiltrated air).
- improve the acoustical properties of the building envelope thus preventing noise from the outside.
- are a prerequisite if heat is to be recovered from the ventilating air.

A drawback of airtight buildings may be that they demand a proper and well-performing ventilation system in order to guarantee a good indoor air climate, which leaky buildings don't.

2.1.2 Whole building measurements

The pressurization test of the airtighness of whole buildings was established in Sweden around 1975. Since then a certain - but unknown - number of buildings, mostly single-family houses, but also large buildings e.g. industrial buildings, have been tested in our country. The test results haven't been systematically assembled so an analysis of the trends isn't possible to perform.

The practical experience among contractors and building inspectors however is that the building code requirement of n50<3.0 ach is not very difficult to achieve in construction practise. However there have been some obvious problems to reach n50<1.0 ach, when this has been demanded by the commisioner of the building project.

2.1.3 Building components, measurements and calculations

For some specific building components - exterior doors and windows- quite extensive measurements have been done during the last 15 years in Sweden. This is primarilly due to the procedure of type approval of these products. The tests have normally been performed by the Swedish National Testing Institute. Brolin(1980 and 1984).

Measurements of the air tightness of whole building components, walls, roof elements etc, and connections between components have been performed only occasionally.

If leak paths can be described geometrically, it is possible to calculate the the air leakage characteristic of, for example, connections between building components. This is described in Kronvall(1980).

2.2 Driving forces for air infiltration

2.2.1 Wind

The wind climate of buildings has been investigated quite extensively in Sweden. Wind tunell measurements have been performed by the Swedish Institute for Building Research and others.

Until recently most investigations have concentrated on wind forces acting on buildings from the structural design point of view. However, during the last few years investigations with the aim of determining pressure coefficients for low-rise buildings have been performed. The results are suitable and useful for infiltration and ventilation calculations. Wiren(1985 and 1987).

Wind pressures on low-rise buildings have been studied by means of full-scale measurements by Gusten(1989). Mean values as well as fluctuations of the wind pressure have been investegated.

2.2.2 Stack effect

Systematic investigations of the stack effect in buildings have not been published in Sweden. However, at a seminar on moisture convection and pressure conditions held in Stockholm 1987, it was agreed upon that the stack effect in many cases had led to moisture damages in roof spaces, most commonly in buildings with high indoor moisture load, but also in "dry" buildings, such as tall office buildings, 1 1/2-storey single family houses etc.

2.2.3 Ventilation system

The pressure conditions due to the ventilation system in a building is of importance for air infiltration and exfiltration. In cases when outdoor air is led more or less directly into the house through supply-air vents, which is the case for naturally ventilated houses with ducts and exhaust fan ventilated houses, a certain part of the supply air is infiltrated through the building envelope. The percentage of infiltrated air compared to the air entering via vents is apparently high in untight buildings.

In mechanically exhaust and supply air ventilated houses it has been demonstrated that risks for indoor over-pressure is evident if the ventilation system is not thorougly adjusted and/or the cleaning of the exhaust air filters has been disregarded. The indoor over-pressure can cause severe moisture damages due to moisture convection.

3 VENTILATION

3.1 Ventilation systems

Today almost all newly produced buildings in Sweden - including single family houses - are equipped with mechanical ventilation systems. Heat recovery from the exhaust air is normal practice. In systems with exhaust fan ventilation the heat recocery is arranged by means of an air to water heat pump heating domestic hot water and/or water in a hydronic heating system.

In single family houses warm air heating is quite common today. The systems are normally exhaust-supply ventilation systems, where the warm air is blown into the different rooms in the house through terminal devices while the make-up air normally is exhausted through a centrally placed device. Recirculation of air is necessary in order to keep the warm air entering the room at a limited over-temperature and to maintain a good ventilation efficiency. However, the use of recirculated air could give indoor air pollution problems.

In multi-family houses, office buildings, hospitals, industrial halls etc. the ventilation system always includes exhaust and supply fan ventilation combined with some kind of heat recovery equipment.

3.2 Ventilation performance

The performance of ventilation systems has been studied early in Sweden. Studies from the late forties, Rydberg(1949), of ventilation in dwellings showed that the real performance of ventilation systems often differs from the intended behaviour. Later studies, especially on modern complicated systems, support this. An oftenly quated study, Allhammar & Sundell(1985), concludes regarding less than 10 years old ventilation systems:

- The flow rate through air terminal devices varied between + 50% and - 100% (i.e. zero) compared to design values.

- Every third fan was not performing well, which caused reduced fan capacity.

- More than 10% of the air heaters and filters were so dusty that the flow rates were reduced significantly.

- More than every tenth fan was not running, in some cases intentionally.

- Every tenth of the ventilation systems was running with 100% recirculated air.

- In almost every tenth of the ventilation systems filters in the supply air ducts were taken away or broken down in their suspensions.

The air flow field in a space, for example a room, could be studied in field, in a physical model or be simulated by means of computer calculations.

In the beginning of this decade equipments for continuous monitoring of ventilation were developed at the National Swedish Testing Institute and the Swedish Building Resaerch Institute. Both systems use a tracer gas and can be run in three different modes: constant concentration, constant flow or decay of tracer gas. Nine different rooms can be monitored simultaneously.

Quite a lot of measurements, combined with introduction of different air indices, such as air change efficiency, ventilation efficiency etc., have been carried out by M Sandberg at the Swedish Institute for Building Research. Most of the measurements have been carried out in a full scale apartment in the laboratory at the institute. There are many of reports, papers etc. describing this. A good summary and state of the art report is published in Swedish, Sandberg & Skåret(1989). This report will soon be published in English by the AIVC.

At Chalmers Institute of Technology, valuable additions to the knowledge of air flow fields in rooms have been brought about by means of computer calculations. Davidson(1989). The calculations are based on a modified k-epsilon turbulence model treating also complex geometries and low Reynolds number flow.

3.3 Ventilation and air infiltration

Ventilation and air infiltration must be considered simulaneously in order to reach a complete understanding of the total thermodynamical and indoor climatical behaviour of a building.
In order to study this system-behaviour, (computer) simulations and field measurements are essential components.

Different computer programs for such analyses have been produced in Sweden during this century. The latest and most promising one, MOVECOMP, Herrlin(1988), is currently used by several researchers, e.g. Blomsterberg(1989) who also compares the calculation results with measurements.

The Swedish researcher in the COMIS-group is responsible for the flow calculation algorithm.

4 NATIONAL REGULATIONS AND STANDARDS

4.1 Air tightness

For a building as a whole the Swedish Building Code(1988) states that the air tightness of the building envelope, tested in accordance with the Swedish standard SS 02 15 51, must not exceed 3 m3/(m2h) for dwellings and 6 m3/(m2h) for other buildings. No specific demands regarding the air tightness of different building components are included in the building code any longer.

For ventilation equipment and -ductwork there are certain air tightness demands in the Building Code.

4.2 Ventilation

Regarding air exchange the following is quoted from the Swedish Building Code(1988):

A room should have a continous air exchange. This should be arranged so that emissions from people and building materials, moisture, air pollutions, odour and health affecting substances will not be accumulated.

The flow rate of outdoor air into rooms with normal height, where people are present more than occasionally, should be at least 0.35 l/(sm2 floor area). For dwellings, this demand regards the dwelling unit as a whole as well as the different rooms. Rooms demanding a higher degree of ventilation should have at least the ventilation capacity shown in the table below. Spaces where people are present only occasionally should have a ventilation in such a manner that no health risks occur and that damages to the building and it's installations are prevented.

Table (shortly abbreviated)

Dwellings, hotels etc.	
Bedroom	4.0 l/(s*person)
Kitchen	10.0 l/s
Kitchenette	15.0 l/s
Bath room	
with openable window	10.0 l/s *)
without "- "-	15.0 l/s *)
WC	10.0 l/s
Laundry room, drying room etc	10.0 l/s *)
Workplaces, conference halls, shops etc.	
Rooms for sedentary work	5.0 l/(s*person)
Rooms for mobile work	7.0 l/(s*person)
Rooms where smoking could be expected	10.0 l/(s*person)
WCs	15.0 l/(s*WC-unit)
WCs for the public	20.0 l/(s*WC-unit)

If the floor area is greater than 5 m2, the ventilation should be increased by 1 l/s for each m2 exceeding area.

5 PROSPECTS FOR THE FUTURE

General development:

Due to the general interest in the country regarding indoor air quality it could be expected that:

- routines for quality assesment and -control of air-handling systems will be established and used.
- the control of building materials and other possible pollutants indoors will be intensified.
- ventilation-, heat recovery- and heating systems which are more energy efficient and reliable from performance point of view will be developed.
- the technology for control, service and maintenance will be more adopted to man than is the case today.
- the demands on good performance and efficiency of ventilation in individual rooms will increase.
- the demands on airtight building envelopes will increase.

Research:

Research fields likely to be dealt with in the future are:

- Development and use of techniques for computer simulations of air flow patterns in rooms and between rooms in a building, thus providing a better understanding of ventilation performance.

- Air quality studies on emissions in the indoor environment, human ventilation demands, feasibility of different ventilation strategies and -equipment.

- Moisture convection, it's dependence on indoor over-pressure, air tightness and moisture in indoor air.

- Measurement technique. Development of refined tracer gas techniques.

- Development of heating and ventilation strategies for energy- and other crises.

Industrial and/or scientific development:

Important areas which regard attention are:

- Development of equipment for automatic control and supervision of complicated house installations. (Smart houses)

- Developments of better manuals, hand books etc. for heating and ventilation equipment.

- Development of equipment for human and technical demand control for heating and ventilation.

- Development of less expensive and reliable warm air heating systems for houses.

- Development of electricity efficient heating and ventilation equipment

- Development of outdoor supply air ventilation devices causing less discomfort (especially in the winter time) than existing ones.

- Development of hard- and software for tracer gas measurements.

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PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

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Poster 23

INFILTRATION AND VENTILATION IN SWITZERLAND - PAST AND FUTURE

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1. CLIMATIC CONDITIONS, BUILDING CONSTRUCTION AND USER NEEDS, AN INTRODUCTION TO VENTILATION IN SWITZERLAND

The Swiss, it is said abroad, all yodel and are closely aligned with agriculture (Swiss cheese). This stereotype represents only a small part of the population, whereas ventilation by means of windows (in residential buildings) is in fact a national characteristic. Critics call the Swiss ventilating customs "random ventilation" ("Zufallslüftung") and this expression is in fact on target. It is regretable that the populace still have no means on hand to overcome this randomness. In any case, there are a few points which will be reported here. Also regretable is the fact that in the last 20 years, window constructions with finely adjustable sash - such as the sliding unit pictured here (part of a casement window with two panes in series) - have been replaced with large side or bottom hinged sash. The following observations are, as a rule, applicable for residential construction.

Why has the Swiss random ventilation system not been replaced? The answer lies in the fact that, for a large part of the Swiss people and for the greater part of the year, the amount of air changes in a room is not as important as it is in other places. This can be explained by several facts:

- Very cold days occur very seldom so that the heating systems, which are often dimensioned with a generous safety factor, can meet the heating demand even if one or the other window is left open too long.
- Very hot days, when window opening at night would improve comfort, occur seldom.
- The buildings were better insulated than was the case in neighboring countries, with the result being that mildew problem occurs less often.

Naturally, there are extreme climate situations, also alpine regions, which are much colder and more wind exposed than the average. There, as a rule, the building occupants understand the appropriate behavior regarding controlling ventilation (e.g. a protection zone before the house entrance door, the closing of shutters and the weatherstripping of casement windows). The Swiss climate data and construction guidelines were written up and illustrated in detail in the first AIVC

Handbook (the Swedish Handbook). In this regard a few summary statements are appropriate.

For the majority of buildings:
- massive construction is the rule,
- in alpine regions post and beam construction also occurs,
- prefabrication is rare,
- double glass windows have been common for a long time.

Regarding residential ventilation:
- older houses had natural ventilation shafts,
- bathrooms and toilets as well as kitchens had windows.

2. DEVELOPMENTS IN THE DESIGN OF BUILDING COMPONENTS AND VENTILATION SYSTEMS DURING THE LAST DECADE

The extensive efforts to save energy and the multitude of ensuing energy saving guidelines have lead to the drastic reduction of heat losses due to transmission through the building envelope. Thereby, there has been a drastic increase in the proportion of the heat losses caused by infiltration. For all building surfaces the U-value has been reduced, while the construction in general and windows in specific have become increasingly airtight. This has led to the fact that, for massively constructed buildings, the air change rate is too low. In light construction the previously extreme leakiness has now been reduced to an acceptable level. (Also, noteworthy is the growing trend to convert attic space into a habitable room.) By contrast, methods of providing ventilation have not benefited from any apprecia- ble improvements. It is true that mechanical ventilation has become increasingly common for interior bath rooms, toilets and for kitchens. There is in general, however, no overall ventilation concept for houses (for example, the conscious placement of ventilation openings, including adjustable sash or dampers or an indicator which advises: "Please open the windows now because it has become too humid").

In parallel to these construction developments of the post energy crisis era, a small group of designers and building physicists have pursued interests regarding questions of air quality, the end effects of inappropriate ventilation and the consequences of a leaky building envelope. Their investigations have been supported through the AIVC and through intensive association with researchers abroad. As a result of these efforts the following has been possible:

- The actual air exchange situation in our buildings has been determined.
- Dependable measurement methods and equipment have been developed and tested (airtightness, air change rates, and locating leaks).
- Efforts have been initiated to develop Swiss design guidelines regarding airtightness and ventilation.
- Interest has grown in developing planning tools and systems adapted to the conditions of the future.

We shall report on recent developments in Section 3 and particularly in the report on research results in Section 4.

3. OUTLOOK, SUPPORT OF RESEARCH ACTIVITIES

The authors are of the opinion that for a large number of Swiss buildings a major change in the means of ventilation will follow. This change will be of the same magnitude as the drastic increase in the insulation of the building envelope or as the halving of window transmission losses. What are the underlying causes for such a major change. A few key aspects can be noted:

- The air exchange rate in most existing buildings is known to be too small (basic air change rate) with the result being too high concentrations of CO_2 in unventilated bedrooms.
- The ever increasing noise burden has led to rulings to subsidize the installation of acoustic windows; other measures have not followed in parallel.
- In new buildings materials are used which emit odors/ pollutants. In combination with tight windows and random ventilation the threat of health hazards of many sorts is very real.
- From experiments in other countries and from a few case studies in Switzerland new types of solutions to these problems are known. These have an economically viable outlook, and promise reduced energy consumption and good indoor air quality.

- Finally, building occupants have become more critical. The "sick building syndrome" and "Wohngifte" have prompted industry to find innovative solutions to these problems.

What distinguishes these new solutions? At least three major steps towards the future can be identified:

- Two years ago an extensive basic research program (ERL) was started (see paragraph 4).
- An intensive effort in writing design and measurement standards has been instigated (Standards concerning ventilating equipment; building regulations judging the necessity of ventilation, cooling and humidification equipment; and guidelines for on site measurement).
- Finally multiple steps have been taken concerning systems development, including work done in the context of:
  - the ERL-Programm, particularly commercial buildings,
  - the IEA Annex 18 concerning demand ventilation in general (active Swiss participation),
  - a series of Swiss projects ("Impuls Programs") in regard to ventilation in apartments; mechanical ventilation with and without occupant involvement should be tested (particularly humidity regulation systems),
  - a planned research project involving ventilation of large volume rooms (e.g. atria and halls).

If and when these developments will occur on a large scale is a matter of speculation. In a broader context, a push for better controlled ventilation will occur with the opening of the European Market in 1992. At that time manufacturers with large series production will create a pressure on the Swiss market. Hopefully, the products which then appear will not embody mediocrity as a result of market pressures, but rather will fulfill their function with superiority.

4. ENERGY RELEVANT AIR MOVEMENT IN BUILDINGS

Interim Report, Status June 1989

4.1 Introduction, Goals

Starting point, organisation

In 1985 the research program "Energy Relevant Air Movement in Buildings" (ERL) grew out of the concept for an energy policy developed by the Swiss Directorate of Education. In 1986 this concept became a reality, addressing two fundamental issues: the transport of air and contaminants within a building and the exchange of air between a building and the outdoor environment. To deal with these issues three subtasks were defined:

- Subtask A: Air and contaminant flow within a room (intrazonal air movement)
- Subtask B: Air and contaminant flow between rooms and to the outdoor environment (interzonal air movement)
- Subtask C: Ventilation systems of the future.

The interdisciplinary nature of the program requires the cooperative efforts of several institutes of both Swiss Federal Institutes of Technology in Zürich (ETHZ) and Lausanne (EPFL), the Swiss Federal Laboratory for Materials Testing and Research (EMPA), several industry firms and various engineering firms. The scientific work, organized into Subtask topical areas, was divided into individual projects. Financing of these projects has been provided from five sponsors: The Swiss Federal Office of Energy (BEW), the National Energy Research Foundation (NEFF), the Commission for the Advancement of Scientific Research (KWF), the National Fund (NF), and the Directorate of Education.

In the following two chapters the problem area of this work is presented along with the goals which were defined at the inception of this program. Thereafter, the present status of the work is reported.

The Problem Area

The air flow within a room, the interchange of air between rooms and the interchange of air between a building and the exterior are topics which until today have received little attention by researchers. The processes are three dimensional, time dependent and take place under complex conditions (weather, occupant behavior and spatial definition). The design of an HVAC system today must draw principally on experience, taking into account all of the above mentioned uncertainties and including conservative safeguards. Today a key concern is to achieve low energy consumption while affording comfort of the occupants. However, more knowledge is needed regarding the mechanisms of airflow, the associated heat and material transport (contaminants) and the criteria for judging the feeling of well-being in a room. The currently available numerical simulation and measurement techniques promise exciting opportunities to deepen our knowledge in these areas of air movement. This improved understanding can then be applied in the planning of buildings and HVAC systems with the end result being the development of more economical systems, better indoor air quality and increased confidence in the performance of buildings and systems optimized for low energy consumption.

Goals

The designers of buildings and HVAC systems should be provided the means of evaluating all the relevant parameters regarding air movement in buildings in order to assure:

- The occupants' well-being, comfort and safety.
- A suitable and economic system for ventilation and heating.
- An extensive use of insolation and internally generated heat.
- An optimal use of purchased energy.

The Swiss conditions of climate, weather and building construction will be given special attention throughout the program. The following steps are envisioned:

- Development of a mathematical description of the air and contaminant transport within a room and between different zones of a building.
- Provision of a method of measuring velocity of air movement and its variation, air temperature and the concentration of contaminants.

- Clarification of which ventilation and heating systems are best suited for different building types (occupancies).
- Translation of the results into design tools, which when possible do not require a mainframe computer.

This overall area of concern can best be addressed by dividing the subject area into three subtask areas:

- Subtask A: Air and contaminant flow within a room.
- Subtask B: Air and contaminant flow between rooms and to the outdoor environment.
- Subtask C: Ventilation systems of the future (for various building types/occupancies).

The goals of these and the interim status of the work are reported in the following section.

4.2 SUBTASK A: Air Movement and Contaminant Flow Within a Room

The work of this Subtask should provide the means of calculating those air movement parameters which are key to assessing the energy and comfort performance of a ventilation and heating system. Also, measurement methods for determining the air flow processes, either in a physical model or on site in a building, should be refined. The results will also be so formatted as to serve as input data for defining the single room conditions to be used in Subtask B. Further they will serve as a basis for the investigation of concepts for future HVAC systems conducted in Subtask C.

In the now completed first step of the project, the suitability of the finite difference method (FD) of calculating air movement in rooms was investigated at the ETHZ. To validate the algorithms of the program package PHOENICS measurement data from well defined test cases were used. These test cases were conducted in the test chamber of the Company Sulzer Brothers using a laser-doppler measurement system (LDA). It was thereby shown that both two and three dimensional problems can be easily defined and solved using PHOENICS. For laminar air flow, as a first estimate, good results were also obtained. This was also true for isothermal, momentum driven, turbulent air flow. If the thermal buoyancy becomes prominent, it would be necessary to extend the turbulence algorithm (k-ε-model) in PHOENICS. For this purpose validation efforts are now in progress.

In addition the firm Sulzer used AIRCOND (an FD program) to calculate heat transfer for given temperature conditions. For the anisothermal case of free and mixed convection, improvements in the model had to be made. Also, for the determination of heat transfer coefficients by free buoyancy modifications had to be made to the model. Further modifications improved the convergence and the calculation run time of the program. The greatest promise for further development lies, however, in hardware advances (parallel processing).

In general it was determined through comparisons of computed and measured data that good agreement occurs in the case of isothermal momentum driven air flow and acceptable agreement occurs for anisothermal buoyancy driven air flow. Model improvements are, none the less, necessary. This work comprises a part of our contribution to Subtask 1 of the International Energy Agency (IEA) Project "Air Flow Patterns within Buildings". Through this cooperative effort essential support and contributions were also received.

A major goal of the work in this Subtask is the delivery of simplified calculation procedures for designers. These tools deliver results of acceptable accuracy at substantially less cost. For this purpose first ideas have been developed.

In parallel to the above mentioned work with FD methods, a team at the ETH-Lausanne will develop a program using the finite element (FE) approach. The method will be checked against the same test cases which should promise to be an interesting comparison concerning possibile applications of these two approaches. The special advantages of the FE method are expected by non cartesian geometries as well as by the calculation of air flow details (e.g. air entry zones, air flow around people and equipment, etc.). This work is now in a beginning phase.

The glass fiber LDA apparatus, adapted by the firm SULZER for this project, has proven to be very useful. By means of this non-disturbing optical method instationary air flow with very small air movement velocities and a high degree of turbulence can be investigated. The method allows measurement of a spatial air flow field on a point basis by determining the components of the velocity vectors from two directions. Thereby, the method proves to be very appropriate for the validation of computer programs which calculate air movements. As a portable device for conducting field measurements the method is less well suited because of its complex construction. The volumetric results discussed until now fulfill the laws which were expected. For determining the instability of air flow with very low frequencies (approximately 0.1 Hertz), the system should be expanded to allow the simultaneous measurement of all three velocity components.

As a compliment to this technique a tracer imaging technique will be developed to help visualize and quantify a complete air flow field. This work is to be done outside the scope of the ERL Project at the ETH in Zürich. The method is based on computer supported image processing of tracer tracks (soap bubbles) in a darkened room. The tracer movement is made visible by means of a special optical approach (light sheet method). This approach could, in its principal mode of operation, be readily validated. The construction and testing of a portable apparatus is currently in progress.

4.3 Subtask B: Air and Contaminant Flow Between Rooms and to the Outdoor Environment

The original goal of the Subtask B is still valid, namely, to develop a simplified calculation program to predict the air and contaminant flow between rooms and between a room and the outdoors. For the occasion of the last "Swiss Status Seminar" in 1988, the latest possibilities for using such a program for evaluating variations of a buildung and its equipment were definded. Also, the grouping of the work remains as originally planned, namely:

- Program development
- Definition of a suitable measurement technique and collection of data for program validation.
- Working out detailed physical phenomena (interaction of wind, thermics; characterization of the occupant behavior; preparation of climate input data recognizing the specifics of the topography and built-up environment).

Changes have occured, however, in the leadership of the Subtask. J. Hertig replaced the Subtask initiant and original leader, Ch. Zürcher. This was neccessary because support of the activity was lacking on the part of the ETHZ. This left a critical gap in the topical groups: detailed physical phenomena and program development. It remains to be seen to what extent the consequences will be merely a delay in the time schedule or an adjustment of the responsiblities of the group. The situation is somewhat less critical due to the international cooperation in these activities (COMIS and the IEA Annex 20). The status of the Subtask as of the end of 1988 can be described as follows:

- The specifications for the computer program have been coordinated with the COMIS project.
- A phase of joint work between the Swiss principal researchers and the COMIS team has been fixed.
- The partially completed work on statistical description of the behavior of occupants was begun and led to an overview of available measurement data and to a first model.

Only minimal delays have been suffered in the evaluation and correction of ANETZ wind data. The publication of a report and calculation rules is expected by the end of 1989. In addition, supporting tests in a wind tunnel were carried out upon completion of the necessary topographical dummy forms. As planned, the preliminary study for the planned research on thermal kinetics and air exchange in buildings was completed. A detailed plan for the whole project is now available. Also completed is the development of a tracer gas method CCGT (constant concentration with the use of three gases) for the investigation of air exchange and contaminant transport. Meanwhile, the work on the EDA-Method (PFT-Tracer-Method; constant emission, absorbers) is still underway. An apparatus for determining the air permeability or airtightness of buildings or zones within buildings is now operational at the EPFL/LESO.

The team of the EPFL/LESO has determined with great care all the key parameters of the LESO building and for selected periods quantified the air change rates. These reference data sets should prove useful for the COMIS calculation program as well as for the future variations of the ERL multizone computer program. A sensitive hole in the construction of the ERL multizone program is the not yet completed structuring of the Swiss building population and accordingly the appropriate modification of the wind fields in the close proximity to the building. For 1989 several preliminary studies are planned, but it will be difficult to allocate the priorities among the experienced researchers.

4.4 SUBTASK C: Ventilation systems of the future.

The goal of the work of this Subtask is the advancement of the development of future-oriented heating, ventilation and cooling systems for buildings with air as the transport medium. The project encompasses building types of various categories: residences, schools, office buildings and industrial buildings. The focus of the investigations lies by commercial buildings (defined to include office buildings, schools, banks, etc).

In the first phase of the project, a comprehensive data survey, literature review and questioning of "opinion leaders" (researchers, manufactures, system designers, contractors and installers) and real estate owners has been assembled and will now be evaluated. Thereby, a realistic picture of the status of the technique and the capabilities of the different systems should be possible. In addition, new insights about the necessary improvements of the planning and dimensioning material should be gained. Lately, it has become common to expect a high degree of thermal insulation of the building envelope, increased air tightness of construction (energy conservation measures) and escalating room air quality and comfort. One particular need of designers concerns the optimization of the ventilation system for different building types. Designers need answers regarding necessary fresh air quantities, air quality and energy balances. Consideration must also be given to improved controls responding quickly to changing loads. Finally better understanding is needed of room air movement, internal heat sources and occupant behavior.

On the experimental side, a preliminary (qualitative) investigation of new systems has been started as case studies. Here, smoke tests in rooms will be used to determine the air movement behavior in rooms. The quantities of air supplied and exhausted, humidity, CO_2 concentration and relevant temperatures will also be determined.

An additional area of investigation concerns the free and forced convection in an apartment building. Concerning this latter subject, interesting results have already been obtained on comparisons of the air change rate by natural and by mechanical ventilation, and on the influence of occupant behavior on the thermal energy balance. In the ongoing measurement phase, moisture measurements and CO_2 measurements are being conducted. From these results it should be possible to obtain a rough picture of the energy and contaminant transport in rooms with natural ventilation.

A final report on the results of the first phase of the project should become available shortly. Besides a description of the present state of knowledge on ventilation systems, the report contains material on new development tendencies in Switzerland and abroad, the needs of building clients and designers and experiences with newer systems (case studies). Through the joint work started in 1988 with the IEA Annex 18 "Demand Controlled Ventilating Systems" the necessary international connections have been provided. This cooperative effort brought, in addition to an overview and analysis of sensor technics, experience in the measurement of characteristic parameters for determing the ventilation efficiency of systems.

With the implementation of an advisory panel from industry, a vital link to practice has been achieved. This is of great importance for the practical application of research results. The concept for disseminating results for Subtask C is determined together with the participants of IEA Annex 18 and the potential users in Switzerland.

In the second phase of the project a test chamber will be used to clarify uncertainties of systems and to test new systems. Of primary interest in these investigations will be pure air systems (all-air) and systems with separated ventilation and heating/cooling components. To be determined is the influence of the major parameters of the systems (the system use conditions as well as the building use) upon the deployment, economics and energy savings potential of the systems. The behavior of the systems by practical uses should be examined. These investigations will concentrate on commercial buildings in the larger context of this definition. Ventilation of apartment buildings and special ventilation systems in the industrial sector shall not be considered.

4.5 Outlook

After completion of the first phase, necessary modifications to the original enthusiastically set goals have become evident. In Subtask A the interim goals were most closely achieved regarding the development of numerical methods. Here, the work is also the most advanced. However, the development of measurement methods and the preparation of relevant data for design materials must be reduced. For example, the LDA method for the point measurement of air flow proved to be an outstanding for the validation of computer codes, as a portable apparatus for application in field measurements it proved, however, inappropriate. Limitations were reached in the calculation of air flows in complex large spaces, in the determination of emission and absorption data of building materials (transport of contaminants) as well as of the propagation of specific heavy air contaminants.

In Subtask B problems appeared by the accurate determination of pressure distribution in border zones of single rooms/ subparts of rooms. Also problematic is the determination of the reference data (temperature and pressure), as they are influenced by many factors such as the surroundings of a house.

In Subtask C gaps exist in the planned field measurements as a consequence of lacking measurement techniques. It will have to be determined in detail to what extent these will affect the output of the project. As regards to investigation by building types, it is clear that in Switzerland the main interest lies by commercial buildings (office buildings schools, banks, hotels, etc.). In the industrial sector the requirements of a ventilation system are so different and specific (often for extreme circumstances) that practically a special solution must be found for every single situation. Therefore, the ERL-C Program will concentrate on commercial buildings. The focus of the third phase of the subtask will be the dissemination of the results for use in the practice. Scientific reports and data bases for future research projects and international collaboration will be prepared by the researchers. But, of specific importance are the directly useable products of the work, namely, well documented calculation and measurement methods, and a sourcebook. These materials should then be integrated into a handbook for designers. Because this document is aimed to serve designers, designers must have an influence during its creation. In all of the above the interests of several target groups must be taken into account. These groups have been defined as:

- Researchers (national and abroad)
- System designers (in the larger context)
- Architects and their clients
- Schools (University and technical schools)
- Software firms (producers and marketers).

How the newly gained results and directly useable products are transferred will be determined in discussions with the different target users.

4.6 Examples of results from each subtask

Subtask A:
"Test case with natural convection"

3-D Laser-Doppler Anemometry (LDA) measurements have been conducted in a test chamber (below) to evaluate the performance of the flow simulation code AIRCOND in predicting natural convection.

Non-isothermal Flow with Free Convection

Measured quantities:
U, V, W, u-rms, v-rms, w-rms, uv, uw

Seeding:
Paraffin oil droplets = 0,5-2μm

Data rate:
5-50 [1/s]

Averaging over:
ca. 3000 Samples

Accuracy:
Velocity: ±1% - ±10%
flow angle: ±3%
Position in the room: ±10mm

Measuring time:
2D: 10 min
3D: 25 min

Number of measurement points: 330

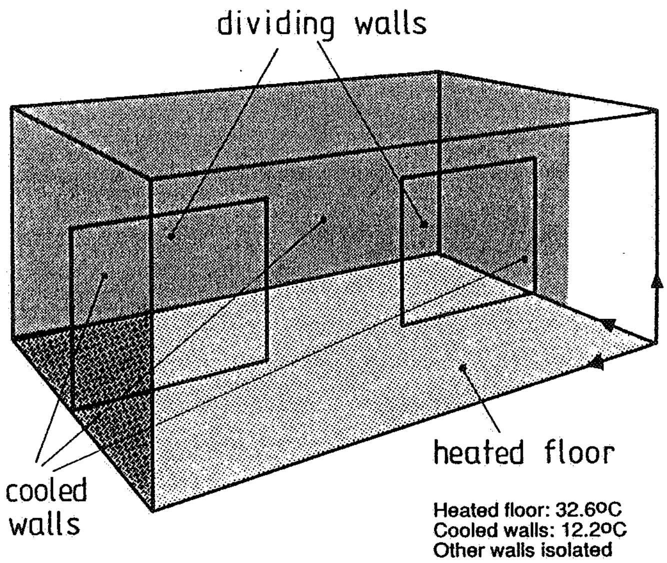
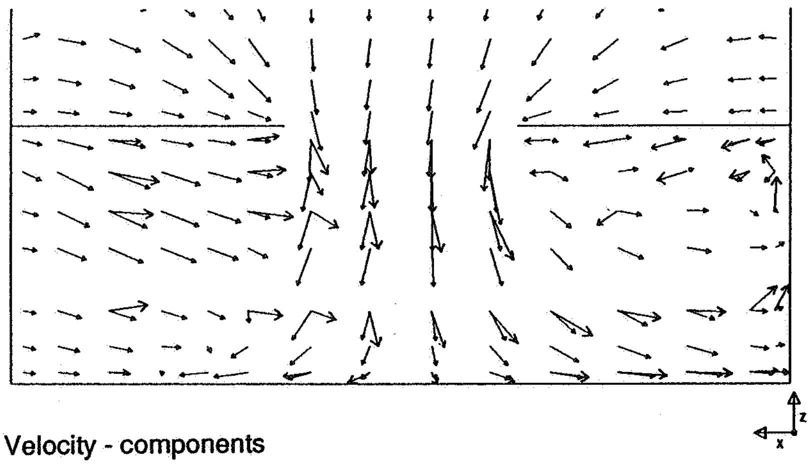


Figure 1: Test case and its characteristics

Non-isothermal Flow with Free Convection



Velocity - components

0,25m / 0,15m/s

X - Z plane Y = 0,10 m

→ measured value

→ calculated value (by "AIRCOND)

Figure 2: Comparison of measured and calculated airflow

Comparison of the velocity field in a horizontal plane 10 cm from the floor of the room: computation (black arrow heads) agrees well with measurements (open arrow heads) for the cold flow between the dividing walls but is in error in the re-circulating zone at right. AIRCOND* uses the standard k/ε-Model and empirical correlations for natural convection heat transfer.

* (AIRCOND is distributed by AVL, Graz)

<u>Subtask B:</u>
<u>"Stochastic analysis of users behaviour in regard to ventilation"</u>

Goal:
To develop stochastic models of user behaviour regarding ventilation using the theory of probability. These models will be able to generate synthetic time series of window and door openings whose characteristics will be similar to reality. They will be integrated in to air ventilation simulation programmes to improve their accuracy and realism.

Method of work:
- Collect the appropriate monitored data regarding window and door openings (LESO test building)

- Determine the most appropriate influential variables
- Develop stochastic models using simple stochastic processes (Markov chains)
- Apply the models to different rooms and users behaviour (direct gain facade, high insulation technique)
- Validate the models via comparison with reality
- Publish a short description of the models and their application procedure.

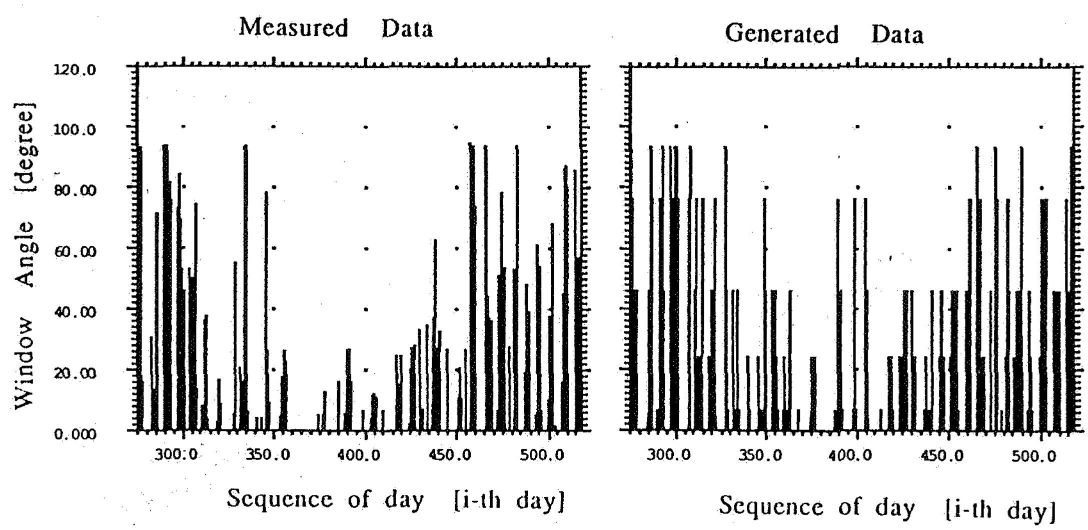


Figure 3: Measured and generated data

Real and synthetic time series of window angles (LESO test building, winter 1984-85, direct gain room west). Synthetic data are discretized for purposes of stochastic modelling, showing a less "continuous profile" then real data. Impact of this on simulation results is not significative.

<u>Subtask C:</u>
<u>Investigations on 10 built examples</u>

Goals of the case Studies

- Overview over modern ventilation systems for office buildings and dwellings in Switzerland
- describe the behaviour of systems under real working conditions
- determine air quality and thermal conditions in the room
- visualize air flows in the room

Investigated system types

1. Complete mixing
   - induction
   - fan-coils
   - ceiling outlets (jets)

2. Displacement ventilation
   - low-level diffusers
   - ceiling diffusers
   - floor mounted twist outlets

Investigation Programme

1. Global building description

2. Building construction

3. Main ventilation system
   - air conditioning
   - air distribution

4. Local ventilation system
   - principle of ventilation
   - supply and exhaust devices
   - room conditions (air quality, thermal comfort)

5. Control and flexibility

6. Influence on the joice of the system
   - contractor
   - architect
   - engineer

7. Measurements
   - air movements
   - temperatures
   - CO_2
   - humidity

8. Smoke trials
   - visualization of the air flow patterns

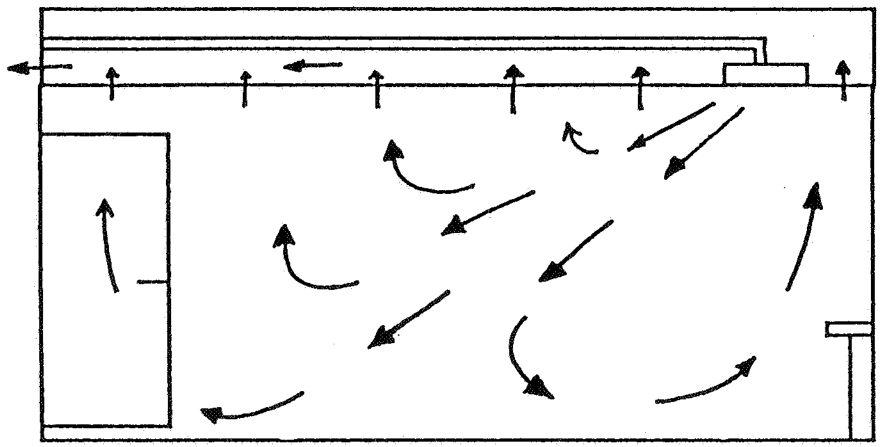


*Fig. 4:Typical air flow pattern determinded by smoke visualization
System type: Complete mixing / ceiling outlet*

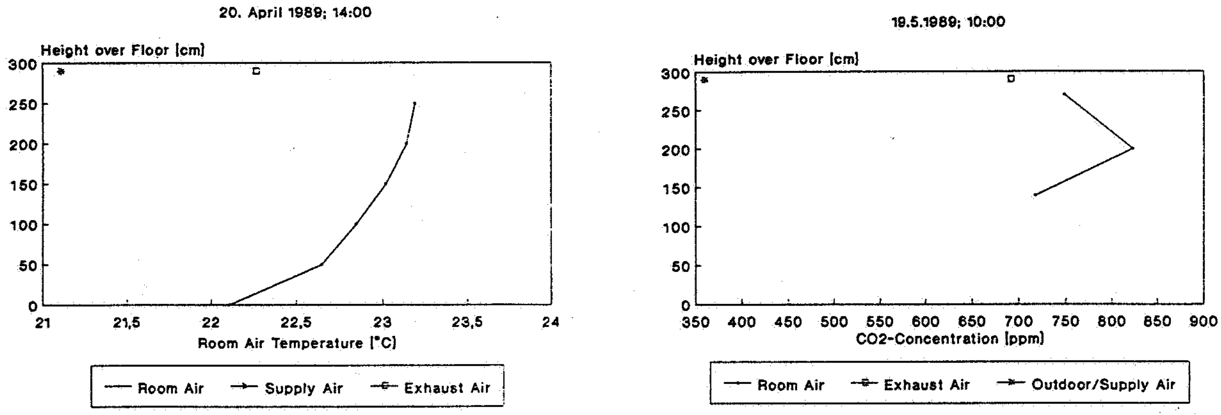


Fig.5: Typical measurement results
System type complete mixing / ceiling outlet

4.7 Relationship of the ERL Research Programme with the Plans of Other Countries.

In the last year and also today there has been a major effort on the part of the Swiss to start and participate in appropriate international projects. One example is the initiative for a first exchange of experience concerning air exchange which took place in a seminar in 1968. Another example is a first effort for a "window ventilation project" which was followed by a real project start 5 years later.

At the present time four projects, in which the ERL team is involved, are in the foreground:

- The Air Infiltration and Ventilation Centre (IEA "Buildings" Annex 5).
- The "IEA Buildings Annex 18", "Demand Controlled Ventilation Systems".
- The "IEA Buildings Annex 20", "Air Flow Patterns within Buildings".
- The LBL instigated project "COMIS".

It is evident that all ERL researchers can profit from the basic work and publications of the AIVC; that Annex 18 is complementary to ERL Subtask C, that ERL Subtask A profits from support by Annex 20. Finally the work in Subtask 2 of Annex 20 as well as the work of COMIS are well linked to the work of the ERL Subtask B. It commands from the researchers, however, an intensive concentration and common thought process at international project meetings. In this context it is possible to list products desired from this planned work, which are not unrealistic:

- From the AIVC: a clean list of definitions about ventilation efficiency. An updated version of the measurement technique guide and the editing of a new version of the calculation guide.

- From Annex 18: narrowly grouped, strictly conceived field tests about demand controlled systems.

- From the participants of the COMIS Program: a good description (source book) of their results.

4.8 Publications

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[4] Roulet C.A., Scartezzini J.L.: "Measurement of air change rate in an inhabited building with a constant tracer gas concentration technique", ASHRAE Transactions 93/1 (1987), #2882

[5] Scartezzini J.L., Fürbringer J.M., Roulet C.A., Feustel H.E.: "Data needs for the purpose of air infiltration computer codes validation", Proc. of 8th AIVC Conf., Ueberlingen, FRG, Sept. 1987

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[7] Scartezzini J.L.: "Programmes de calcul détaillés et simplifiés des échanges d'air dans le bâtiment: validation à l'aide de mesures expérimentales", Statusseminar ERL, Winterthur, Nov. 1987

[8] Roulet C.A.: "Energierelevante Luftströmungen in Gebäuden", 3. Weimarer Symposium, Weimar, DDR, 1988

[9] Zollinger H.J.: "Energierelevante Luftsströmungen in Gebäuden", Technische Rundschau Sulzer, 2/1988

[10] Roulet C.A., Compagnon R.: "Multizone tracer gas infiltration measurements interpretation algorithms for non isothermal cases", Building and Environment, August 1988

[11] Eggimann J.P., Gay J.B.: "Premiers résultats obtenus sur les logements collectifs solaires à Préverenges", Statusseminar Energieforschung im Hochbau, EMPA-KWH, Sept. 1988

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[17] Widder F., ed.: Forschungsprogramm "Energierelevante Luftströmungen in Gebäuden", Tagungsunterlagen 2.Statusseminar, Lausanne, Nov. 1988

[18] Leuzinger R.: "Energierelevante Luftströmungen in Gebäuden", Heizung und Lüftung 56/1 (1989), p.18-19

[19] Van der Maas J., Roulet C.A., Hertig J.A.: "Some Aspects of Gravity Driven Air Flow through Large Apertures in Buildings", ASHRAE Transactions 95/2 (1989)

[20] Roulet C.A., Compagnon R.: "Multizone Tracer Gas Infiltration Measurements-Interpretation Algorithms for Non-Isothermal Cases", Building and Environment 24/3 (1989) 221-227

<u>Remark:</u> The annual progress reports of the particular subprojects are published in [6] and [17] and not quoted here. Copies of [6] and [17] are still available and can be ordered by Dr. F. Widder, Paul Scherrer Institut, CH-5232 Villigen PSI (Switzerland)

PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

10th AIVC Conference, Dipoli, Finland
25-28 September, 1989

Poster 24

NEW DESIGN OF CENTRAL UNITS IN AIR HEATING SYSTEMS FOR HEATING AND VENTILATION IN DOMESTIC BUILDINGS

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NEW DESIGN OF CENTRAL UNITS IN AIR HEATING SYSTEMS FOR HEATING AND VENTILATION IN DOMESTIC BUILDINGS

Synopsis

In central units of air heating systems the supply air flow must meet the actual heating demand. Most of central units for air heating systems have only one fan, which is designed for the maximum air flow at the maximum heating capacity. Fan motors are designed for variable-voltage-drive to enable a reduction of air flow to the necessary value at different heating demands. However, the electrical efficiency is decreasing strongly. The supply air fan is working mostly under part-load conditions. Therefore the control strategy used now is very ineffective. It is suggested to install 2 fans in an air handling unit. One of them is working at full capacity all over the year, but the second one starts only to deliver the additional air flow required in case of higher heat demand. Charts basing on calculations will demonstrate the relation between outdoor air temperature and the demand of energy for both systems.

1. Introduction

By advanced insulation standards of buildings and development of window-constructions the heating demand for building decreased strongly. Nevertheless it is necessary to exchange a certain rate of indoor air: pollutant concentrations and vapour have to be kept below a certain level to avoid sickness of persons or damage of the building construction. Some years ago gaps in walls and window-components provided sufficient air infiltration to minimize pollutant concentration. Modern building constructions however require intensive ventilation by opening windows or better by mechanical ventilation. Several researches have been finished with the conclusion, that an air exchange rate of 1 h^{-1} up to 1.5 h^{-1} should be reached. One very important advantage of mechanical ventilation is the possibility of heat recovery. This method helps to reach a considerable potential for saving energy as shown in figures 1 and 2 by the increasing percentage of ventilation heat demand.

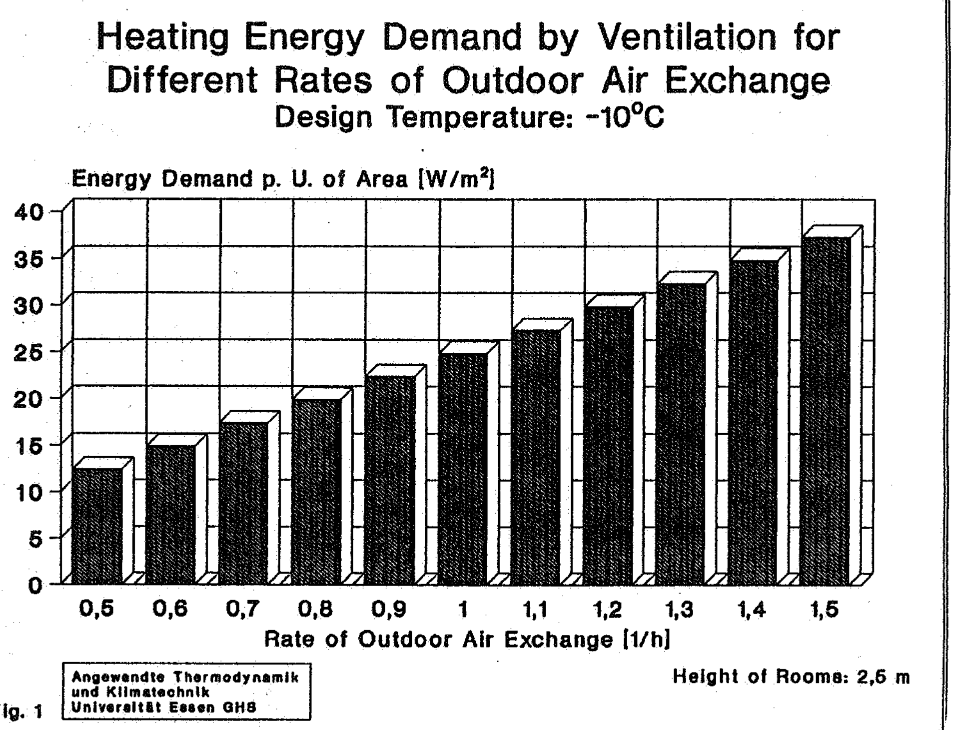

Fig. 1

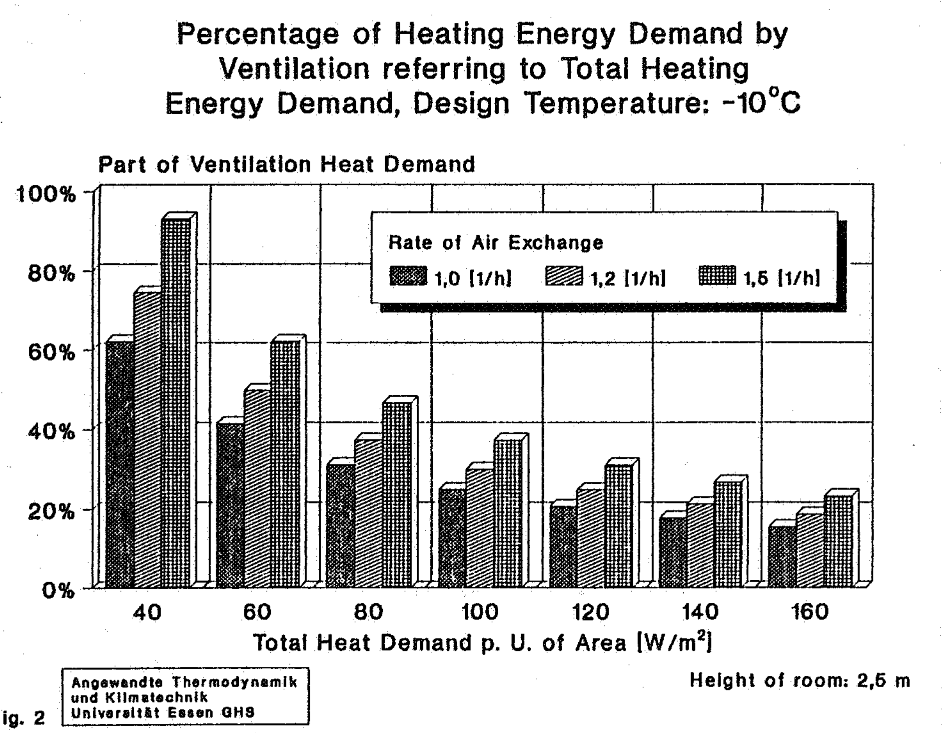

Fig. 2

In order to reduce first costs the combination of heating and ventilation systems is quite obvious. Without any additional water circulation system the supply air will be used to transport heating energy to the rooms. According to the above mentioned range of air exchange the following calculations base on 1.0 h^{-1} as the minimum value. Considering the comfort limits concerning the maximum supply air temperature (55°C just at the central unit) air heating system provide an energy transport of 29 W/m^2. The heat demand for the outdoor air change is delivered directly in the central unit to the air flow. It is not necessary, therefore, to transport this heat to the rooms (See figure 3). Only in buildings with a heat demand by transmission above 29 W/m^2 the transport capacity of ventilation air rate is not sufficient: it is necessary to increase the transport capacity by return air.

System Design

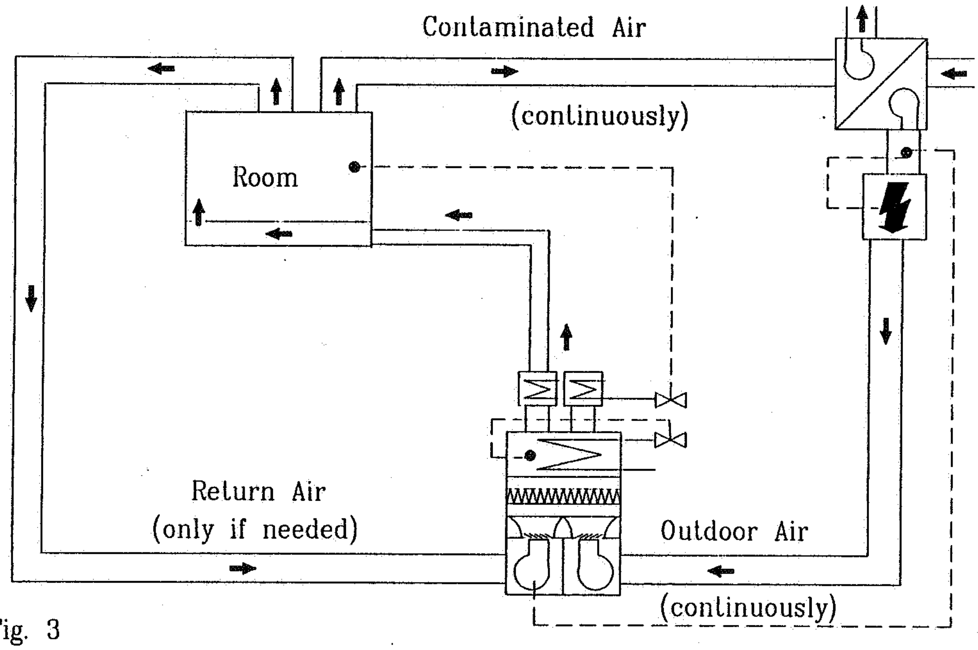

Fig. 3

2. Conventional Conception of Control Strategies

For those systems in buildings with a transmission heat demand below 30 W/m^2 the supply air temperature for each room can be modified by separate heat exchangers. An adaption of the heating capacity by reducing the supply air flow to the rooms has to be declined as waste of energy. Moreover, the ventilation function of the system is restricted by reducing the supply airflow: this is an obvious contradiction to the desired equivalence between heating and ventilation combined in one system.

In buildings with higher heating demands the maximum supply air temperature will be reached at outdoor air temperatures above the heating design-temperature. This will actually happen while reaching an outdoor air temperature range between -4°C and 0°C in buildings with an average insulation standard (Figure 4). An additional return air flow will be necessary to perform sufficient transport capacity. Of course, highly contaminated air must not be used as return air. This air from kitchen and bathrooms will go directly to the heat recovery system.

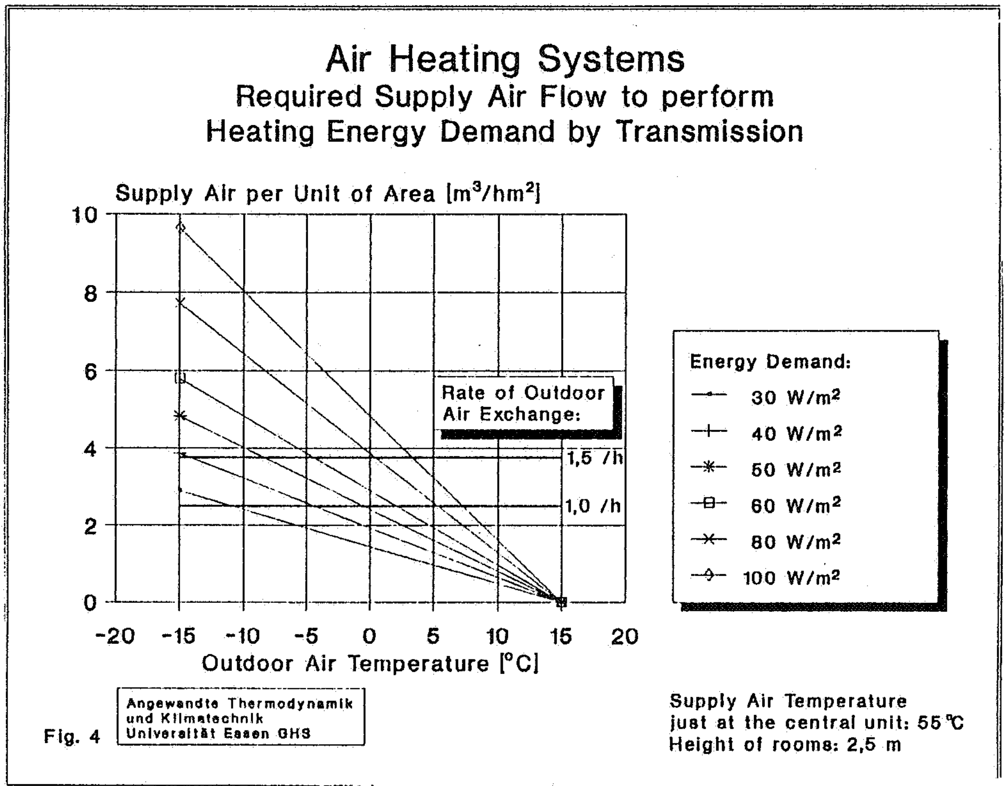

Fig. 4

Conventional central units of air heating systems with only one fan for supply air can increase the fan speed to perform higher heating capacities. Therefore special disc-type motors are available allowing a modification of the driving capacity by reducing the motor operating voltage. Usually the central units are equipped with a sequence switch for up to five operating points. At low outdoor air temperatures the return air flow damper will be opened and the operating voltage of the fan motor will be raised. Other central units are available performing continous adaption of fan capacity. Of course these systems cause higher first costs although there is no significant reduction of operating costs.

As a matter of principle the fan-motor has to be designed for the maximum desired air flow corresponding to the maximum operating voltage.

3. Improved Methods of Designing Central Units

A well designed fan shows an efficiency characteristic which follows exactly the system pressure-volume characteristic. That means that the mechanical efficiency is not influenced by the speed variation. The electrical efficiency of the motor however decreases strongly with reduced operating voltage. Characteristics of such fan-units measured by the producers show that the reduction of air flow to 50% causes the total efficiency dropping down to 20% of the full-load efficiency (Figure 5). The physical reason has to be seen in the low part-load efficiency of motor.

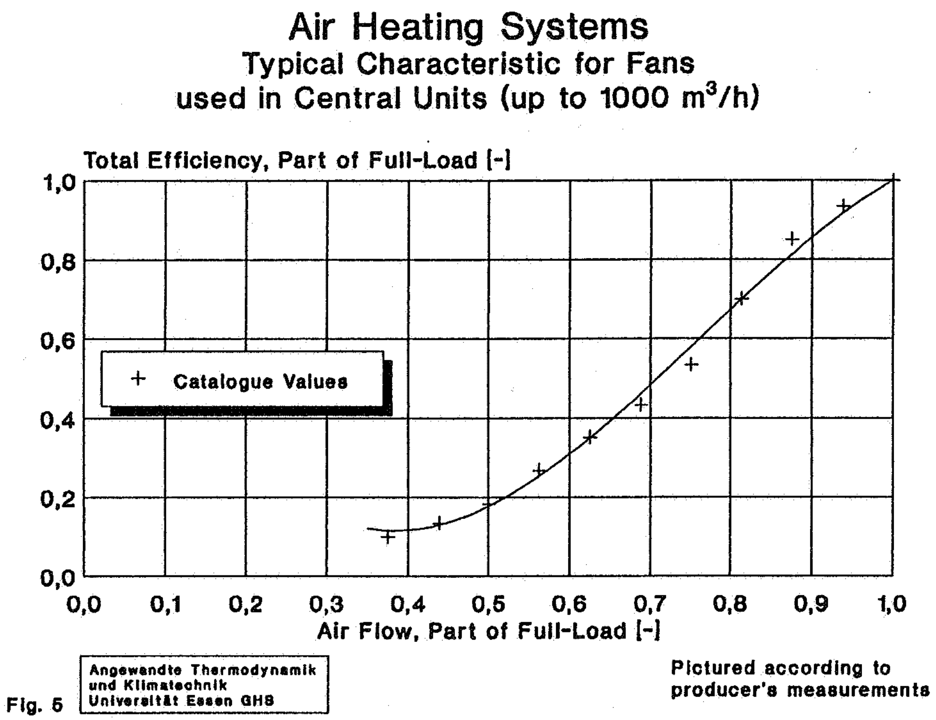

Fig. 5

It is a fact that fans in the central units of air heating systems will be operate under part load conditions most time of the year. Figure 9 shows the frequency distribution of outdoor air temperatures in the German town of Essen. Only 300 h/a outdoor air temperatures below -4°C have to be expected. Considering the low efficiency of air feed along this time the operation of those plants is more expensive than necessary. One method to improve the electrical efficiency under part-load conditions is to accomodate the speed by means of frequency transformation. The costs for such electrical controls are much higher than costs for a complete fan unit. So another way has to be found to guarantee lower driving energy demand for the air feed.

An effectful possibility would be a new construction of central units equipped with two fans. One of them is designed to transport ventilation air for outdoor air temperatures above -4°C. This fan will always operate at its maximum electrical efficiency.

The second fan is designed to transport both ventilation air and additional return air considering the higher pressure-drop in the supply air system. So the air flow capacity of this second fan has to be more than twice of the first stage fan. During operation of this fan the first one stops.

A modification of this concept can be made by operating the ventilation air fan all over the year even for low outdoor air temperatures, when the second fan is working additionally. In this design the return air fan can be designed smaller than above mentioned. The total air flow will be produced by both fans for low outdoor air temperatures.Indeed, the operating point of the ventilation air fan moves to a lower airflow as a consequence of the increasing pressure drop. This will cause a lower rate of outdoor air exchange and an insignificant decrease of the total efficiency for this fan. Both effects can be accepted without any problem. At low outdoor air temperatures the absolute humidity is very low so that the indoor air humidity will not rise to an inacceptable value with the lower air exchange rate. The reduced ventilation fan efficiency for low outdoor air temperatures can be neglected, because the operating time (300 h/a) is very short.

In comparison with the conventional concept the average efficiency of fans can be raised remarkable.

4. Conclusion

Depending on the above mentioned facts a comparison between air feed energy demand of a conventional plant (1-fan-system) and improved construction (2-fan-system) is made. The following charts basing on fan characteristics from a well-known German producer and meteorological dates from DIN 4710 proove that annual operating costs for the driving energy can be reduced to about 50% to 30%. This new concept could help to improve the economy of operation of air heating systems without raising first costs.

This seems to be an important fact to help air heating systems getting more distributed. Discomfort of persons caused by exceeding maximum pollutant concentration on indoor air as well as damages of building construction surely can be avoided by using ventilation systems. Combining heating and ventilation in domestic buildings should find greater acceptance than the urgency to spend more money for separate systems. Therefore further development of modern air heating systems will be a duty for producers and scientists.

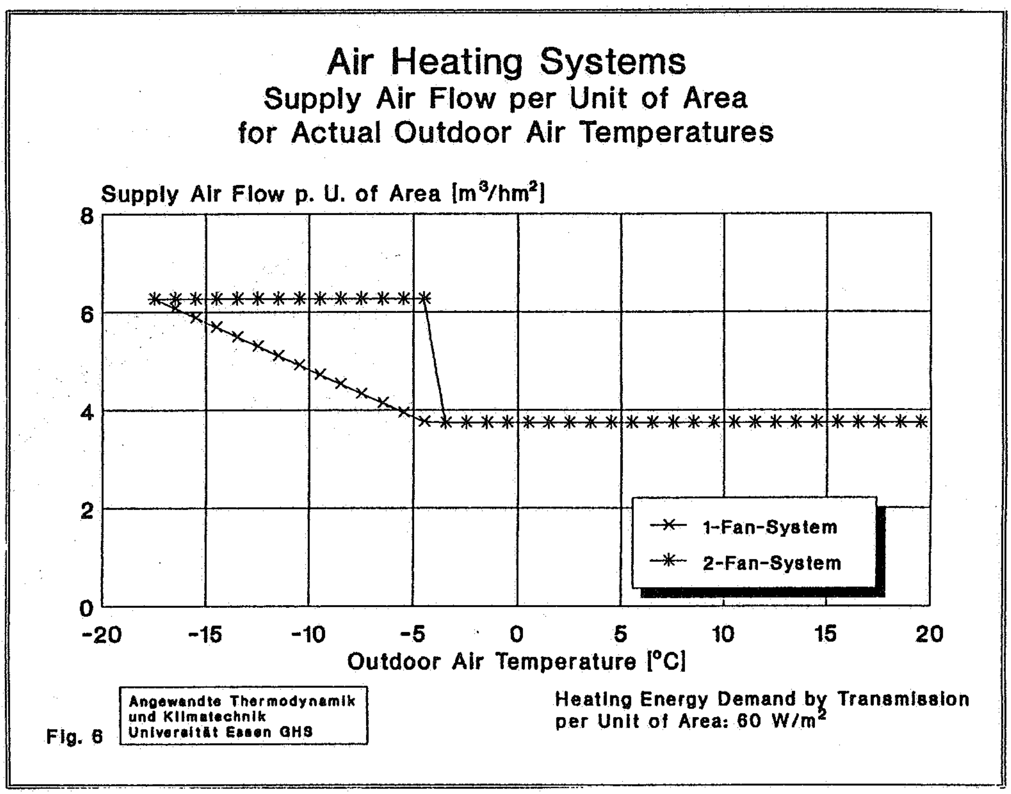

Fig. 6

Air Heating Systems
Fan Efficiency for Actual Outdoor Air Temperatures

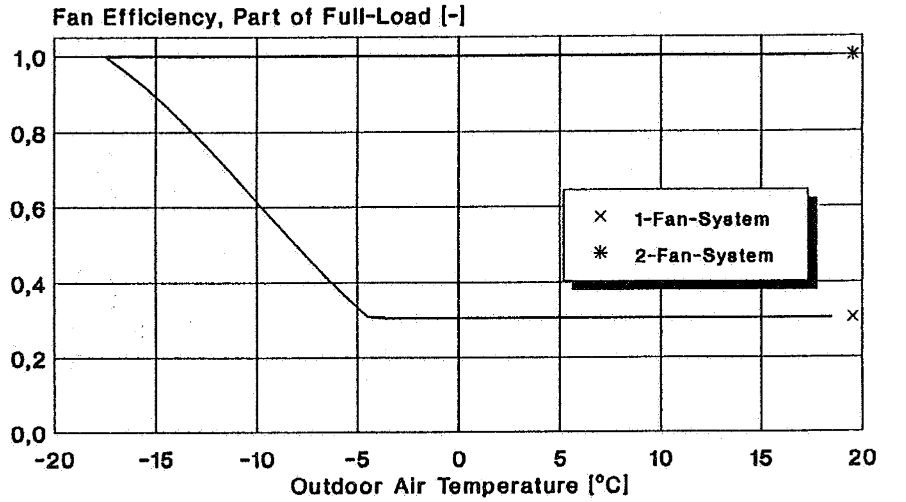

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Fig. 7

Heating Energy Demand by Transmission per Unit of Area: 60 W/m²

Air Heating Systems
Fan Power Input for Actual Outdoor Air Temperatures

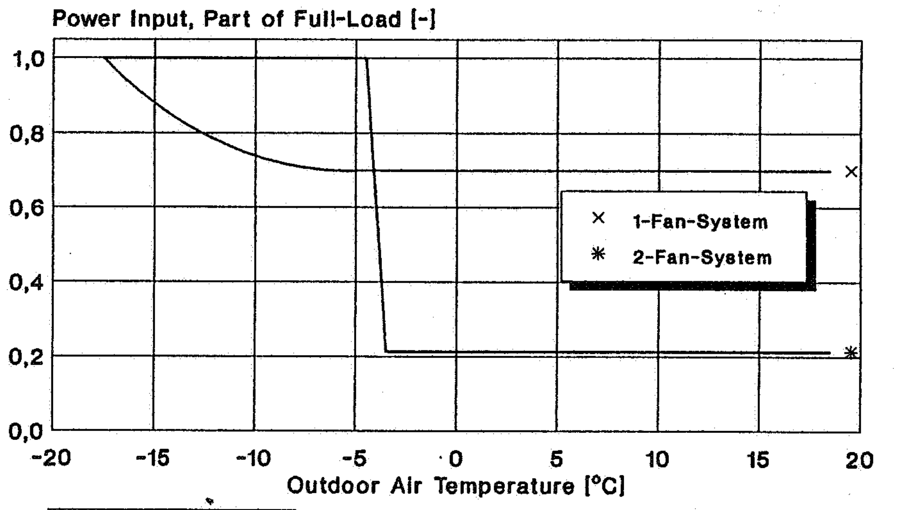

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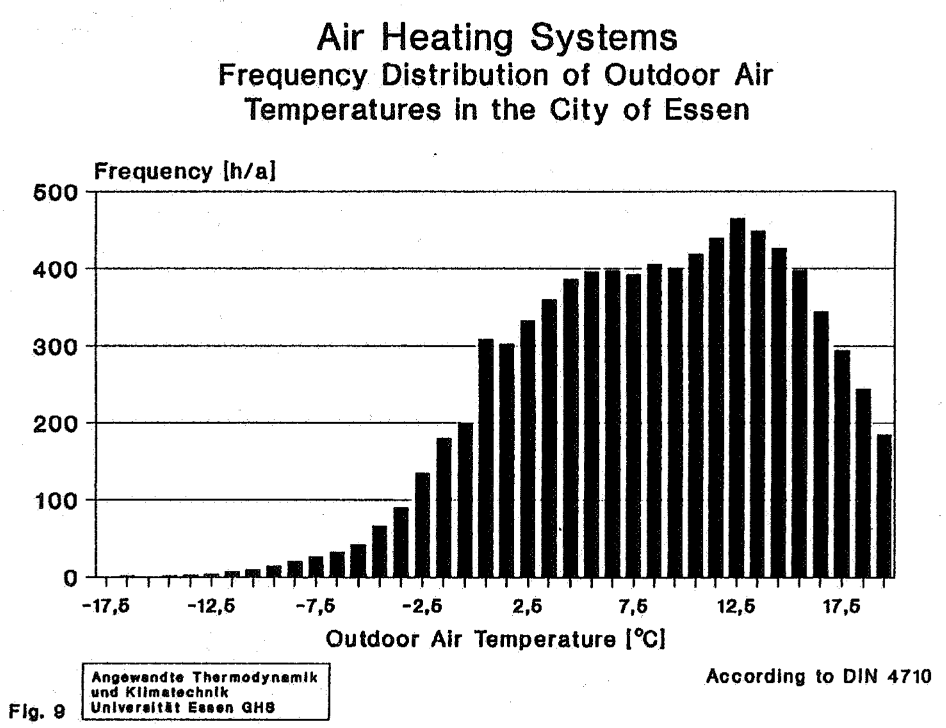
Fig. 8

Heating Energy Demand by Transmission per Unit of Area: 60 W/m²



Air Heating Systems
Frequency Distribution of Outdoor Air Temperatures in the City of Essen
Frequency [h/a]
500
400
300
200
100
0
-17,5
-12,5
-7,5
-2,5
2,5
7,5
12,5
17,5
Outdoor Air Temperature [°C]
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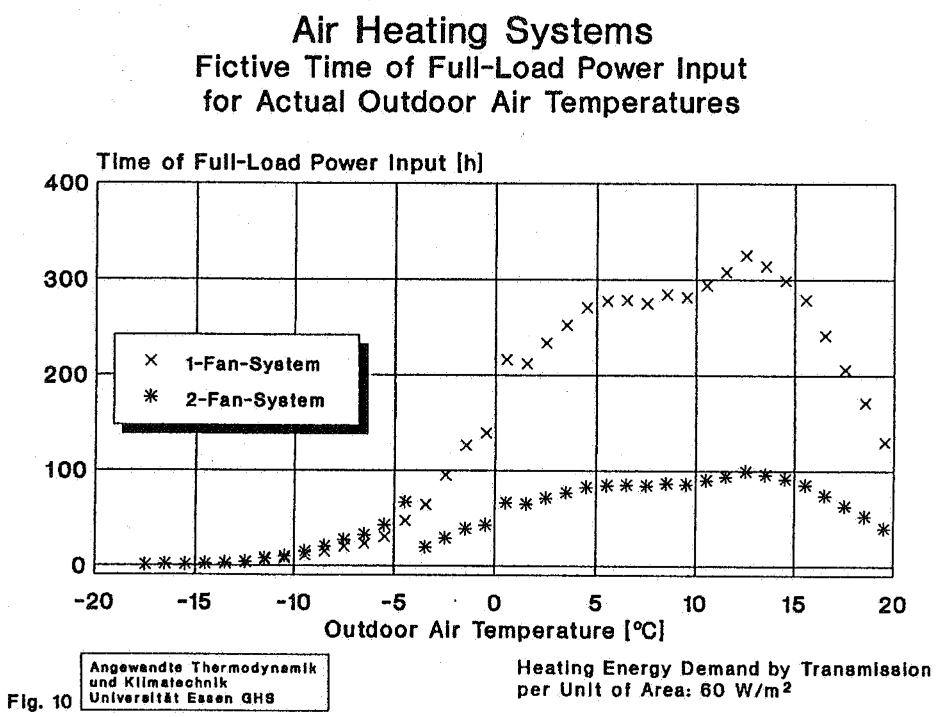
According to DIN 4710

Fig. 9



Air Heating Systems
Fictive Time of Full-Load Power Input for Actual Outdoor Air Temperatures
Time of Full-Load Power Input [h]
400
300
200
100
0
1-Fan-System
2-Fan-System
-20
-15
-10
-5
0
5
10
15
20
Outdoor Air Temperature [°C]
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Heating Energy Demand by Transmission per Unit of Area: 60 W/m²

Fig. 10

Air Heating Systems

Cummulative Curve for Fictive Time of Full-Load Power Input

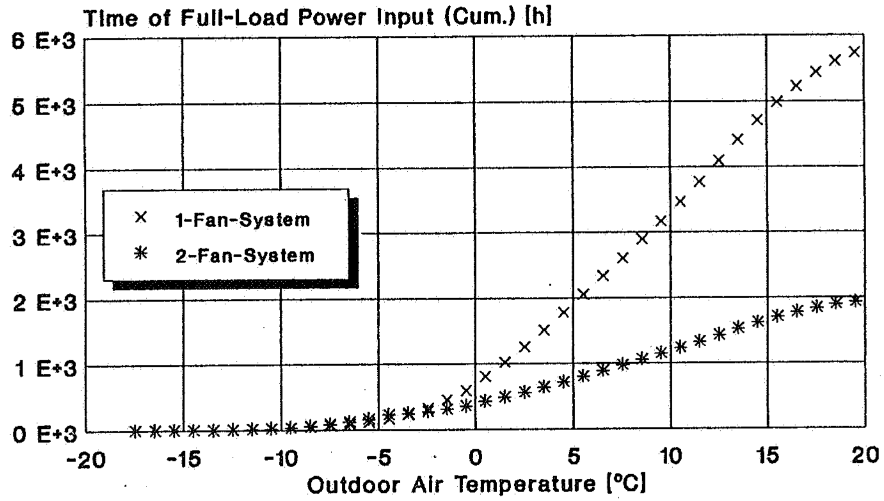

Fig. 11

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Heating Energy Demand by Transmission per Unit of Area: 60 W/m²

PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

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Poster 25

AIRTIGHTNESS OF SWEDISH RESIDENCES

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SYNOPSIS

Pressurization, or depressurization, of buildings is a tool to assess the airtightness of building envelopes. A common working pressure is 50 Pa, and the airtightness is expressed in terms of the number of air changes per hour at 50 Pa. To compare buildings of different size a more efficient measure is to define a non-dimensional leakage area.

We suggest a method to define and calculate the relative leakage area from pressurization data. The method corrects for calibration errors and the effects of aeromotive and buoyancy forces. It is demonstrated that the pressurization can be carried out at pressures much lower than 50 Pa, it is sufficient to apply pressures in the range from 10 to 20 Pa. The lekage areas predicted agree well with those predicted from tracer gas measurements of the air change rate.

The method has been used to calculate relative leakage areas of 300 Swedish dwellings. A comparison is made of the airtightness of residences of different age.

LIST OF SYMBOLS

A	area of building envelope [m^2]
n	rate of air exchange [h^{-1}]
n(50)	rate of air exchange at 50 Pa [h^{-1}]
p	pressure difference [Pa]
q	volumetric air flow rate [m^3/h]
v	air flow speed [m/s]
V	wind speed [m/s]
α	relative leakage area [cm^2/m^2]
$\alpha(4)$	relative leakage area at a pressure difference of 4 Pa [cm^2/m^2]
Δp	pressure correction [Pa]
ΔT	temperature difference [K]

AIRTIGHTNESS OF SWEDISH RESIDENCES

1. INTRODUCTION

Pressurization, or depressurization, of buildings is a tool to assess the airtightness of building envelopes, for example, to test building designs and, for particular buildings, to evaluate the impact of air infiltration and exfiltration on the thermal balance of the building. Combined with infrared thermography pressurization is also applied to detect leakage sites of building envelopes. When testing building designs, all ventilation slots and openings should be sealed, while this should not be the case when testing to predict infiltration rates.

The pressurization, or depressurization, is usually carried out at a pressure difference across the building envelope sufficiently high to ensure that the effect of aeromotive forces and buoyancy forces can be neglected. A common working pressure difference is 50 Pa and the airtightness is expressed in terms of the number of air changes per hour at 50 Pa. Airtightness norms for whole buildings have been specified in the Building Code of some countries, for example Sweden and Norway.

Building pressurization in most cases requires the use of portable fans mounted on an adjustable frame that can be fitted into a window- or doorframe. This assembly is called a blower-door.

For large buildings, an interesting concept is to use the fans of the ventilation system for pressurization. However, it is then not always possible to achieve a pressure difference of 50 Pa across the building envelope. This may be the case also for leaky houses even if a blower door is used.

Expressing the airtightness in terms of the number of air changes per hour at 50 Pa makes it difficult to compare the airtightness of buildings of different size. It is not self-evident how to normalize with respect to the area of the building envelope.

In this paper we investigate in more detail a method earlier proposed (see ref. 1). Applying this method one can:

1. Use low pressure data to assess the airtightness of building envelopes, and

2. Express the air leakiness in terms of a non-dimensional entity, the relative leakage area, which makes it possible to compare even differently sized buildings.

The data base used in the analysis consists of pressurization- depressurization data from about 300 houses and apartments collected by the indoor climate measurement unit of the Swedish Institute for Building Research.

2. DESCRIPTION OF THE METHOD FOR ANALYSIS

Using building pressurization data to plot the air flow across the building envelope versus the pressure difference, one in general obtains a plot where data points for pressurization and depressurization fall on two slightly convex curves displaced relative to one another (see Fig. 1)

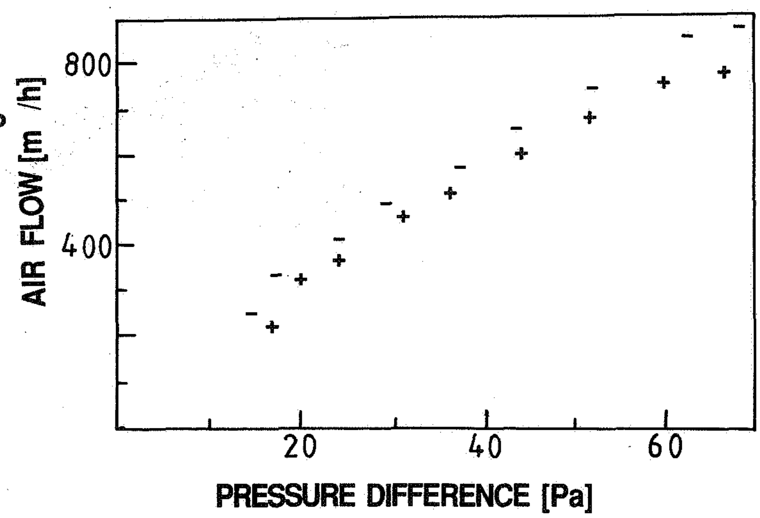


Fig. 1
An example of data points from a pressurization test, air flow versus pressure difference across the building envelope, pressurization (+) and depressurization (-). Note the convexity of the data ponts. Normally, the airtightness of the building envelope is expressed in terms of the number of air changes per hour at a pressure difference of 50 Pa, obtained from the data points by interpolation.

One can then interpolate the pressurization and depressurization curves to a pressure difference of 50 Pa, form the average of the corresponding air flows, and express this average in terms of air changes per hour of the building. Extrapolation of the curves in the plot to pressure differences occuring in real life, rarely above 10 Pa, is difficult because of:

1. The curvature of the data, and
2. The influence of aeromotive and buoyancy forces, and other factors.

It is possible to calculate theoretically a correction, Δp, to the pressure difference measured, p, that compensates for aeromotive and buoyancy forces. This is not of much practical use as one has to consider also factors such as windows and doors moving slightly inwards or outwards with changing pressures, the onset of threshold effects for flows in cracks and, perhaps the most important source of error, the placement of the outdoor pressure gauge which affects the calibration. All these factors are responsible for the displacement of the pressurization and depressurization curves, and their exact impact on data collected is not known. Therefore, instead of a theoretical determination of the pressure correction, Δp, one has to determine the magnitude of the correction from data.

What is required is an approach that can compensate for the above factors by bringing the pressurization and depressurization curves on top of one another and, at the same time, take away most of the curvature to facilitate extrapolations to low pressures. The air flow rate through openings should, for relevant flow regimes, grow as the pressure difference raised to some power, the power taking a value between one half and one. This has been confirmed by field tests to be true also for pressurization data from buildings, even if there is no reason a priori why this should be so due to the complexity of the air flows across building envelopes.

To take away most of the curvature, we use instead of the variables pressure difference, p, and air flow rate, q, a new set of variables, the flow speed, v, defined from

$$v=\sqrt{(2p/\varrho)},$$

ϱ being the air density, and the variable α, the relative leakage area, defined from

$$\alpha=q/(vA),$$

where A is the area of the building envelope.

The variable v has the dimension of velocity and is a measure of the average flow speed across the building envelope, while α is a dimensionless variable describing the effective cross-sectional area of cracks and holes per square meter of the building envelope.

Suppose we have originally two sets of data points, (p^+, q^+) and (p^-, q^-), where the upper indices + and - refer to pressurization and depressurization data, respectively (as in Fig. 1). Now apply the following procedure:

Construct data sets in the new pair of variables v and α, (v, α), by first replacing the pressures p^+ and p^- by $(p^+ + \Delta p)$ and $(p^- - \Delta p)$, respectively, where Δp is the pressure compensation whose value is to be determined. Defining the corresponding flow speeds:

$$v^+ = \sqrt{[2(p^+ + \Delta p)/\varrho]} \text{ and } v^- = \sqrt{[2(p^+ + \Delta p)/\varrho]}$$

the new data set is now given by the points:

$$(v,\alpha) = \begin{cases} (v^+, q^+/(A*v^+) & \text{for pressurization data} \\ (v^-, q^-/(A*v^-) & \text{for depressurization data.} \end{cases}$$

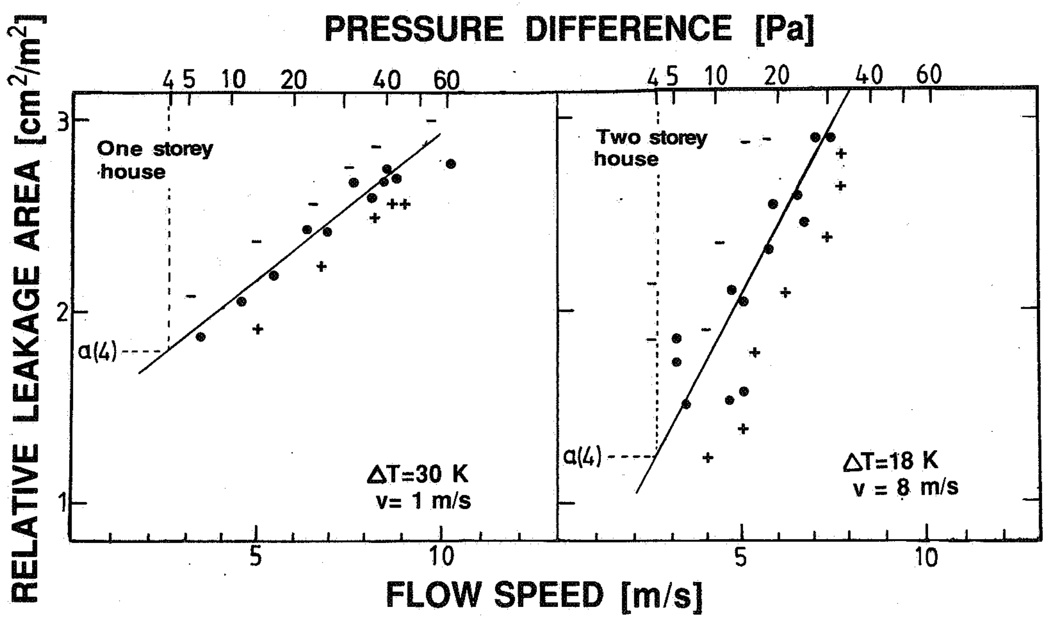


Fig. 2
Measured data points from pressurization (+) and depressurization (-) for two houses, one (left) where the pressure is dominated by aeromotive forces (the wind speed, v= 8 m/s), and another (right) where the pressure is dominated by stack effects (the indoor- outdoor temperature difference is 30 K). The data points have been plotted in the two variables, flow speed and relative leakage area, by the method described in this paper. The resulting data points are given as circles. The straight lines are those giving the best fit to the circles. The hatched lines indicate the extrapolation to obtain α(4), the leakage area at a standardized pressure difference of 4 Pa.

All those data pairs are now to be regarded as belonging to one common data set (see Fig. 2). The pressure correction Δp, which may be positive or negative, is now chosen so that it maximizes the value of the linear regression coefficient if a linear fit to <u>all</u> (pressurization <u>and</u> depressurization) data points in the (v,α) plot is carried out for different values of Δp. This step brings the pressurization and depressurization data together. In most cases, the pressure correction Δp takes a value of a few Pa, or less.

One can now choose a reference pressure (or flow speed), read off the corresponding value of the relative leakage area by interpolation or extrapolation of the straight line in the (v,α) plot, and use this value to characterize the air leakiness of the building envelope. One may use a value of 4 Pa (corresponding to a flow speed of 2.5 m/s) as reference pressure (see ref. 2), a pressure roughly corresponding to the average pressure across the building envelope for external temperatures and wind speeds normal to many climatic regions. We will denote the leakage area at 4 Pa by $\alpha(4)$ and the air exchange rate per hour at 50 Pa by $n(50)$.

When does the above procedure not work? Out of 300 tested cases this method worked in all but two cases. For an example see Fig. 3.

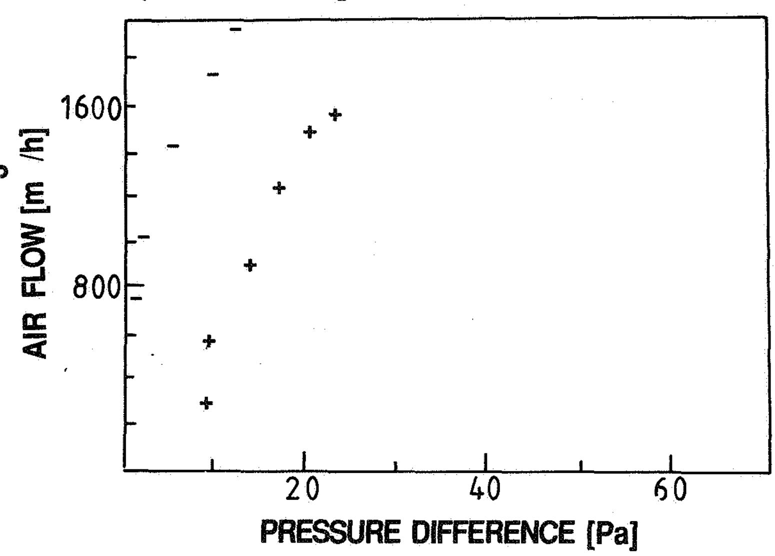
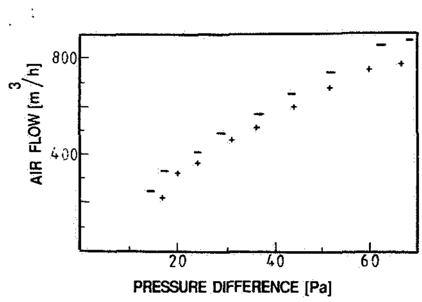


Fig. 3
Pressurization (+) and depressurization (-) data from one of the two cases out 300 for which the method described did not work. Data are from a very leaky house, using an ordinary blower door it was not possible to attain a pressure difference of more than about 20 Pa.

The method desribed above is shortly illustrated in Fig. 4.



1. Take data pairs (pressure p and air flow rate q) for pressurization (+) and depressurization (-).

2. Transform to the new variables

flow speed $v = \sqrt{(2p/\rho)}$
relative leakage area $\alpha = q/(vA)$

where ρ is the air density and A the building envelope area

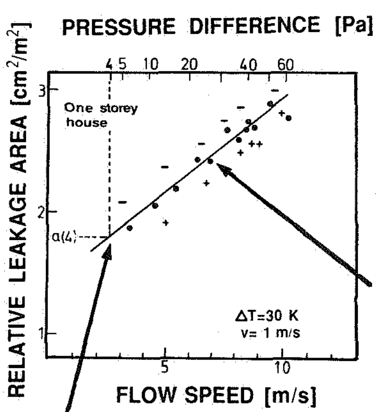
By defining the flow speeds

$v^+ = \sqrt{(2 (p^+ + \Delta p)/\rho)}$ and
$v^- = \sqrt{(2 (p^- - \Delta p)/\rho)}$

where Δp is a pressure correction, obtain new data pairs.

$$(v,\alpha) = \begin{cases} (v^+, q^+/(v^+A)) & \text{for pressurization} \\ (v^-, q^-/(v^-A)) & \text{for depressurization} \end{cases}$$

points



3. Choose the pressure correction Δp so that it maximizes the linear regression coefficient in a linear regression of all data points, pressurization as well as depressurization data •.

4. Extrapolate the one, from the regression resulting, best fitted straight line to a pressure of 4 Pa to obtain the relative leakage area $\alpha(4)$ at the standardized pressure 4 Pa.

Fig. 4
Overview of the method described in this paper for the derivation of the relative leakage area.

3. APPLICATION OF THE METHOD

To determine if the method described in the previous section can be used in practice, there are some questions that have to be answered (for a more thorough error analysis, see ref. 3):

1. What is the correlation between the relative leakage area at 4 Pa, $\alpha(4)$, and the rate of air change at 50 Pa, n(50)?
2. Does the method yield the same result if low pressure data, say pressures in the range 10 to 20 Pa, are used instead of pressures in the more normal range 20 to 70 Pa?

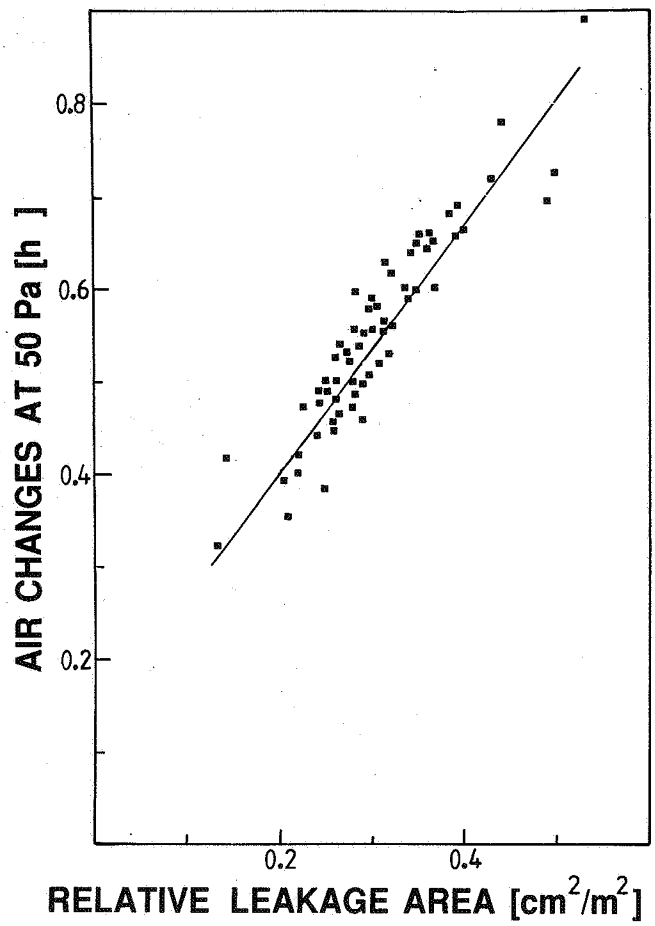


Fig. 5
The measure normally used for evaluating the airtightness of building envelopes, the number of air changes per hour at a pressure difference of 50 Pa, versus the relative leakage area at 4 Pa. The data are from a group of nominally identical townhouses. The square of the linear regression coefficient takes a value of about 0.9.

To answer the first question, we have studied a group of nominally identical two-storey townhouses. The air flow rate n(50) has been plotted versus the relative leakage area $\alpha(4)$ in Fig. 5 for all the houses in the group. The data are from measurements where all ventilation slots and openings had been sealed.

To provide an answer to the second question, we have used data on houses and apartments where there are available data points in the range from 10 to 70 Pa. We have compared the resulting value of the leakage area $\alpha(4)$ when all data points in the range 10 to 70 Pa have been used <u>to</u> the resulting value of $\alpha(4)$ when only data in the ranges 10 to 20 Pa, 10 to 30 Pa and 20 to 30 Pa, respectively, have been used.

The data are displayed in Fig. 6. The average number of data points available for the determination of $\alpha(4)$ in the low pressure range was four. Only data sets from houses containing at least three data points in the respective low pressure range have been used.

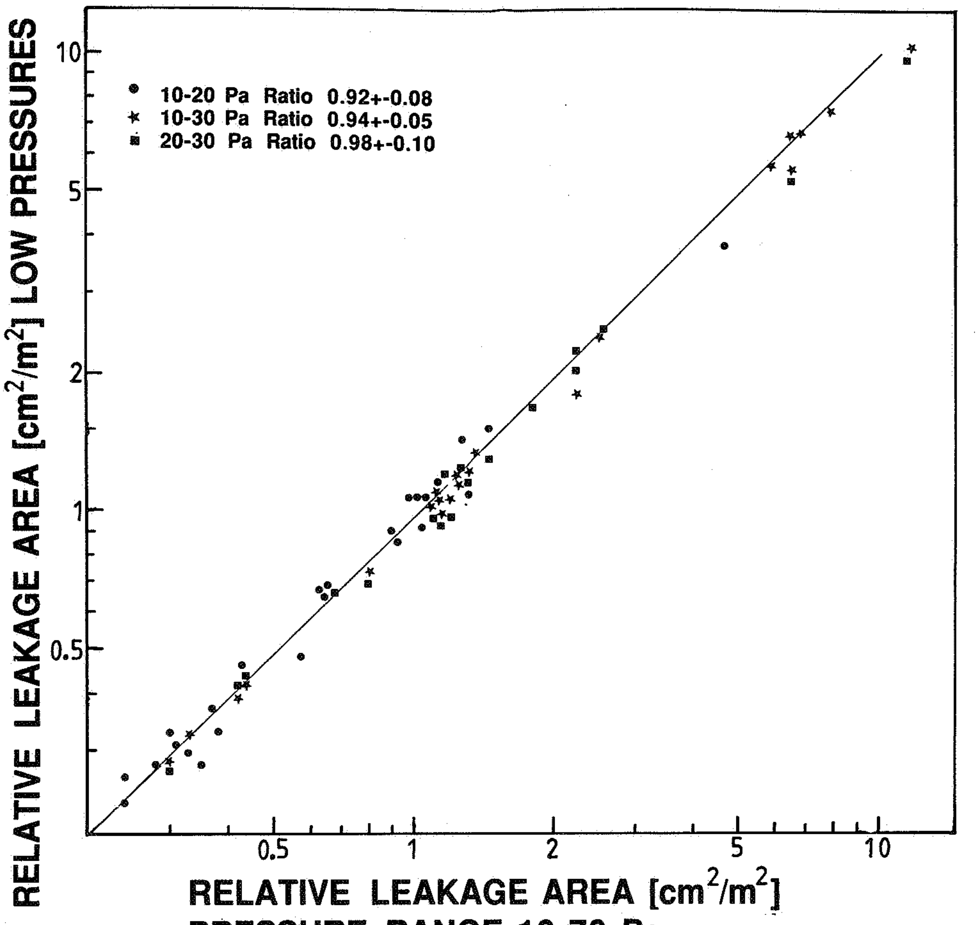


Fig. 6
The relative leakage area as calculated from data sets containing a pressure range from 10 to 70 Pa versus the corresponding leakage area when only the data points contained in the pressure ranges 10-20, 10-30 and 20-30 Pa, respectively, have been used. Also given is the ratio and the standard deviation of the ratio (leakage area from low pressures)/ (leakage area for the range 10 to 70 Pa). All ratios are compatible with being equal to one. Data are from houses, townhouses and apartments.

The resulting ratio of the value of $\alpha(4)$ using low pressure data to the value obtained using the full pressure range is, also, given in Fig. 6. There is an indication that low pressure data may yield somewhat lower values of $\alpha(4)$ than data in the pressure range 20 to 70 Pa, even if all ratios are compatible with being equal to 1.0. The data span more than one order of magnitude of the relative leakage area.

One can then conclude that it should be possible to use just low pressure data to determine a relative leakage area at a pressure difference of 4 Pa serving as an indicator of airtightness of buildings.

To assess if the method can be applied to detect differences in airtightness even for very airtight buildings, the relative leakage area has been calculated from pressurization data for a group of 44 identical townhouses. The data are displayed in Fig. 7. In this case, all inlets and outlets of the houses have been sealed.

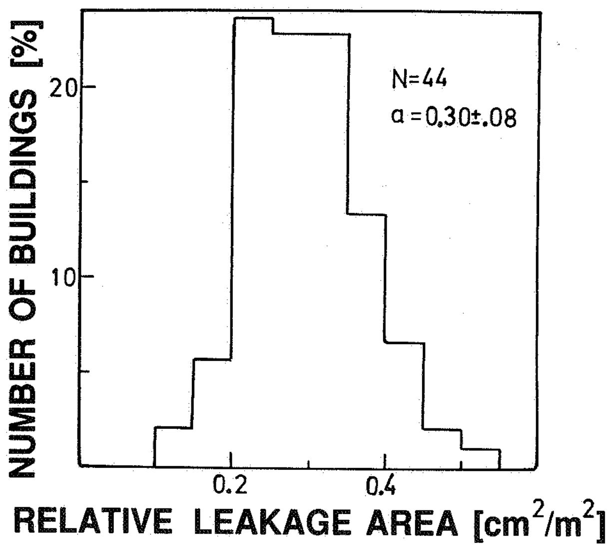


Fig. 7
Distribution of the airtightness in terms of the relative leakage area at 4 Pa for a group of 44, nominally identical, townhouses. The standard deviation is 25 %, indicating the difficulty to obtain a uniform quality of the airtightness of building envelopes even for identical houses.

The leakage area obtained from pressurization data and the method described above can be compared to the leakage area for the same buildings derived from tracer gas measurements of the air change rate (see ref. 1).

The result of this comparison is displayed in Fig. 8. In this case, all air inlets and outlets of the buildings had not been sealed. There is a fairly good agreement, the standard deviation of the ratio of (the leakage area predicted from pressurization) to (the leakage area predicted from the air change rate) is about 15 %.

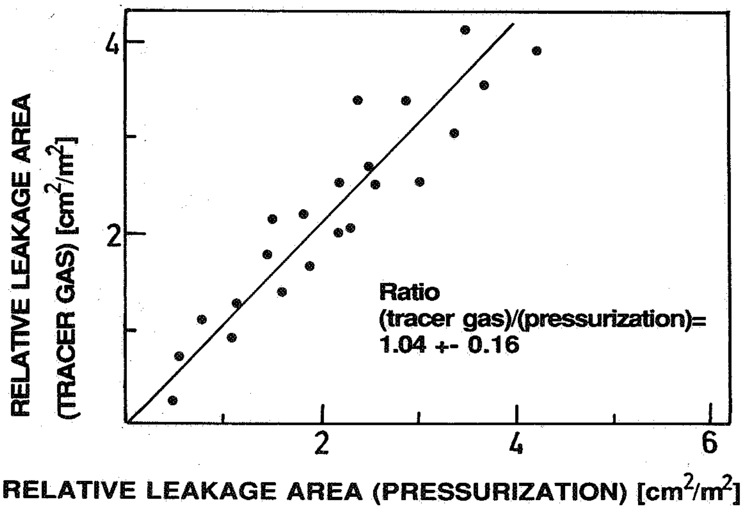


Fig. 8
The value of the relative leakage area as obtained from tracer gas measurements of the air change rate (see ref. 1) versus the relative leakage area for the same buildings as obtained using the method described in this paper. The data are for measurements when air inlets and outlets were not sealed. The ratio (leakage area from air change rate data)/ (leakage area from pressurization data is compatible with being equal to 1, the standard deviation is about 15 % of the average ratio of 1.04.

Using a data set of pressurization measurements from 300 Swedish residences, we have calculated the average relative leakage area (all ventilation openings sealed) for Swedish houses of different age. In Fig. 9 this is compared to measured values of the air change rate for houses of the same age (ventilation openings <u>not</u> sealed). For air change rates, the fall-off with the year of construction is less dramatic than it is for the relative leakage areas.

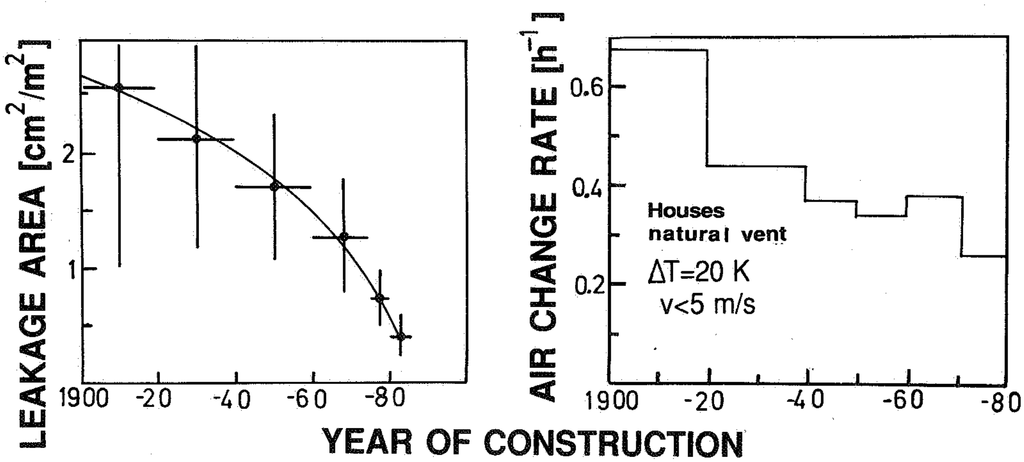


Fig. 9
The airtightness, expressed in the relative leakage area at 4 Pa, of Swedish houses with sealed ventilation slots and openings (left) and the average rate of air exchange for Swedish houses with natural ventilation (right) constructed this century. The air change rate has been corrected to an indoor- outdoor temperature difference of 20 K and measurements carried out at wind speeds exceeding 5 m/s have not been considered.

4. CONCLUSIONS

We have described a method for the calculation of the relative leakage area of buildings using data on air flow rate and pressure difference from pressurization tests. As an indicator of the airtightness of building envelopes, one can use the relative leakage area at a pressure difference of 4 Pa. The value of the leakage area is obtained from a plot where the original data on the variables air flow and pressure difference have been replaced by a new pair of variables.

There is a good correlation between the relative leakage area at 4 Pa and the rate of air exchange at a pressure difference of 50 Pa.

The method yields approximately the same value of the relative leakage area whether low pressure data from 10 to 20 Pa or pressure differences in the range from 10 to 70 Pa are used. The method has previously been shown to give a value of the relative leakage area that is close to the value deduced by measuring the rate of air exchange using tracer gas techniques (1).

To confirm that the method described can be put to practical use, the method should be verified by pressurization tests on more building types. One should also carry out several pressurization tests on the same building using low pressures to better estimate the error of the method.

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PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

10th AIVC Conference, Dipoli, Finland
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Poster 26

DEMAND CONTROLLED AIR DUCTWORK

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SYNOPSIS

A demand controlled air ductwork should be so dimensioned that the flow controllers have good flow and acoustical operation conditions. From the air flows in a room and its highest permissible sound level, the highest differential pressures allowable to the air flow controllers (duct air flow controllers and terminal devices) are selected. The required minimum differential pressure is 50-100 Pa or a higher differential pressure determined according to the outdoor temperature (-20°C), the height of the building and the air conditioning system required to control the thermal disturbing forces. From these differential pressures and the preadjustability of the air flows are determined the average pressure level of the ductwork and the highest permissible pressure losses of the air ducts, which at most may be about half the average pressure level. Also determined on the basis of the pressure level of the ductwork are the air tightness requirements in order to have control over the leakage noise. Suitable duct sizes have been ready-calculated. The pressure losses of the ducts are simply estimated from the pressure losses from local losses (bends and branches). Friction losses in the ducts are disregarded.

The dimensioning of the ducts should be adapted to the standards of the equipment technology (flow, acoustics, air tightness) that is available. The recommended total pressure loss (ducts and air flow controllers) of a demand controlled ductwork should be a maximum of around 200-300 Pa excluding the pressure loss of the central unit.

1. INTRODUCTION

The most important requirements in future techniques of controlling the indoor climates of buildings where human beings are concerned will be a steady, locally (by the room) controllable ideal temperature and quiet ventilation free of draughts, which will be controllable according to need.

In the air conditioning system of the future, air flows and thermal power will be controllable, in time and in place, in line with people's needs. To facilitate this, calculations and working experiments have been made from flow and acoustical simulations on air duct systems, and performances required of the equipment have been defined. This has been done in order to produce new principles of dimensioning air ducts, and simple methods of doing so.

2. PROBLEMS WITH PRESENT PRACTICE

2.1 Methods of dimensioning air ducts

Air ducts are usually dimensioned on the basis of pressure loss calculation for the air flow routes, or on the basis of old "empirical" duct velocities. Such traditional methods of dimensioning result in excessive velocities and pressure losses in air ducts, or in air ducts that are difficult to adjust and balance and that are acoustically uncontrolled, or in architectonically ugly air ducts often varying in diameter and subject to leakage. The dimensioning of the ducts should be adapted to the standards of the equipment technology that is available.

2.2 Adjusting, balancing and noise control

The higher the pressure loss of the ducts, the higher will be the differential pressures under which the air flow controllers (duct air flow controllers and terminal devices) have to function to balance the air flows, i.e. the pressures. The higher the differential pressures under which the air flow controllers have to function, the more difficult (or even impossible) and expensive will be the control of flows, and of noise levels in particular.

The problems and acoustical nuisance in adjusting and balancing present air ducts are chiefly due to the high pressure losses caused by excessive duct velocities. Consequences include the great throttling requirements on the dampers in branch ducts that cannot always be met even with dampers /5/, which produces ducts that are uncontrollable in terms of flow and acoustics. Air ducts intended for variable air flows cannot function as the air flow controllers, owing to high pressure losses at the beginning and/or end of the ductwork, may find it beyond their range of operation. This implies an obvious design fault, as the dimensioning of such ducts has not taken the technical level of the equipment into account. The high differential pressures cause the pressure in the ducts to become unreasonably high. The result will be noise arising from increased leakage and also great aesthetic nuisance at the places where the leaks occur, greater cleaning needs and an increase in the fan energy requirements. Because of the leakages, the pressure variations in the building may become uncontrollable, which is likely to cause disturbances in the form of draughts and condensation, and the spreading of bad odours and impurities.

The air flows, the pressures and the sound levels in the air duct system cannot be controlled unless the system is sufficiently airtight.

3. NEW PRINCIPLES OF DESIGNING AND DIMENSIONING

The ventilating, heating and cooling demands of buildings and the various premises they contain, and the quality of the air are the starting points for dimensioning the air flows and thereby the air ductwork. In actual practice the functions of the various premises in a building are often subject to alteration, perhaps even at the time when it is being designed and built. Once the building has been put into use, the purposes, loads and required quality levels of its various spaces may be subject to almost constant change.

The traditional determination of economical duct sizes and pressure losses has usually been based only on a comparison of capital costs for the ductwork and operating costs of the fans. But overall economy should be based on a far more extensive examination of the entire building, and in particular on the use to which it is being put.

The examination should cover the effects of the dimensioning of the ducts on the quality of the indoor climate produced by the air conditioning system and thus on human satisfaction; its effects on working efficiency, on the performance figures for energy and especially control technology, on ease of flow and acoustic adjusting, and on alterations to ducts and structures prompted by changes in use, and also on the costs of re-adjusting.

The method of dimensioning and designing the ductwork should be one in which errors in design or implementation do not essentially impair the flow or the acoustics of the system. The flow balance in the duct system should not be sensitive to changes in the flow of individual components. Good operating conditions should be created for flow controllers through the dimensioning of the ductwork.

4. TARGET FEATURES OF AIR DUCTWORK

Table 1 shows the target features for an air ductwork, controlled according to demands, which should be the basis of the dimensioning.

A dimensioning of the ducts that will implement these target features would even nowadays provide the opportunity for an economical implementation of air conditioning with variable air flows, which means one that can be controlled in terms of time and place. This is actually what the new Finnish regulations and guidelines /7/ aim at.

Table 1. Target features of demand controlled air ductwork.

* Significant performance in energy and control technology
* Functioning with variable air flows, i.e. allowing load variations
* Stable air flows
* Acoustically controlled
* Airtight
* Simple to design
* Self-adjusting or easy to preadjust
* Easy to use (operating status easily to check)
* Architectonically attractive

5. CONTROL OF DUCT PRESSURES AND NOISE LEVELS

The air flows in the duct system balance at a level at which the total pressure loss of each air route from the fan to the room is equal. For the air flows to remain up to design in all flow conditions, ductwork should be designed to remain as symmetrical as possible in terms of pressure loss under variable air flows.

The air flow controllers should have sufficient authority for adjustment, meaning the differential pressure relative to the total pressure loss of the ductwork.

The air flows of the system cannot be controlled unless the pressures in the system are controlled. In the designing of the system there should be a shift from an examination of air flows alone, mainly to the controlling of air flows and of noise levels through duct pressures. By controlling air flows through duct pressures, also noise levels can be easily controlled with existing equipment.

6. DIMENSIONING OF AIR DUCTWORK

6.1 Principle of overall dimensioning

Pressure loss and the noise generation of various duct sizes and duct components have been examined by means of flow and acoustical calculation models /3/ of the duct system. If the transmission of sound from the duct into the room through the terminal devices and especially through the duct wall is also taken into account, the highest permissible duct velocities will be found, as will the greatest permissible sound power levels of the duct on the basis of the highest permissible noise level for the room. If account is taken, moreover, of the pressure losses necessary to accomplish adequate stability and control of air flows and to control the noise by leakage of the ductwork, it will be possible generally to select recommended duct sizes.

6.2 Noise attenuation requirement of air ductwork

The highest permissible sound power level of the fan and the air flow controllers in the ductwork must be held down to the highest permissible sound level for the room according to the recommended values in Table 2.

Table 2. Maximum recommended sound power level in duct, dB.

Flow controller	Sound level in room, dB(A), 10 m^2-sab	Octave band centre frequency, Hz 63	125	250	500	1000	2000	4000	8000
Attenuated on side of terminal device	25	60	51	45	40	38	38	41	44
	30	65	56	50	45	43	43	46	49
	35	70	61	55	50	48	48	51	54
Unattenuated	25	57	47	37	29	25	22	21	21
	30	62	52	42	34	30	27	26	26
	35	67	57	47	39	35	32	31	31

The difference between the sound power level of the fan and the air flow controller (positions of adjustment agree to design pressure level and air flow) and the maximum recommended sound power level of the duct, by octave band, determines the sound attenuation requirement. In addition to the primary silencer of the fan, secondary silencers for the air flow controllers and to avoid "cross-talk" between rooms must be opted for when necessary. Further sound attenuation will not be necessary.

6.3 Pressure level in air ductwork

From the air flows in the room and the highest permissible sound level, the highest allowable differential pressures to the air flow controllers can be selected. The lowest differential pressure required is 50-100 Pa or the higher differential pressure required to control thermal disturbing forces /2/ which is determined according to the outdoor temperature

(-20°C), the height of the building and the air conditioning system. From the preadjustment of these differential pressures and air flows, the average pressure level for the ductwork and the highest permissible pressure losses for the air ducts can be determined, which should not exceed about half the average pressure level, Table 3.

Table 3. Aproximate depencence on pressure level of inaccuracy of preadjusted air flows.

Inaccuracy of pre-adjusted air flows, %	PRESSURE LEVEL of air flow controllers
10	5 • (pressure loss of ducts)
20-30	2 • (pressure loss of ducts)

The recommended pressure level of a variable air flow rate controller (maintaining a constant air flow rate when the pressure differential between high and low pressure sides varies) is the average of the highest and the lowest pressure differentials that are acoustically permissible. The pressure level of an ordinary damper (air flow rate controller, which change the air flow rate by modifying the resistance), is the same as its differential pressure with a design air flow and position of adjustment.

An upper limit for the pressure level of an air flow controller is usually imposed by its own noise generation or by the leakage noise of the ductwork and highest permissible sound level of the room, Figure 1. On the basis of the pressure level and the selected positions of adjustment, the air flows will balance by themselves with ordinary dampers, as in Figure 1. When variable air flow rate controllers are used, the inaccuracies of air flows are also often of the order of magnitude of Table 3.

From the pressure level of the ductwork, the air tightness requirements of the air ductwork are also worked out, to allow for control of leakage noise. Figure 2 /6/ shows an estimate based on field measurements /4/ and calculations /3/ of the sound power level generated by a flow of leaking air according to various air tightness classes. The air ducts should meet the requirements of air tightness class C /7/, Figure 3, if the flows of leaking air are not to cause an unreasonable noise nuisance.

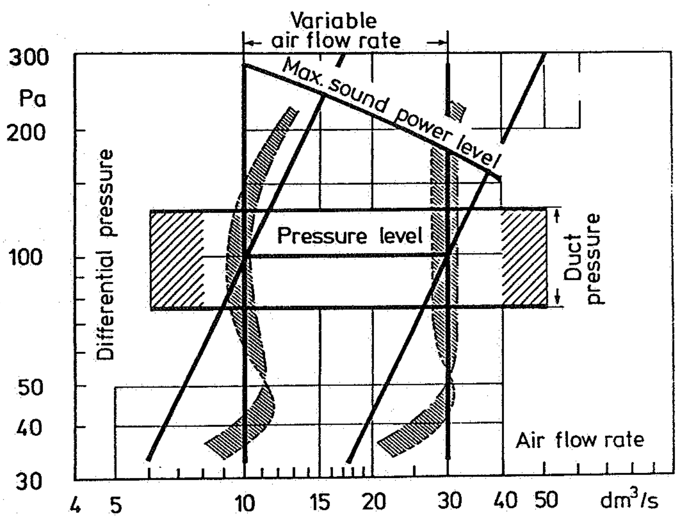

Figure 1. Air flows are determined on the basis of the positions of adjustment of various types of air flow controllers and on that of the selected pressure level. The sound level of the room is controlled when working below the maximum recommended sound power level.

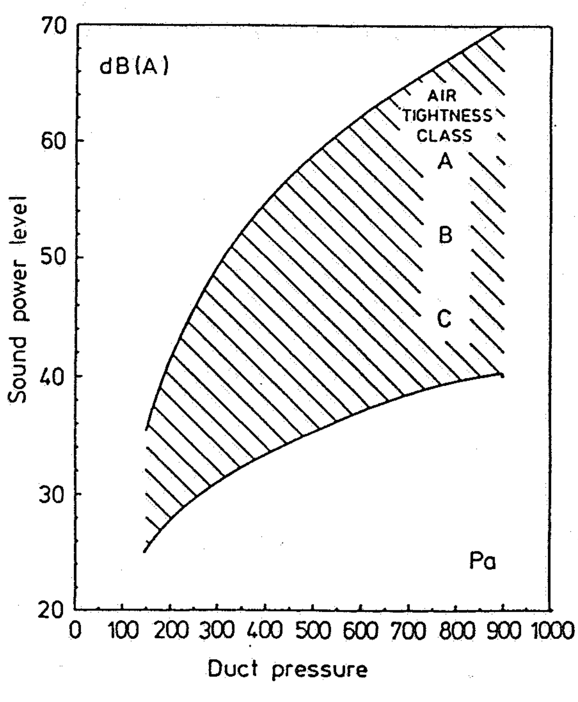

Figure 2. The estimated sound power level caused by air leakage in air ductwork system in various air tightness classes.

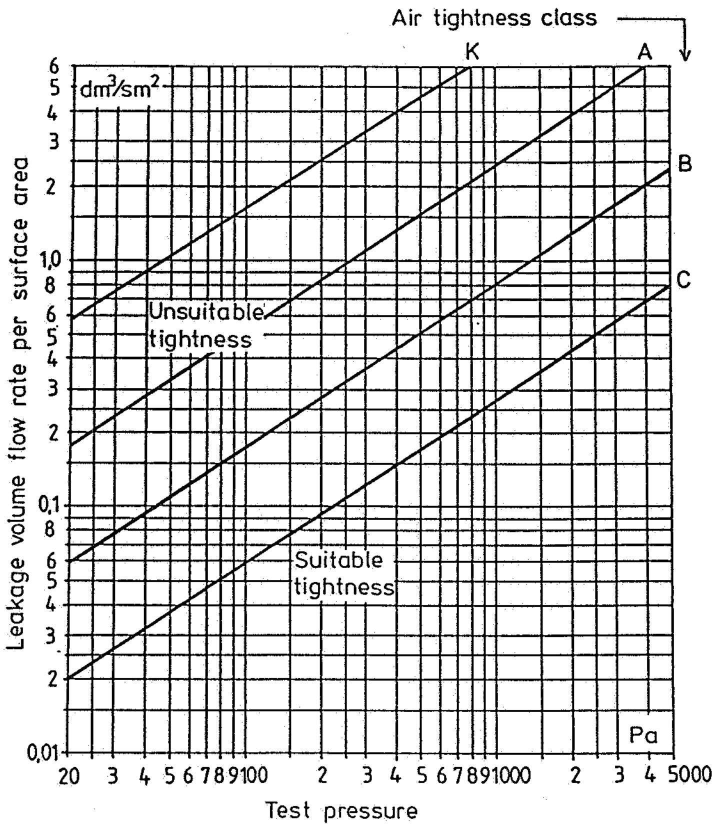

Figure 3. Air tightness classes for air ducts and central units according to Finnish regulations and guidelines.

Air tightness class A:

Visible ducts in a ventilated space.
Pressure difference relative to indoor air < 150 Pa.

Air tightness class B:

Ducts outside the ventilated space.
Ducts separated from ventilated space by means of covering panels.
Ducts in the ventilated space, pressure diffefence relative to indoor air > 150 Pa.

Air tightness class C:

Shall be applied as considered case by case:
- Ducts in high pressure systems.
- When duct leaks may produce pronounced adverse effects on the ventilation system operation, building air pressure conditions, indoor air quality or sound level.

Air tightness class K:

Enclosed air conditioners.
Equipment rooms and chambers for fans and other assemblies.

6.4 Choice of duct dimensions

The duct dimensions are selected according to highest air flow in the duct, from Figure 4. In branch ducts the maximum air flow is the same as the maximum air flow designed for the terminal devices. In the main ducts the maximum air flow need not always be equal to the sum of the maximum air flows of the individual rooms, but may be less. The dimension of the main duct is selected, where possible, on the basis of the sum air flow of the maximum air flows of the individual rooms or smaller than that by at most one duct size.

The demand controlled ductwork will work the better, the lower the duct velocities and the better the tightness of the ducts. In the limiting case of the maximum recommended duct velocity, it is always worth choosing the next larger duct size according to Finnish standard SFS 3282 /1/. Table 4 shows the favourable effects of such a choice as an approximate rule of thumb.

Table 4. Favourable effects of the choice of the next bigger duct size (Finnish standard SFS 3282 /1/).

Quantity	Effects of duct size	Effects of next bigger duct size
Duct velocity	v	$\approx \frac{v}{2}$
Pressure loss	Δp	$\approx \frac{\Delta p}{3}$
Sound power level	L_w	$\approx L_w$ - 7 dB
Leakage sound power level	L_{wl}	$\approx L_{wl}$ - 8...-20 dB

6.5 Estimation of pressure losses of ducts

Select duct velocities from Figure 4 on the basis of the highest permissible sound level for the room and the adjustability of the ductwork. A quick estimate of the pressure losses of the ducts can be made by counting the numbers of bends and branches. The average single local loss coefficient of the branches and bends is estimated as ζ = 1. On the basis of the duct velocity,

Figure 4 indicates the magnitude of the single local loss. The sum of the single local losses thus calculated is regarded as the total pressure loss of the ducts. The friction pressure losses of the ducts are disregarded in dimensioning ducts.

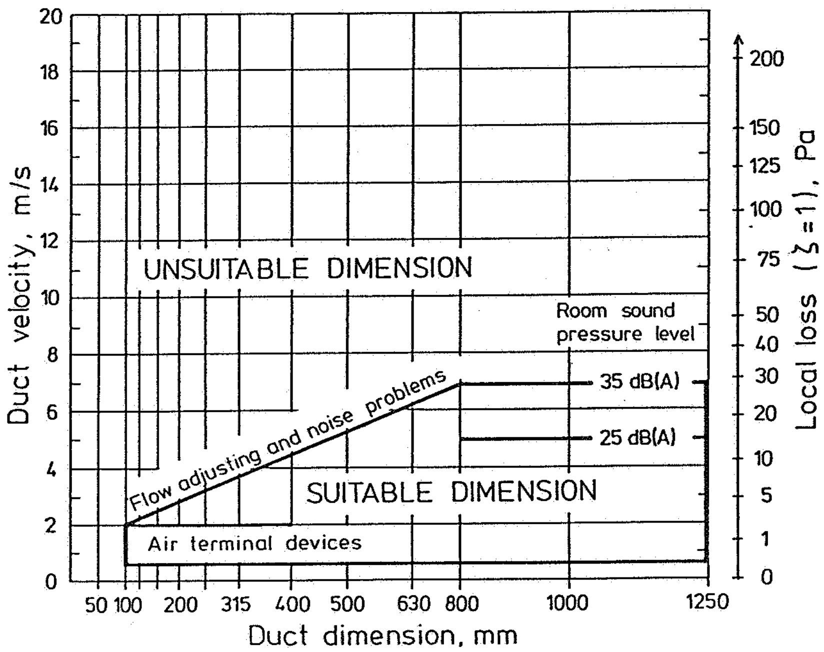

Figure 4. Recommendation for dimensioning of air ducts.

6.6 Adjusting and balancing of air ductwork

The traditional adjusting and balancing of an air ductwork which has been dimensioned overall is not required. A simple preadjusting and balancing can be made as indicated in Figure 5.

Checking of the preadjusting and balancing of a ductwork performed as simply as this will ensure that each room will get the air flows designed for it at the desired sound levels. If this does not occur, the ductwork is leaking too much or the pressures in the ductwork are not under control due to some other fault of implementation; or performance data supplied by the manufacturer for his equipment (air flow controllers and terminal devices, silencers, fans, etc.) is not valid, i.e. the quality level of the equipment does not accord with the stated values.

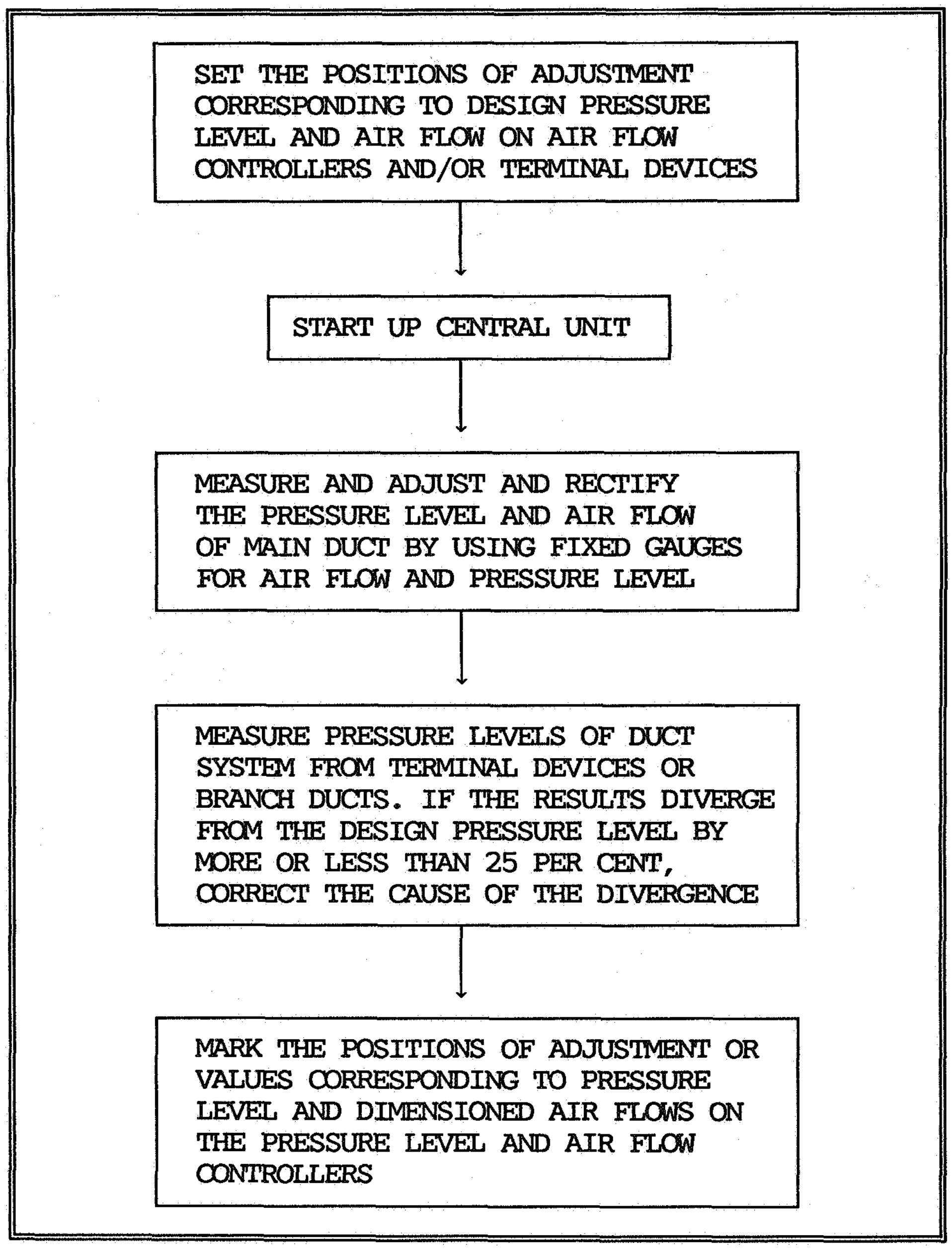

Figure 5. Checking of the result of preadjusting and balancing of air ductwork dimensioned in an overall manner.

7. CONCLUSIONS

A demand controlled ductwork should remain as symmetric as possible in terms of pressure losses in all variable flow conditions. The flow controllers should therefore have sufficient authority available for control, i.e. differential pressure relative to total pressure loss of the ductwork. Good flow and acoustical operating conditions should be created for the duct air flow controllers and terminal devices by means of the dimensioning of the ductwork. The noise of the fans must be attenuated by means of primary silencers, to comply with the highest permissible sound power level of the duct.

The standard of available technology determines the greatest permissible pressure losses in air ducts. Without silencers the present good duct air flow controllers and terminal devices can operate with pressure losses of at most 100-150 Pa, and the pressure losses of the ducts could thus be at most 50-75 Pa. The pressure losses of the flow controllers, attenuated with secondary silencers on the side of terminal device, could be at most 150-250 Pa, whereby the pressure losses of the ducts could then be at most 75-125 Pa.

When use is made solely of effective primary attenuation of the fan, the total pressure loss of the air ductwork should be a maximum of around 200 Pa. If use is made of flow controllers attenuated with secondary silencers on the side of the terminal devices, the total pressure loss of the air ductwork should be at most some 300 Pa. The air ductwork should meet the requirements of air tightness class C of the National Building Code of Finland /7/ if leakages are not to cause noise nuisance.

Supply and exhaust air fans of demand controlled ductwork should also be under control, for the flow controllers of the branch ducts cannot be used to change air flows in the central unit without causing a noise nuisance. The supply and exhaust air flows should be in control of each other so that the pressure relations of the building remain as desired.

Acknowledgements

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PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

10th AIVC conference, Dipoli, Finland
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Poster 27

AIR CHANGES AND SCATTER IN MECHANICAL VENTILATION RATES IN SWEDISH RESIDENCES

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SYNOPSIS

In Sweden, the energy crisis in the early seventies resulted in attempts to lower the air change rate in buildings to reduce energy consumption. For many building categories, this lead to a deterioration of the indoor climate or problems with moisture and mould growth. Today, many residents demand higher ventilation rates, often the motivation is concerns about health and comfort. In this paper is presented results of measured air change rates in Swedish dwellings during the period 1974-88.

For some residential building categories, the average air change rate is much smaller than that prescribed by the Swedish Building Code, 0.5 air changes per hour (ACH). About half the apartments and houses with natural ventilation constructed after 1960 have a rate of air exchange that is 0.3 ACH, or lower. This is also the case for about fifteen per cent of the houses with exhaust air ventilation.

For buildings with mechanical ventilation, it has been investigated what are the differences in ventilation rates for apartments in the same building and for groups of identical houses after the final installation and control of the ventilation system. In both cases the scatter around the average is about 20 %.

AIR CHANGES AND SCATTER IN MECHANICAL VENTILATION RATES IN SWEDISH RESIDENCES

1. INTRODUCTION

Measurements of air change rates in dwellings have been carried out by the Swedish Institute for Building research since 1974. The rate of air exchange has been calculated from the measured decay rate of tracer gas concentration. The data bank now contains data on the air change rate in about 1000 dwellings.

Data do not describe the air change rate in dwellings at this instant, but is rather to be regarded as representing an average for the period 1974 -88. Also, data do not represent a true statistical sample, but have been collected in research projects having aims other than the measurement of air change rates. However, most residential categories and buildings of different age are represented in the material. A more detailed description of the data base has been given in ref. (1).

2. RESULTS

Air change rates are presented in Fig. 1. Many dwellings have a rate of air exchange smaller than that prescribed by the Swedish Building Code, 0.5 air changes per hour (ACH). About half of the apartments and houses with natural ventilation constructed 1960 or later have an air change rate of 0.3 ACH , or less. This is also the case for about fifteen per cent of the houses with exhaust air ventilation.

Regarding the houses with natural ventilation, the measured low ventilation rates can probably be explained by the gradually improved airtightness of building envelopes during this century (see ref. 2).

For dwellings with mechanical ventilation, the explanation has to be sought in terms of the quality regarding installation and control of ventilation systems. We have, therefore, investigated what is the dispersion of the ventilation rate for groups of identical townhouses and for apartments in the same building. The results are given in Fig. 2.

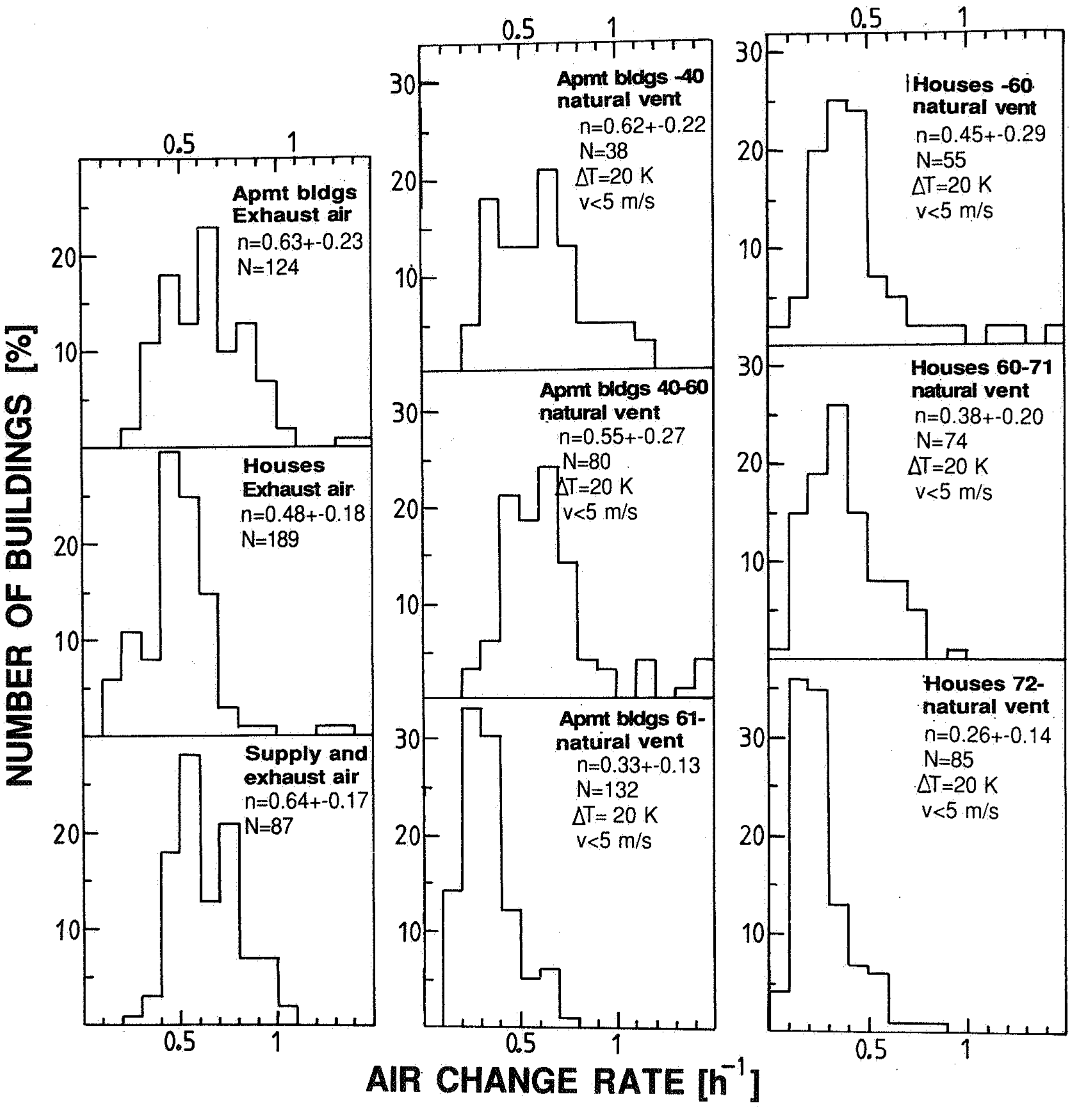

Fig. 1
Air change rate in Swedish dwellings, dwellings with mechanical ventilation (left), houses with natural ventilation (middle), and apartments with natural ventilation (right). Dwellings with exhaust air ventilation have been divided into houses and apartment buildings. Dwellings with natural ventilation have been subdivived according to the year of construction. Also given are the number of dwellings in each group, N, the average air change rate, n, expressed in air changes per hour (ACH), and the standard deviation of the air change rate. The air change rate has, for natural ventilation, been corrected to an indoor- outdoor temperature difference of 20 K and measurements for wind speeds exceeding 5 m/s have been excluded from the data set.

Many dwellings have an air change rate below the requirement of the Swedish Building Code, 0.5 ACH. One half of the apartments and houses with natural ventilation constructed since 1960 have an air change rate of 0.3 ACH, or less. This is also the case for 15 % of the houses with exhaust air ventilation.

For apartments as well as for houses, the standard deviation of the average ventilation rate is about 20%. This means that after an installation and control aiming at a ventilation rate of 0.5 ACH, about 70% of the dwellings have a ventilation rate between 0.4 and 0.6 ACH, while the remainder may have ventilation rates down to 0.3 or up to 1 ACH.

3. CONCLUSIONS

It is found that about half of the apartments and houses with natural ventilation constructed 1960, or later, have an air change rate of 0.3 ACH, or less. This is also the case for about fifteen per cent of the houses with exhaust air ventilation.

For apartments as well as for houses, the standard deviation of the average ventilation rate is about 20% after installation and control of a mechanical ventilation system.

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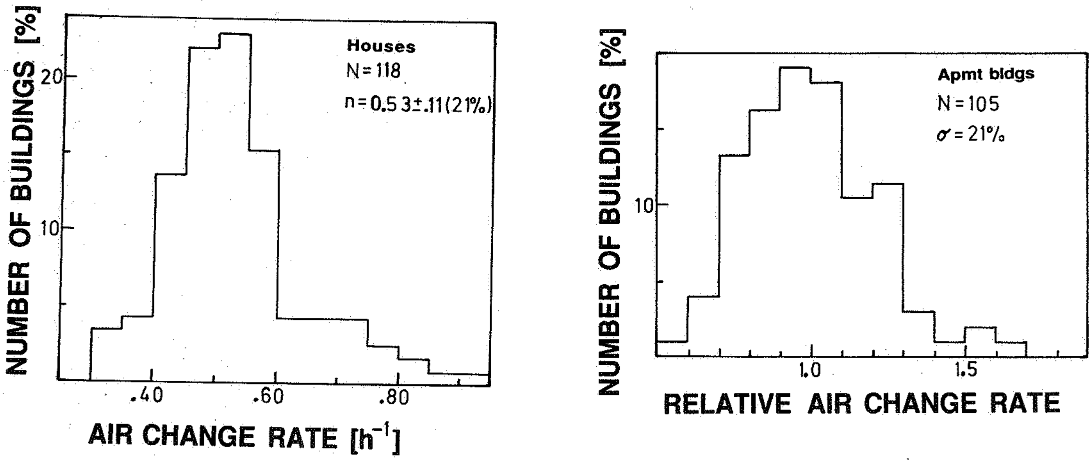

Fig. 2
Distribution of ventilation rates in townhouses (left) and apartments (right) with mechanical ventilation. The data for townhouses are from three groups of houses, the houses in each group are identical. The total number of townhouses, N, is 118. All three groups have an average ventilation rate between 0.52 and 0.54 air changes per hour. The data for the apartments are from apartment buildings where measurements have been carried out in a number of apartments varying from 5 to 15. Due to average ventilation rates differing between the buildings, all measured ventilation rates in apartments have been normalized with the average ventilation rate of the respective building.

The scatter of the ventilation rates (one standard deviation) is about 20 % for apartments as well as for houses.

PROGRESS AND TRENDS IN AIR INFILTRATION
AND VENTILATION RESEARCH

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Poster 28

PUBLIC POLICY CONSIDERATIONS AND THE DEVELOPMENT OF A CODE FOR THE CONTROL OF RADON IN RESIDENCES

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SYNOPSIS

Building codes that address radon control in residential buildings are a relatively new development in the larger trend toward increased efforts to understand and control indoor air quality. A residential radon construction standard has been developed in the Pacific Northwest region of the United States. The Northwest Residential Radon Standard (NRRS) seeks to provide a measured public policy response that is commensurate with current knowledge of both the health risk and the state of building science. This paper reviews the range of potential public policy responses available to deal with radon as a public health problem, describes the policy framework upon which the NRRS is structured, and explains the development process.

Time and budget constraints limited the scope of the NRRS to identifying that minimum set of measures necessary to reliably achieve radon reductions without impairing structural integrity, capability to control other indoor air pollutants, occupant comfort, or energy efficiency. Though it looks more favorably at measures that enhance the linkages between durability, indoor air quality, and comfort; it does not require them unless they are part of the minimum set of requirements necessary for radon control. The NRRS, then, serves to provide a useful interim step toward the larger goal of a systemic approach.

1. INTRODUCTION

The NRRS was developed under the auspices of the Washington State Energy Office with funding support from the Bonneville Power Administration(BPA), a federal regional power-marketing agency.

Radon is an indoor pollutant requiring a different policy response than some other indoor contaminants. As an external pollutant source, radon is dependent on certain aspects of building science for control. There is a need for governmental intervention to increase public awareness of the issue, encourage voluntary action by individuals, and create the opportunity for individuals to live free of high radon exposures.

Ventilation and infiltration play a critical role in the design and construction of residential buildings capable of controlling many pollutants, including indoor radon concentrations. A whole systems approach which attempted to optimize residential buildings for durability, health and safety, comfort, and energy efficiency would include measures addressing envelope tightness, ventilation systems, and their pressure impacts.

Though the NRRS looks more favorably at measures that enhance the linkages between durability, indoor air quality, and comfort; it does not require them unless they are a part of the minimum set of requirements necessary for radon control. As a result, the potential for optimizing net system performance and cost is not impaired, but it is also not realized by the NRRS. Better control of radon is possible, but it requires broader dispersion of already available information, further development of technical support, supportive changes in other building codes, and the different emphasis of a whole systems approach.

2. THE REGIONAL CONTEXT

The Pacific Northwest region encompasses several states in the Northwestern United States. The region is blessed with a large hydroelectric resource which historically provided abundant low-cost electricity. In the past decade the region has taken aggressive steps to preserve its hydroelectric resource and avoid the cost of new electrical generation capacity. A major component of this effort has been the acquisition of energy efficiency in buildings -- a conservation resource.

In 1980, the U.S Congress established the Northwest Power Planning Council, a regional body mandated to develop a regional plan for ensuring adequate supplies of electrical energy. The initial plan (and subsequent revisions) have emphasized conservation as the most cost effective resource in the region.[1]

The Power Council's plan encourages the Bonneville Power Administration -- the agency which manages and distributes much of the region's electricity -- to pursue the conservation resource aggressively. This region has long been recognized for the pioneering work of BPA, both in the transmission of electricity and for the development of public power in the United States. More recently, the BPA has pioneered the development of conservation resources. It is now estimated that BPA has spent $1 billion (U.S.) on conservation programs, purchasing electrical energy savings at an average cost of $.02-.03 per Kwh saved.[2]

Over the past decade, BPA has supported (through participating publicly-owned electric utilities) the weatherization of homes that use electricity for space heating. In about 1980, as a component of its weatherization activities, BPA began to study indoor air quality in homes. Initially, restrictions on available weatherization measures were imposed pending an Environmental Impact Statement. Then, in 1984 the EIS was completed, restrictions were softened, and indoor air quality information was provided to all program participants.[3]

Radon testing was initiated as part of BPA's indoor air quality effort. Participating electric utilities tested residences throughout their service territories for radon levels. Measurements were made with alpha track monitors for a minimum of three months during the heating season. The result is one of the largest data sets ever collected on radon levels in residential buildings. Over 32,800 residential sites in approximately 400 townships were measured in Oregon, Washington, Idaho, Montana, and Wyoming (see Figure 1). The average measured radon concentration in roughly one-half of the 400 townships was greater than 37 Bq/m^3. None of the townships had an average measured radon concentration at or below 7.4 Bq/m^3, the new long-term national goal enacted by the U.S. Congress. A few areas of the region show a large number of homes with elevated radon test results. Notable, is the Spokane River Valley region on the border of Washington State and Idaho. In the City of Spokane, Washington nearly half the homes tested at levels above 150 Bq/m^3.

As part of the Pacific Northwest's aggressive pursuit of energy efficiency savings, the Northwest Power Planning Council developed the Model Conservation Standards (MCS) for the construction of new buildings. These standards require higher insulation levels in the building envelope, tighter building construction to reduce air leakage, ventilation provided by mechanical systems, and certain indoor air

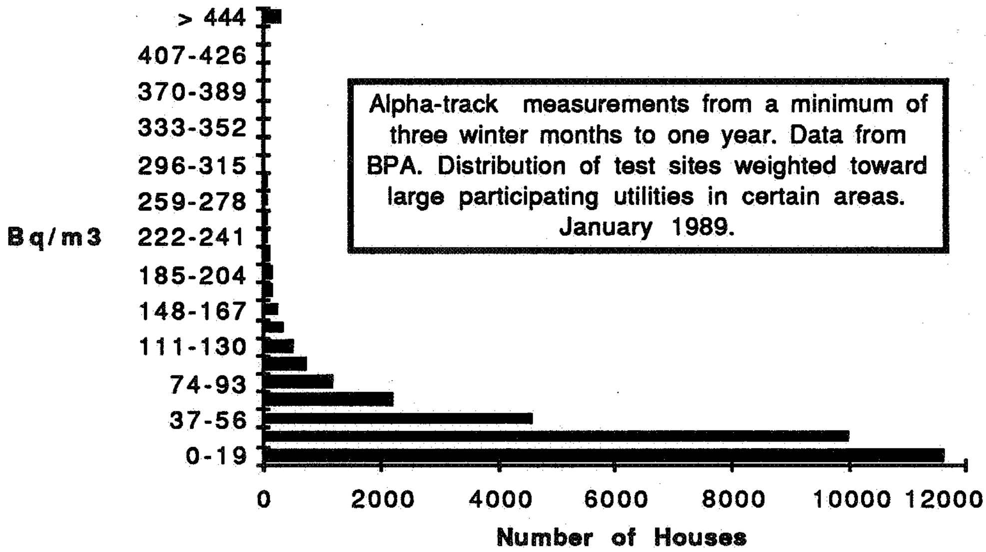

Figure 1. Distribution of 32,885 Radon Measurements in the PNW

pollutant control measures. Roughly 25 building code jurisdictions in the region have adopted the MCS.

The MCS are the first adopted and enforced standards in the U.S. that begin to address radon. In addition to specific requirements for sub-slab gravel and crawlspace ventilation, the MCS contain an appendix which specifies technical measures to be incorporated into certain residential buildings.

These standards implicitly recognize that more stringent energy codes do not necessarily create elevated radon levels. In fact they may provide opportunities to decrease the probability.

In 1987, BPA became interested in the development of a model radon code for new residential construction. In the summer of 1988, BPA contracted with the Washington State Energy Office's Energy Extension Service to research and develop a model radon code.

The Washington Energy Extension Service(WEES) has had an active public education program on indoor air quality for the past decade. When radon became an issue of public concern, WEES had been able to respond quickly with educational services. In a one year effort WEES developed the Northwest Residential Radon Standard.

3. A REGULATORY APPROACH - LOOKING FOR PRECEDENT

The task of determining an appropriate public policy response to the public health issue of radon presents interesting challenges. As an indoor air pollutant that largely originates from outside the building, radon is categorically different from many other indoor contaminants. It is not generated by occupant activity and it is not responsive, in large part, to behavioral adjustments by the occupant. In this light, radon appears as a more appropriate pollutant for some level of regulation.

The range of policy options for addressing public health threats are quite diverse (see Figure 2). At one end of the continuum, society does nothing. This, of course,

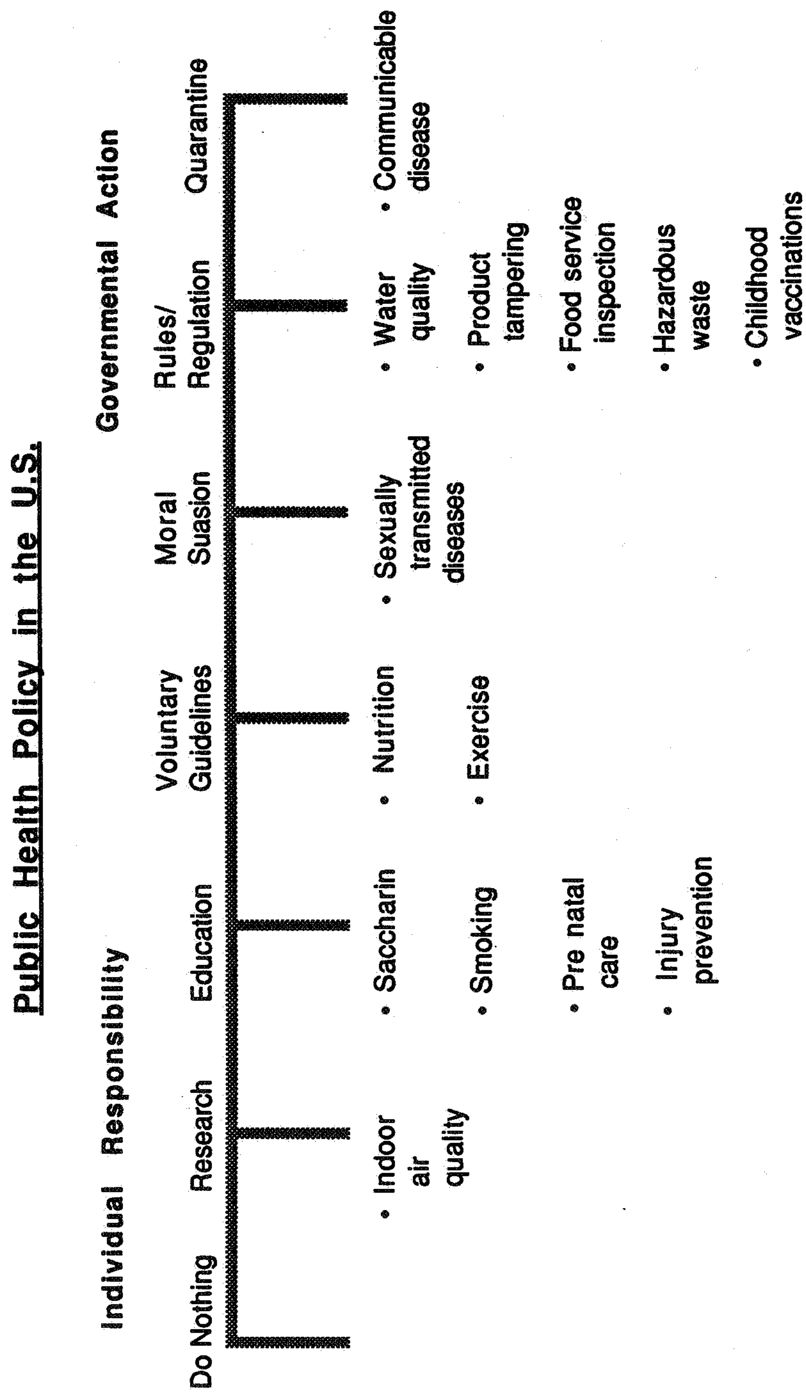

Figure 2. Public Health Policy Options in the U.S.

maximizes individual freedom, but may do little to protect the public good. At the other end of the continuum is the more draconian measure of quarantine, obviously reserved for only the most severe of public health threats. A number of potential policy responses exist along this continuum. Some possible responses include funding research, public education, expanding administrative efforts within existing regulations, moral suasion, and various degrees of restrictive rules and regulations.[4]

There are multiple levels of governmental response to health issues, and policy responses vary in the United States for different public health issues. It is illustrative to look at some health issues in light of the governmental response. Saccharin use and tobacco smoking rely almost exclusively on personal choice and public education (Though in the case of tobacco smoke, local communities are becoming more aggressive in regulating where the activity can take place). Childhood vaccinations and AIDS are two well publicized health issues that have received a stronger regulatory response. Though in both cases there remain elements of personal choice, the public health response has relied heavily upon administrative regulations (e.g. public schools require evidence of vaccination before enrolling children) and moral suasion. A most notable recent example was a produce tampering case (spring 1989) in which two Chilean grapes were found to be tainted with cyanide. The public health response was swift and aggressive: All Chilean produce was pulled from retail shelves throughout the United States.

4. THE SHARED RESPONSIBILITY VALUE

There has been very little regulatory control of radon in buildings in the United States. As such, the project required at the outset many value choices about both the technical structure and policy framework of the code. The fact that a code would be developed at all assumed that the problem was serious enough to warrant intervention by government - but at what point in the regulatory continuum?

WEES assumed public health in the area of radon is a shared responsibility between individuals and government. Unlike outdoor air quality (where the costs and benefits of clean air cannot be rationally apportioned to an individual, and attained through voluntary individual action[5]); the benefits that accrue to the individual from voluntary actions to maintain healthy indoor air are clear. WEES assumed it was the role of government to empower individual choice by providing:

- education about radon health effects, measurement, control, etc.
- access to necessary resources by nurturing the development of necessary technology.
- quality control through industry coordination and regulation.
- regulation necessary to provide the opportunity to live in healthier indoor air (including construction standards).

WEES assumed the individual's freedom of choice should be preserved to the extent possible, and that it was the individual's responsibility to:

- recognize the value of healthy indoor air.
- choose whether or not to live in it.

Therefore, the NRRS was structured as a governmental intervention that enables voluntary action. It regulates the building in order to enable radon control and preserve the individual's option to live in a healthier indoor environment. It stops short of requiring an individual to test or mitigate in order to continue to live in that environment. Because it is a construction standard, its scope is very focused and it addresses but one of several important regional and national issues with regard to radon and health.

5. SCIENTIFIC UNCERTAINTIES AND POLICY MOMENTUM

The issue of radon as an indoor air contaminant has a relatively brief history. In the U.S., an ever-increasing understanding of radon as a threat to public health has generated governmental activity at federal, state, and local levels.

At the federal level, the U.S.EPA issued action guidelines to the public for mitigation activity based on radon test results. As research more firmly established radon as a public health threat and as the public's awareness of the radon issue increased, governmental response also increased. The U.S. Environmental Protection Agency recommended the testing of all homes. The U.S. Surgeon-General issued a report on the health threat presented by radon. He encouraged all Americans to test their homes. The U.S. Congress recently passed legislation that provided funds to the states for radon programs, established a long-term national goal to lower radon levels in buildings to outdoor levels (7.4 Bq/m^3), and mandated the development of National Model Construction Standards by June 1990.

Despite increased levels of governmental activity in the area of radon, some uncertainty remains:

- estimates of the level of risk to human health at various exposure levels still vary.
- measurement protocols need improvement.
- we do not have long-term experience with the techniques of radon control and several technical questions remain unanswered (and probably unasked).

It is within this environment of scientific uncertainty and governmental desire to respond to the perceived threat, that the NRRS had to be developed.

WEES is confident that techniques required by the NRRS represent a reasonable and appropriate "good practice" standard at this time. It is redemonstrably evident that the radon control approaches required by the NRRS are very effective. Several radon mitigators utilize these techniques in the mitigation of existing residential buildings and guarantee their performance. However, it should be clearly understood that new information will likely emerge that results in the need for these measures to change.

6. THE VALUE OF A SYSTEMS APPROACH

The control measures required by the NRRS are intended to represent the minimum set of measures necessary to reliably achieve radon reductions <u>without impairing</u> structural integrity, capability to control other indoor air pollutants, occupant comfort, or energy efficiency. These measures are designed to mesh with current building practices, materials, and building codes. Hence, the NRRS requires:

1. practical techniques that reduce the number of openings available for soil gas transport to the indoor air.
2. a pressure difference control system designed to override other house/soil pressure differences contributing to soil gas transport.

However, it does not require:

1. as-tight-as-possible building envelope construction.
2. mechanical ventilation in all residential occupancies.
3. decoupling of all combustion appliances from the indoor air.
4. attention to pressure difference control in design of HVAC systems.

These additional measures serve multiple purposes and cannot always be justified for one purpose alone. For example:

- Mechanical ventilation (properly installed) would contribute to further reduction of indoor radon but its contribution is more than an order of magnitude less than that of the sub slab depressurization system capability required by the NRRS. Yet mechanical ventilation would enhance the control of other indoor air pollutants and increase occupant comfort, if installed in a tight house.
- Envelope tightness could reduce the volume rate of soil gas transport by enhancing pressure difference control capability at minimal energy cost. It would also enhance mechanical ventilation effectiveness, moisture control, comfort, and energy efficiency.

These and other measures could contribute significantly to further radon reductions. However, they would serve multiple purposes and the costs should be appropriately proportioned. A reciprocal effect is that part of the cost of the required radon control measures, such as substructure/crawlspace sealing and sub slab depressurization, could be charged to comfort, control of other indoor air pollutants, control of moisture(several tons/heating season from the soil), and control of other soil gas pollutants. (Jim White, of Canada Mortgage and Housing Corporation, reported that garbage gasses have been measured several kilometers away from land fill sites. Also some bacteria, fungi, and viruses found in soils can produce serious health problems[6]).

A whole systems approach which attempted to optimize residential buildings for durability, health and safety, comfort, and energy efficiency would include at least the additional measures listed above. Such an approach would further rationalize the cost of radon control. The increased durability, safety, comfort, and energy efficiency could increase the net value of residential buildings.

Because of these limitations the proposed NRRS is not an optimal standard. Better control of radon is possible, but it requires broader dispersion of already available information, further development of technical support, supportive changes in other building codes, and the different emphasis of a whole systems approach.

WEES is encouraged to think that the NRRS serves to provide a useful step toward the larger goal of a systemic approach.

7. INTRA-REGIONAL VARIABILITY

Radon exposures in some areas of the Pacific Northwest are relatively low: in some areas relatively high: in some areas unknown. It was an original intention that the NRRS would be offered to the region for optional adoption by local jurisdictions. Jurisdictions that were sufficiently concerned could adopt the NRRS.

8. FLEXIBILITY - THE ROLE OF A DUAL PATH STANDARD

The national model codes of the U.S., such as the Uniform Building Code, are performance codes. Performance codes specify levels of performance rather than specific materials or procedures. You must attain the end goal but are free to choose the means of attainment. Performance codes allow flexibility, cost optimization, and readily allow the development of new and improved materials and systems.

On the other hand, a prescriptive standard requires installation of certain materials and systems. It specifies a path that must be taken. Prescriptive standards/codes are simpler and easier to follow, but they lack the flexibility of performance codes, as well as the potentials for innovation and cost reduction.

The Council of American Building Officials (CABO), an umbrella organization of the three national model code organizations, distributes the CABO One and Two Family Dwelling Code. It is a prescriptive code. According to Dick Kuchnicki, President of CABO, the One and Two Family Dwelling Code was developed as a response to builders' requests for a prescriptive standard for residential construction.[7] Currently the National Association of Homebuilders favors a prescriptive standard for radon control if and only if it relieves builder liability. However, not all builders concur. Some would like to see a performance standard, because it allows them the flexibility to determine the most cost effective path.

Jim Gross, Deputy Director of the Center for Building Research, of the National Institute of Standards and Technology, has encouraged a dual path standard: a performance standard with the option of specified measures "deemed to satisfy" the standard.[8] This seems the most practical approach. The proposed NRRS follows this dual path pattern.

The NRRS seeks to provide increased protection for all new and significantly remodeled residential occupancies in any jurisdiction of the Pacific Northwest that chooses to adopt it. It seeks to limit exposure to indoor radon for occupants by requiring for every such occupancy either:

- demonstration of post-construction tested indoor radon levels at or below 150 Bq/m^3, or
- installation of certain specified materials and systems during construction that reduce the potential for elevated indoor radon and establish the capability to further reduce radon levels should the owner desire.

Option 1 (Chapter 3 of the NRRS) is a performance requirement. If the building does not meet the performance specification it must be modified until it does. There are no specified control requirements to be met during construction. It allows both flexibility and the demonstration and use of new and different approaches to controlling radon.

Option 2 (Chapter 4 of the NRRS) specifies certain prescriptive requirements, primarily substructure and crawlspace sealing, and the rough-in of a sub slab depressurization system. If the prescribed measures are correctly installed there are no future responsibilities for radon control.

9. NEED FOR A LONG-TERM MEASUREMENT TEST

The performance path of the NRRS requires verification that the performance goal has been reached. The intent is to ensure, within a reasonable level of certainty, that a building will perform as required.

The EPA's Interim Protocols for Screening and Follow-up Radon and Radon Decay Product Measurements state that "The EPA does not recommend taking any significant remedial action on the basis of a single screening measurement." [9] The screening measurement is a short term test.

The short-term test can be a reasonably accurate measure of the radon levels during the actual test period, but the range and period of variation is too great to enable a reasonably accurate measure of the long-term average radon levels. Arthur Scott of American Atcon Inc. has suggested that the decision level of a short-term (3 day charcoal for example) radon test is really very different from that of a long-term test (6 to 12 month alpha track), and that short-term tests are not being interpreted correctly. Short-term tests cannot predict long-term averages. He indicated that if a long-term average radon exposure is really 185 Bq/m^3, then the probability of a short-term test result of 37 Bq/m^3 is the same as the probability of a short-term test result of 750 Bq/m^3. [10]

Currently the short-term test is being misinterpreted in many sectors. William Ethier, an attorney for the National Association of Homebuilders, recently suggested at the National Radon Conference (Cincinnati, March 1989), that utilization of a short-term test to imply that radon levels are below 150 Bq/m^3 and therefore acceptable, could provide reasonable grounds for a claim of fraud or misrepresentation if a long-term test later showed levels over 150 Bq/m^3. According to Ethier, NAHB takes the position that short-term tests should not be part of a real estate transfer contract.[11]

In its report to the U.S. House of Representatives, the U.S. Committee on Energy and Commerce noted concern "about people making decisions not to mitigate based on low readings from short-term radon tests. Accordingly, the Committee expects EPA to evaluate the appropriate use of results from short- and long-term tests by the public. In particular, the Committee expects EPA to consider whether the Agency should recommend that only results from long-term tests should be used."[12]

The EPA screening protocol would be inappropriate for the NRRS, because of its reliance on short-term measurements. One NRRS reviewer suggested that a separate measurement protocol be developed, rather than rely on the EPA screening protocol. Another reviewer cautioned that developing a protocol outside that of the EPA might make it difficult to compare the results to measurements elsewhere.

A longer term measurement is necessary in order to attain a reasonable estimate of the building's actual performance and to avoid cheating by "smart" testers, who could affect results by coordinating test periods with rainfall, weather systems, and other factors. This requires addressing the additional difficulty of testing after occupancy. However there is a positive side to this: the occupant has the least incentive for fraud (unless he or she is preparing for resale).

The NRRS requires a long-term test by specifying adherence to certain EPA follow-up measurement protocols. According to the EPA's Interim Protocols for Screening and Follow-up Radon and Radon Decay Product Measurements, "The purpose of the follow-up measurement is to estimate the long-term average radon or radon decay product concentrations in general living areas with sufficient confidence to allow an informed decision to be made about risk and the need for remedial action." [13]

10. SHOULD WE REQUIRE MONITORING FOR ALL RESIDENTIAL BUILDINGS?

Currently the NRRS requires monitoring only for the performance path because it is the responsibility of the builder to meet the performance standard. It does not require monitoring for the prescriptive path because the builder completes his or her responsibility upon complying with specifications which are "deemed to satisfy" the standard. Once the builder has met the requirements of this standard his or her responsibilities have been completed. At this point the responsibility for addressing indoor radon is passed to the owner. The proposed NRRS stops short of governing the owner or occupant.

Neither compliance path guarantees that, for any given residential building, future indoor radon levels will be below 150 Bq/m^3. If a building has conformed to the prescriptive path, the owner or occupant will not know radon exposures until he or she tests. If a building has conformed to the performance path, there is no certainty that future events will not alter long-term average radon levels. Periodic measurements over the course of the useful life of any building, built to this standard, will be necessary if knowledge of radon exposures is desired.

For all governed buildings, the NRRS requires measures to:

- Inform all future occupants of the radon control measures taken.
- Strongly encourage them to test for radon.
- Provide them access to further information about health effects, testing, and mitigation.

Some NRRS reviewers recommended monitoring all new residential buildings. Other policy approaches were offered. For example, a member of EPA's National Radon Standards and Codes Work Group who has been involved in several mitigation demonstration projects, expressed a concern that the only workable way to reduce radon exposure in buildings is to have a standard that is at once both a performance and a prescriptive standard. Buildings would be built to specifications, tested, then mitigated if necessary. He felt that quality control was so essential, yet so lacking, that this approach might be necessary.

Testing following construction or remedial action, plus continued testing for several years afterward, is warranted by the lack of knowledge of:

- the short-term effects of specific measures in specific houses.
- the longevity of the effects of those measures.

WEES concluded that such follow-up testing should be encouraged (perhaps funded for research purposes), but not required.

11. THE NEED FOR EDUCATIONAL SERVICES

There is little system-wide coordination within the building industry. Many builders receive training on the job and then must make do with what they have learned from this rather local sphere of influence. There is significant variation in construction methods by both geographical area and climate.

In addition, builders must survive in an economic milieu in which emphasis on first costs forces builders to resist any increase in housing costs. Builders face a forest of regulation and will, in many cases, be less than eager to comply with additional regulations.

Educational and technical support services will be of significant value. While no radically new construction techniques are required, many are new to large portions of the residential sector.

An example is the Soil Gas Retarder Membrane required by the NRRS. Many reviewers supported its inclusion, considering it feasible and reasonable. Others were concerned about both the difficulty and cost incurred by having this technique as a requirement. It has become clear from several discussions that perceptions about this issue vary widely within the building trades.

Successful (and unsuccessful) experiences with the sub slab membrane are closely linked to perceptions about correct concrete practices, and these perceptions also seem to vary widely.

More stringent aggregate specifications and sealing techniques may also require educational services in the residential sector.

12. THE NRRS DEVELOPMENT PROCESS

Decisions about public health risks (in this case radon) can be extremely complicated. They involve elements of risk assessment, risk management, and risk communication. All too often, difficult decisions about risk assessment and risk management are made remotely by experts, then poorly communicated to the public. Often the result is conflict, with experts feeling misunderstood and the public feeling misused. Often both are right. Conflicts about health risk issues usually contain the underlying issues of equity and control. "Public participation" is usually too late, and does not involve the kind of information and power sharing necessary for the realization of enduring policies. Risk communication, with the goal of an actively concerned public, and within a context of real openness to public input, is vitally important. It may be difficult but it is both possible and necessary.

A good process can serve to align public perceptions with the perceptions of the scientific community. It can serve to eliminate the inappropriate extremes of either panic or apathy. It can empower a community with the sense that it can take charge and address the issues that confront it.

12.1 Participatory Process - Sequenced Input

The process for developing the NRRS was very participatory. Input was solicited from a diversity of economic sectors including realtors; builders and builder associations; technical specialists and generalists in the fields of building science, radon, and ventilation; consumer protection organizations; energy utilities; state and federal agencies; code organizations; and research organizations.

While broadly solicited, the input was sequenced: technical input was solicited first and the range of known technical solutions identified. Technical specifications had to meet criteria for control effectiveness, ease of implementation by typical tradesmen, availability of materials, cost, compatibility with comfort, and compatibility with other indoor air pollutant control techniques. Legal and policy related input followed.

12.2 Chronology - Initial Scoping

The effort began in June 1988. A literature search was conducted. Researchers, mitigators, and policymakers who were known to have radon-related experience were contacted by telephone. By July, 1988 referrals to additional resources had become very circular, and a sense of closure with regard to available national resources had developed.

The initial effort was very broad. Persons contacted were asked to identify and prioritize the radon issues that they perceived to be most important. They were then asked more specifically to identify those issues they thought were important to the development of construction standards for new residential construction, if they were aware of any efforts to develop construction standards, and if they knew of any standards already in place.

The U.S. Environmental Protection Agency is nationally recognized as the lead Federal resource for addressing indoor radon, and WEES assumed that all local, state, and regional efforts to address radon, particularly efforts to develop construction standards, would include communication with EPA regional offices. With assistance from EPA Region 10, all EPA regional radon representatives were contacted. They were all helpful, identifying issues of concern, key people, and any developing construction standards in their areas. Inquiries were also directed to the National Model Code organizations and the National Association of Homebuilders.

In August 1988 WEES visited and interviewed several radon researchers, mitigation contractors, and policymakers who were particularly knowledgeable about the techniques, costs, and policy issues pertinent to new residential construction. This included persons representing the National Association of Homebuilder's National Research Foundation; U.S. Geological Survey; U.S. EPA New Construction Division; New Jersey Department of Environmental Protection; Princeton University Center for Energy and Environmental Studies; Fairfax County, Virginia, a local jurisdiction actively addressing a known radon problem; Camroden Associates, a major radon research contractor; Garnet Homes, a large construction company voluntarily incorporating sophisticated radon control measures in all new home projects; R.F. Simon Co. and Buffalo Homes, two home construction contractors with significant radon mitigation experience.

WEES deliberately avoided the formulation of any specific code structure or provisions until the initial three months of research had been completed. In September, WEES attended, Reducing Radon In Structures, a three day technical training conducted by the U.S. EPA. After completion of this training the first tentative code design was formulated, internally reviewed, and gradually strengthened in technical detail.

12.3 Development Chronology - Technical and Legal Review

On October 1, 1988 an initial draft of the NRRS was completed and circulated for technical review. Circulation for legal review followed. More than 35 technical reviewers contributed comments about the initial draft. They included persons from the EPA; national research laboratories; university researchers; private sector builders, contractors, radon mitigators, tradesmen, engineers, architects and product suppliers; code officials and code organizations; builder associations; state energy offices; BPA; the Northwest Power Planning Council. The time allowed for the technical and legal review comment period had to be extended considerably longer than originally anticipated in order to obtain important and valuable review comments. The need for a longer review period may be in part due to the unanticipated intensity of activity in the radon industry in 1988, which included a national symposium, and the passage by the U.S. Congress on October 28, 1988 of the Indoor Radon Abatement Act which set a new national goal of indoor radon levels no higher than outdoors.

In January 1989, The U.S. Environmental Protection Agency asked WEES to contribute to EPA's effort to develop Model Construction Standards by June 1990 and partake in a National Radon Standards and Codes Work Group. The group included persons representing the national Model Code Organizations (ICBO, SBCCI, BOCA, and CABO), U.S HUD, National Institute of Building Sciences, National Institute of Standards and Technology, National Association of Home Builders, Canada Mortgage and Housing Corporation, members of an ASTM committee on radon, and representatives from states actively working on radon codes. In February WEES presented an introduction to the first draft of the NRRS to that group and received several constructive comments.

12.4 Development Chronology - Policy Review

The second draft of the NRRS was distributed March 30, 1989. It was circulated to a Policy Review Committee consisting of state and local officials in the general government, building code, and public health areas; policy level representatives from BPA, the Northwest Power Planning Council, utilities, and the shelter industry; the EPA National Radon Standards and Codes Work Group; the National Institute of Standards and Technology; the Canada Mortgage and Housing Corporation. The second draft was also recirculated to technical and legal reviewers as a courtesy copy.

The first draft of a generic Implementation Plan was completed in May and circulated for review by a local advisory committee. The Implementation Plan is a guidance document intended to assist local jurisdictions with considering, adopting and implementing the NRRS. The plan seeks to provide the conceptual framework for a reasonable, equitable, and informed process for consideration of the NRRS. It is not meant to encourage adoption of the NRRS. The intent is to encourage and enable a good choice.

The final Implementation Plan and final draft of the NRRS were completed in June 1989.

13. APPENDIX

The chart in this appendix outlines the structural organization of the NRRS. The NRRS is organized into five chapters. Major topics addressed by each chapter are detailed:

APPENDIX -- ORGANIZATIONAL OUTLINE OF THE NORTHWEST RESIDENTIAL RADON STANDARD

Chapter 1
ADMIN. & ENFORCEMENT
- Title Intent & Scope
- Materials & Equipment
- Alternate Materials & Methods
- Plans & Specifications
- Enforcement, Inspections, & Fees
- Validity
- Violations
- Liability

Chapter 2
DEFINITIONS

Chapter 3
PERFORMANCE PATH
- Scope
- Financial Assurance Requirement
- Measurement
- Mitigation
- Verification of Mitigation
- Notification
- Written Description

Chapter 4
PRESCRIPTIVE PATH
- Scope
- Floors in Contact With the Earth
  - aggregate
  - bulk water drainage
  - concrete floors
    - placement
    - strength
    - workability
    - air entrainment
    - curing
    - control joints
    - sealing joints/cracks
    - soil gas retarder
  - other floor assemblies
  - passive sub-slab depressurization
- Masonry Block Walls
  - continuous solid course
  - mortar joints
  - penetrations
  - waterproofing
  - continuous air barrier
- Concrete Walls
  - workability
  - sealing penetrations/ joints/cracks
  - dampproofing
- Crawlspaces
  - continuously vented
  - intermittently vented
  - plenums
- Combinations of Floor Assemblies
- HVAC Systems
  - ductwork
  - condensate drains
- Sub-Slab Depressurization System
  - sealed pipe
  - perforated pipe
  - pipe characteristics
  - condensation/stack assist
  - fan location
  - fan wiring/circuit/label
  - pipe identification
  - system description

Chapter 5
REFERENCED STANDARDS & DOCUMENTS
- Occupant Notice Document
- Referenced Documents

14. ACKNOWLEDGEMENTS

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